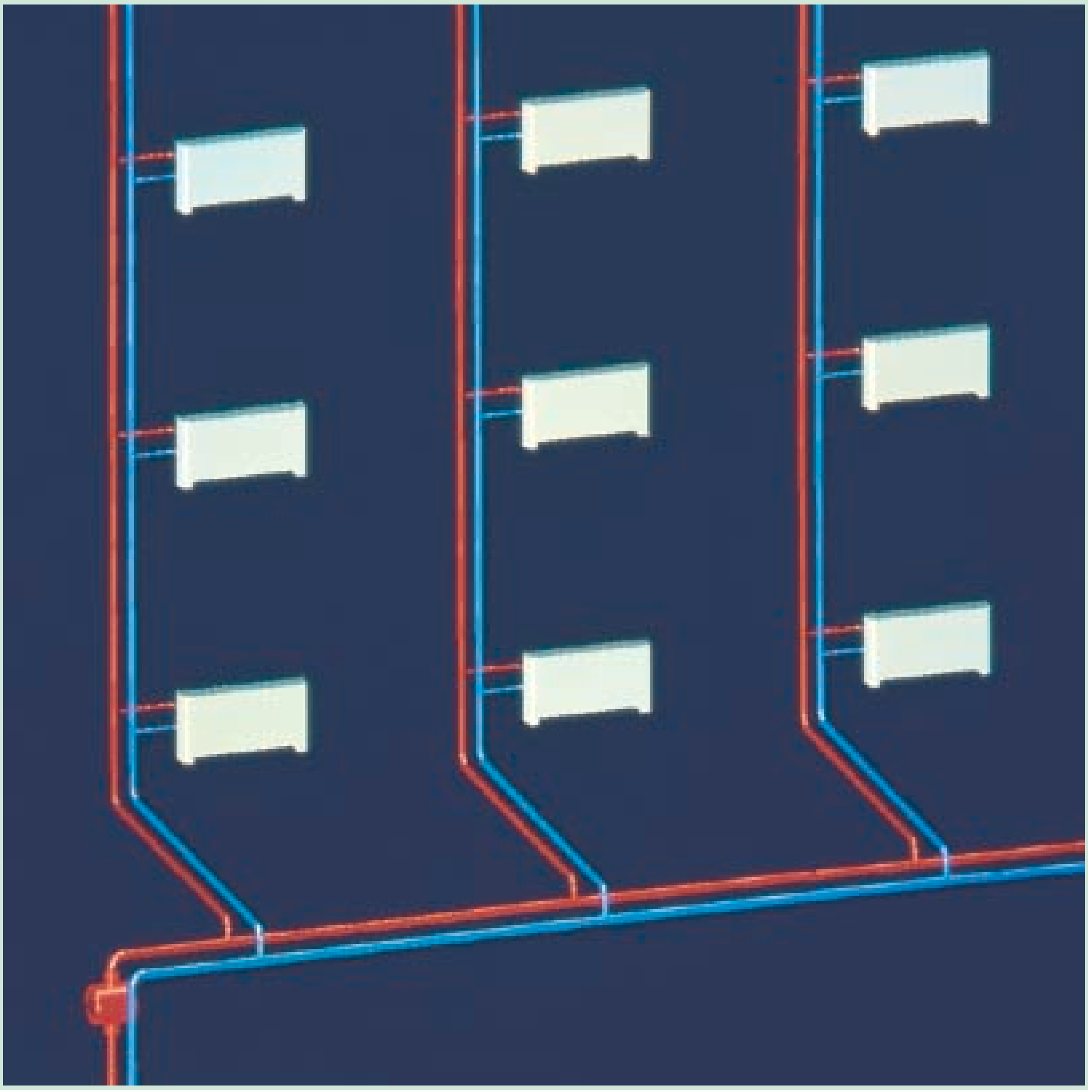
MARIO DONINELLI

DESIGN PRINCIPLES OF HYDRONIC HEATING SYSTEMS



Handbooks Caleffi

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Realization and printing

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MARIO DONINELLI

DESIGN PRINCIPLES OF HYDRONIC HEATING SYSTEMS

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The publication of this second Caleffi manual gives me the opportunity of thanking all those who, after the publication of the first Manual, have supported our initiative with their comments, advice and messages of encouragement.

I must confess that the project has given me a great deal of pleasure even though it si not an easy one. Our aim is not to publish “all” types of books but to bring out new publications designed to meet the specific requirements of those who design and manufacture heating and cooling systems. It is for this reason that we have placed the compilation in the hands of professionals who have wide experience in their particular fields. We feel that it is their direct contact with everyday situations which is essential for highlighting the real problems of heating engineers and for provinding sensible solutions and useful suggestions.

We intend therefore to concentrate on the clarity of the text and the graphic quality of our manuals. BUSY engineers do not have time to work through books which are not clear or have no practical use. We consider it our duty to do everything possible to simplify their design work, and facilitate their obtaining information, by providing clear diagrams, drawings which are easy to follow and tables which are easy to consult. In other words we want to cover the technical details with the same, at times almost fanatical, care with which we design and manufacture our products. It is this commitment to the care we take over our products and our constant desire to improve quality which is the main characteristic of our method of working with those who have, and those who will, place their trust in our company.

Finally I should like to express my thanks to Engineer Mario Doninelli who is also the author of this second Manual. A long friendship and the same spirit of dedication make it easy for us to co-operate harmoniously.

Naturally I shall be very grateful to those who would like to let us have their opinions or advice on this new publication. Comments of this type, apart from providing an important audit of our work, also provide a significant opportunity for our getting to know those from outside company that we at Caleffi consider important.

Franco Caleffi
Chairman of CALEFFI S.p.A.

In this second manual I have rearranged and updated the text of my working files relating to heating circuits and heat emitters.

In the section covering the circuits I felt it was desirable to highlight the numerical examples so that notes and observations of general interest could be drawn from them. I have chosen to avoid hypothetical situations and to base my book on actual systems which are suitable for demonstrating the essential technical aspects relating to the choice and design of the circuits.

I have given a number of examples of a theoretical nature in order to show how the parameters involved interact. To know how to assess the true "significance" of these factors makes it possible to develop an analytical attitude towards the simplified hypotheses and approximations which are adopted.

In addition examples have been included to demonstrate the use of the proposed practical procedures within the scope of the individual items. With procedures of this type it has been possible to significantly reduce the effort required for theoretical calculations.

In the second section of the manual I have examined the heat emitters, analysing them from the following points of view:

1. *structural and performance characteristics*
2. *heat output provided under various operating conditions*
3. *commissioning and maintenance requirements.*

This has been achieved by analysing the main factors governing the choice of type and size, position and maintenance of the emitters.

I should like to thank Paolo Barcellini of Caleffi and my friends in the STC office, Umberto Bianchini and Roger Brescianini for the invaluable assistance they have given me.

I should also like to thank the Caleffi company who provided me with everything I needed to complete this manual and have it printed.

Mario Doninelli

INTRODUCTION

GENERAL LAYOUT

Definitions, graphs, tables, formulae, examples and advice have been collected continuously in the form of headings (or cards). In the first section of the book the headings have been arranged in a logical sequence as shown in the summary on page 2. In the second section however the headings are arranged in albhabetical order.

Every heading is, practically speaking, self-sufficient. The connection between one heading and another is indicated by suitable cross references. Each cross reference is clearly indicated and included between brackets.

Graphs, tables and formulae have been numbered but these are only relevant within the section in wich they occur. Important subject headings, often introduced by a brief family tree, are subdivided into chapters and sub chapters.

DIAGRAMS AND DRAWINGS

The sections are complemented by diagrams and drawings which illustrate the functional aspect of the systems, equipment and components described. Working technical drawings are not included.

SIGNS, SYMBOLS AND ABBREVIATIONS

Signs and symbols (mathematical, physical and chemical symbols etc.) are those in current use. I have tried to avoid the use of abbreviations as much as possible and those which have been used are identified on a case by case basis.

UNITS OF MEASUREMENT

The International System has not been strictly applied in the text and frequently other units of measurement have been used for preference where:

1. they are more comprehensible from a practical point of view;

2. they are the effective units of measurement which form part of the working language of engineers and installers.

GREEK ALPHABET

Physical quantities, numerical coefficients and constants are frequently represented by letters of the Greek alphabet. I felt it was desirable therefore to give below these letters and their relative pronunciation.

Letters of the Greek alphabet					
Capital	Small Case	Name	Capital	Small Case	Name
Α	α	alfa	Ν	ν	ni o nu
Β	β	beta	Ξ	ξ	xi
Γ	γ	gamma	Ο	ο	omicron
Δ	δ	delta	Π	π	pi
Ε	ε	epsilon	Ρ	ρ	rho
Ζ	ζ	zeta	Σ	σ	sigma
Η	η	eta	Τ	τ	tau
Θ	θ	theta	Υ	υ	upsilon
Ι	ι	iota	Φ	φ	fi
Κ	κ	cappa	Χ	χ	chi
Λ	λ	lambda	Ψ	ψ	psi
Μ	μ	mi o mu	Ω	ω	omega

NOTES

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HYDRONIC HEATING AND AIR CONDITIONING SYSTEM CIRCUITS

Summary of contents

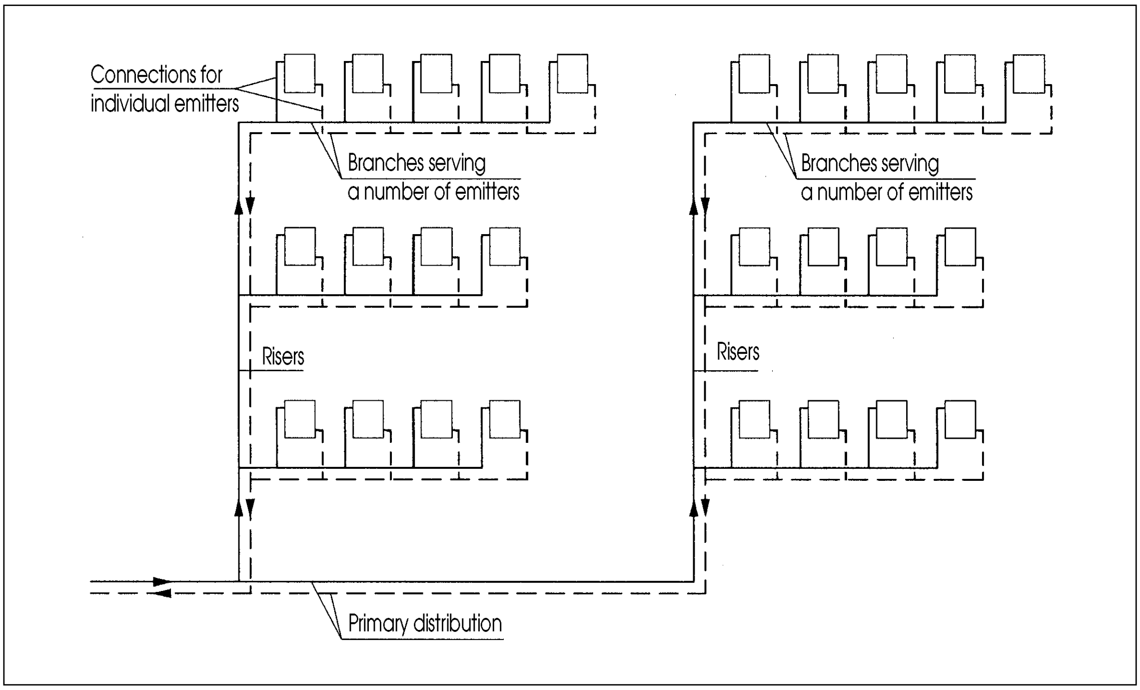
INTRODUCTORY NOTES

DEFINITIONS

Main circuit (circuit under consideration):
This circuit is the one which is being investigated and for which the dimensions are being calculated.

Secondary circuit:
This is any branch circuit which is served directly by the main circuit, e.g. the following apply in relation to the diagram below:

1. **When the dimensions of branches to a number of emitters are to be calculated:** the branches themselves are considered to be the main circuits, whilst the secondary circuits are the connections to the individual emitters.
2. **When the dimensions of the risers are to be calculated:** the risers themselves are considered to be the main circuits, whilst the secondary circuits are the branches to a number of emitters
3. **When the dimensions of the primary distribution pipework is to be calculated:** the circuit itself is considered to be the main circuit, whilst the secondary circuits are the risers.



Nominal flow rate through an emitter

This is the flow rate passing through the emitter under the test conditions, used to determine its heat output (nominal output).

Automatic control valves:

These are valves which automatically regulate the quantity of fluid which passes through them. This may occur for example as the result of a change in pressure - or temperature.

DESIGN CALCULATIONS FOR PIPE SIZING

Pipe sizing calculations are carried out as follows, using the constant linear head losses method.
As a guide it is assumed that **r** = 10 mm w.g./m which represents a good compremise between:

- **minimising system material and installation costs** and
- **limiting the energy consumption of circulating pumps.**

REFERENCE TABLES

TABLE 1 - **COEFFICIENT OF LOCALISED LOSSES**
(1st Manual LOCALISED PRESSURE LOSSES section)

TABLE 2 - **LOCALISED PRESSURE LOSSES**
(1st Manual LOCALISED PRESSURE LOSSES section)

TABLE 5 - **PRESSURE LOSSES IN STEEL TUBES**
(1st Manual STEEL TUBES section)

Note:
These tables are collected at the end of this book.

BALANCED CIRCUITS

Balanced circuits are defined as those circuits which are capable of supplying their emitters with the correct fluid flow rate, i.e. the flow rate required such that **the emitters can be operated correctly in accordance with the design specification.**

The design and installation of balanced circuits is used:

- **to ensure the correct performance of the emitters;**
- **to prevent excessive water velocities** which can cause noise and abrasion;
- **to prevent circulating pumps from operating under low efficiency conditions, leading to overheating;**
- **to limit the value of the differential pressures acting on the regulating valves** thus preventing hunting.

On medium to small systems with a constant flow rate satisfactory sizing of the pipework will normall be adquate to ensure that the circuits are balanced.

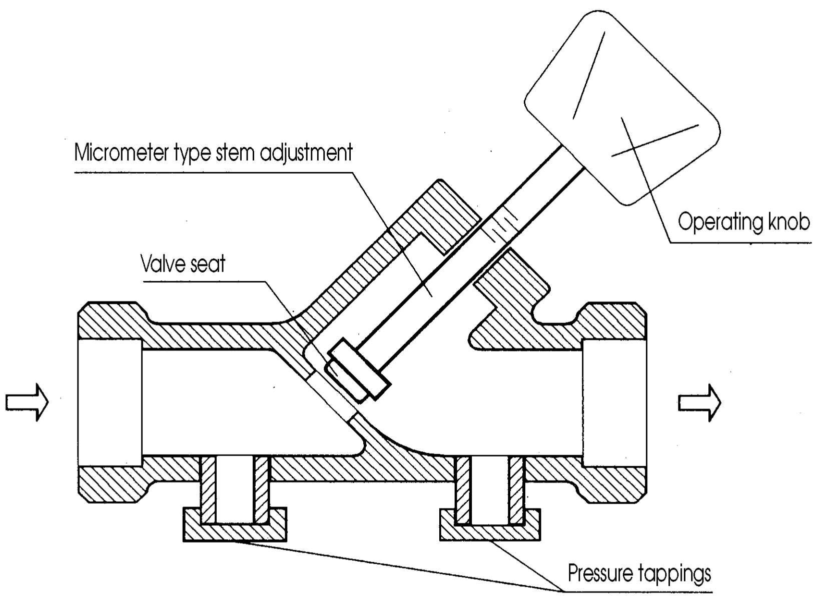
Conversely, **on systems with extensive networks, or with variable flow rates,** in order to achieve balanced circuits **it is necessary to include equipment capable of regulating the flow of water.** The types, and main characteristics, of various types of suitable equipment are analysed below.

BALANCING VALVES

These are valves through which the flow of water can be adjusted with known pressure losses so that it is possible to regulate the flow passing through the branches on which they are located.

The most important parts of these valves are: **the valve seat, the stem and the pressure tappings.**

The valve seat must be capable of providing a regular and uniform flow so as to prevent vibration, noise, cavitation phenomena and wear on the seat and/or gaskets. **The stem** should have micrometer type adjustment (i.e. with a fine thread) and reference points capable of providing accurate positioning and control of the valve seat. **The presssure tappings** must abe in areas of low turbulence and mounted in such a way as to enable accurate "in situ" measurements of the resistance of the valve to the fluid flow to be taken.

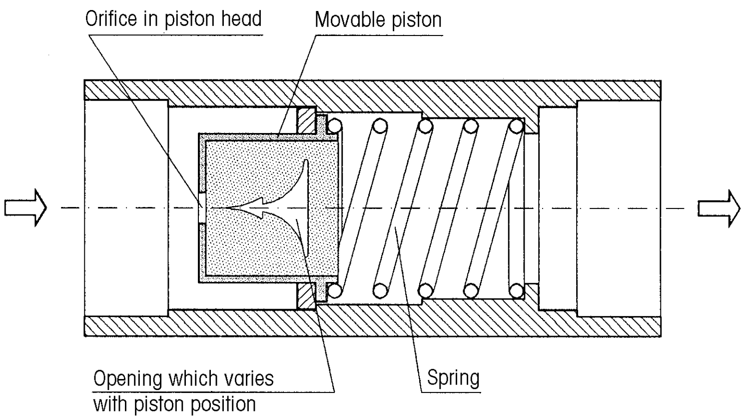


Balancing valve

AUTOFLOW REGULATORS

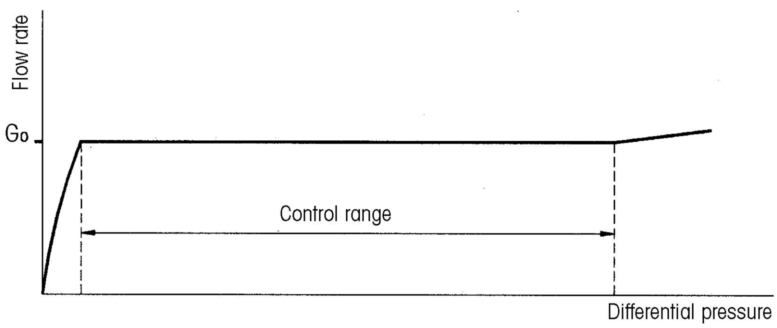
These are valves capable of maintaining automatically a constant water flow rate through the branches on which they are located.

The regulating element of these valves is a springloaded piston with an end port and variable side orifice for the fluid passage.



Autoflow regulator

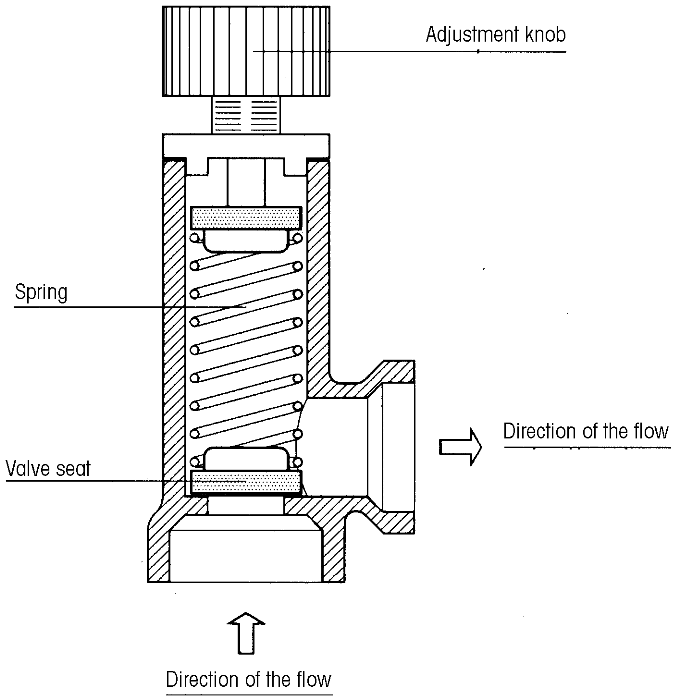
The valve is adjusted by the pressure of the water working against the helical spring, **which ensures automatically that the flow rates are virtually constant over a wide range af differential pressures** as is shown below:



AUTOMATIC BYPASS VALVES

These valves are used to control water flow into a bypass by means of differential pressure. A bypass may be necessary to prevent the differential pressure across a circuit exceeding the design value. This may occur when a number of emitters, or branch circuit, shuts down.

The regulating element of these valves is a seat normally closed by spring action, adjustable (relating to maximum differential pressure) by means of the knob provided. **The valve opens and activates the bypass circuit** (discharging the excess pressures) **only when it is subjected to a thrust greater than that of the spring.**



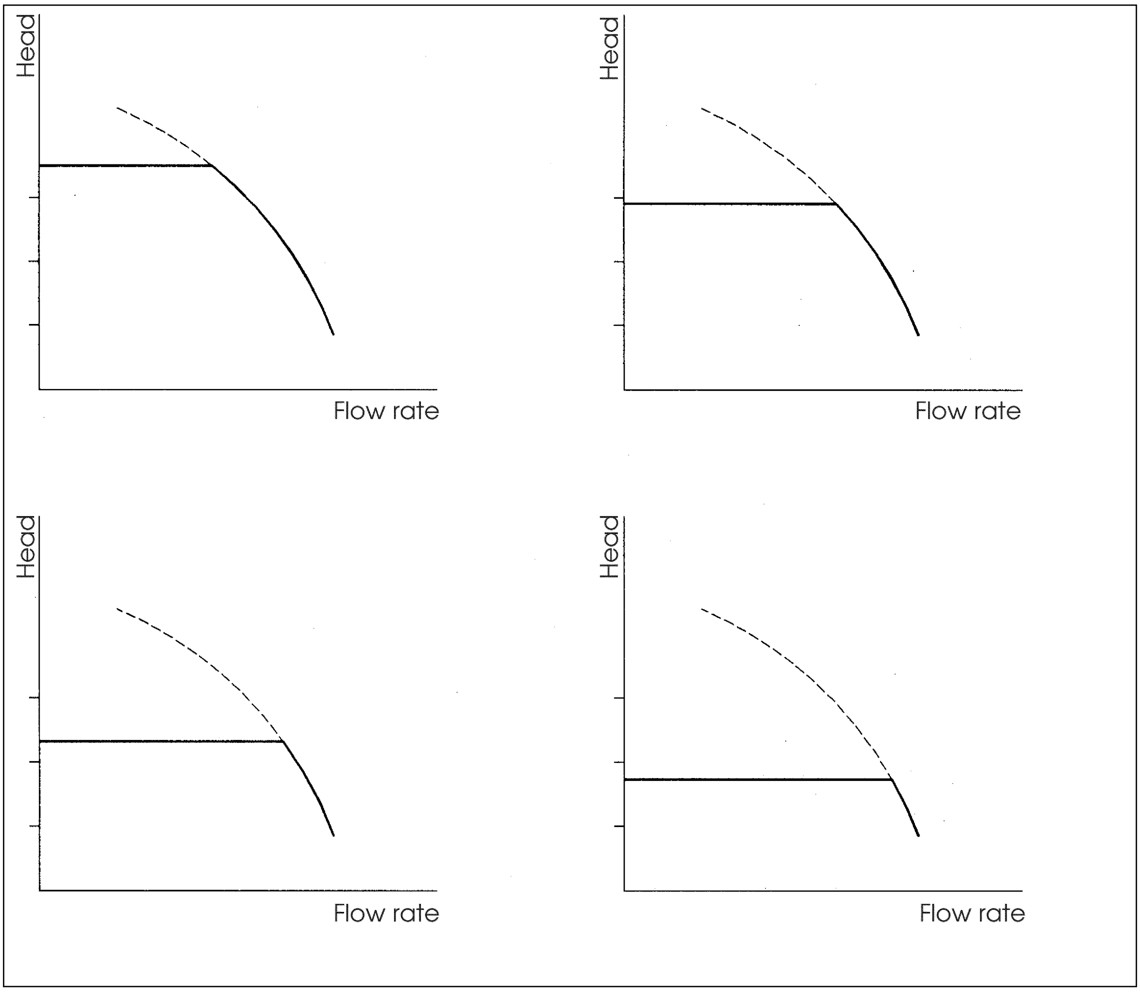
Automatic bypass valve

DIFFERENTIAL PRESSURE CONTROLLERS FOR CIRCULATING PUMPS

These controllers are used to adjust automatically the speed of the circulating pumps and ensure that the differential pressure (between two points of a circuit) **does not exceed the design value.**

A **control panel** is provided and **pressure tappings** may be at the pump or remote from it, depending on requirements.

The characteristic curves of a circulating pump operating under the control of a differential pressure controller calibrated to various maximum pressure levels are shown below.



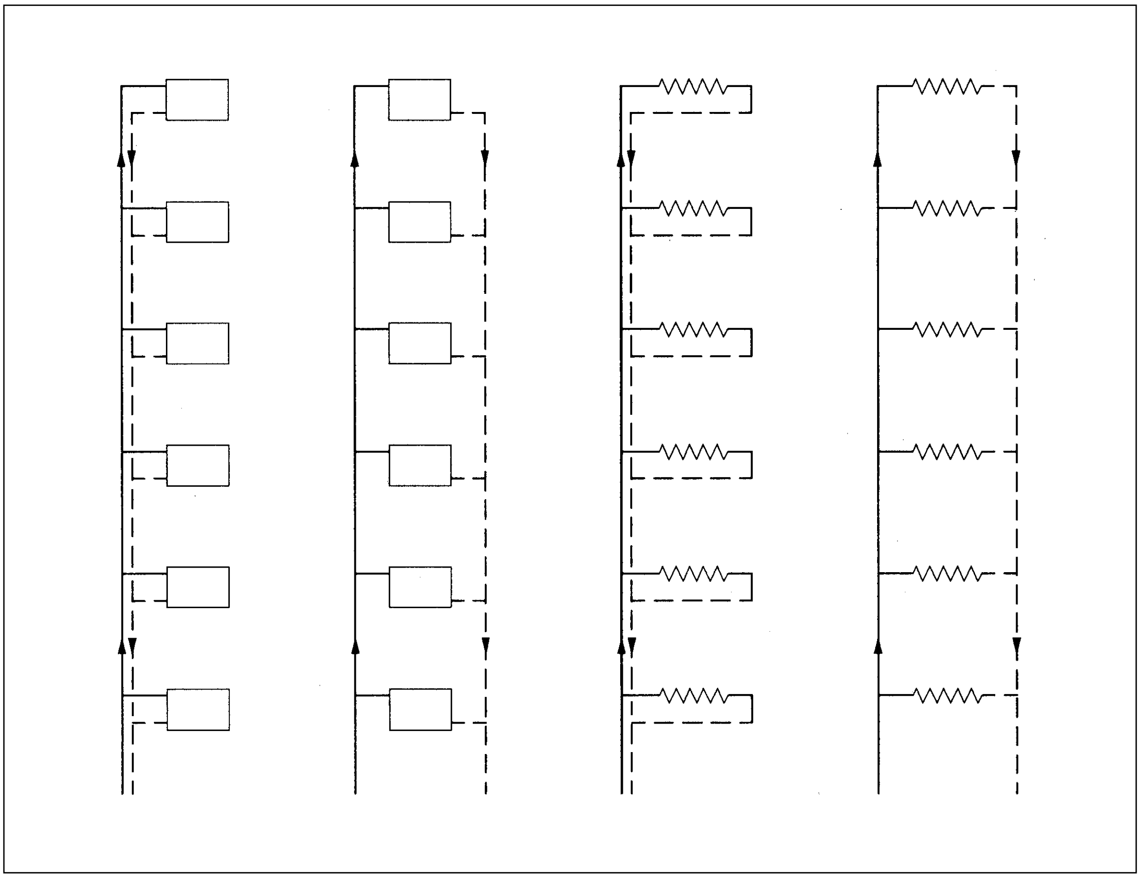
SIMPLE CIRCUITS

THEORETICAL CALCULATION OF SIMPLE CIRCUITS

PRACTICAL CALCULATION OF SIMPLE CIRCUITS

CHARACTERISTICS OF SIMPLE CIRCUITS

APPLICATIONS OF SIMPLE CIRCUITS



There are two pipe circuits without balancing valves or autoflow regulators. They are shown in the diagrams on the previous page.

CALCULATIONS FOR SIMPLE CIRCUITS

The calculation for these circuits are made as follows:

1. **The dimensions of the final secondary circuit are calculated,** based on the required flow rate, by determining:
   - **the diameter** and
   - **the pressure loss**

2. **The dimensions of the final sections of the main circuit** (i.e. those between the final and penultimate secondary circuit) are calculated by determining:
   - **the flow rate** (this is equal to that of the final secondary circuit),
   - **the diameter** and
   - **the pressure loss**

3. **The dimensions of the penultimate secondary circuit are calculated, based on the required flow rate,** by detemining:
   - **the diameter** and
   - **the pressure loss**

   The flow rate and pressure loss determined **in this way should balance with the effective pressure available at the circuit connections.**

4. **The dimensions of the penultimate sections of the main circuit are calculated** by determining:
   - **the flow rate** (this is obtained by adding the flow rates of the secondary circuits served by the sections under examination),
   - **the diameter** and
   - **the pressure loss**

5. **The dimensions of the other secondary circuits and the other sections of the main circuit are determined as follows:**

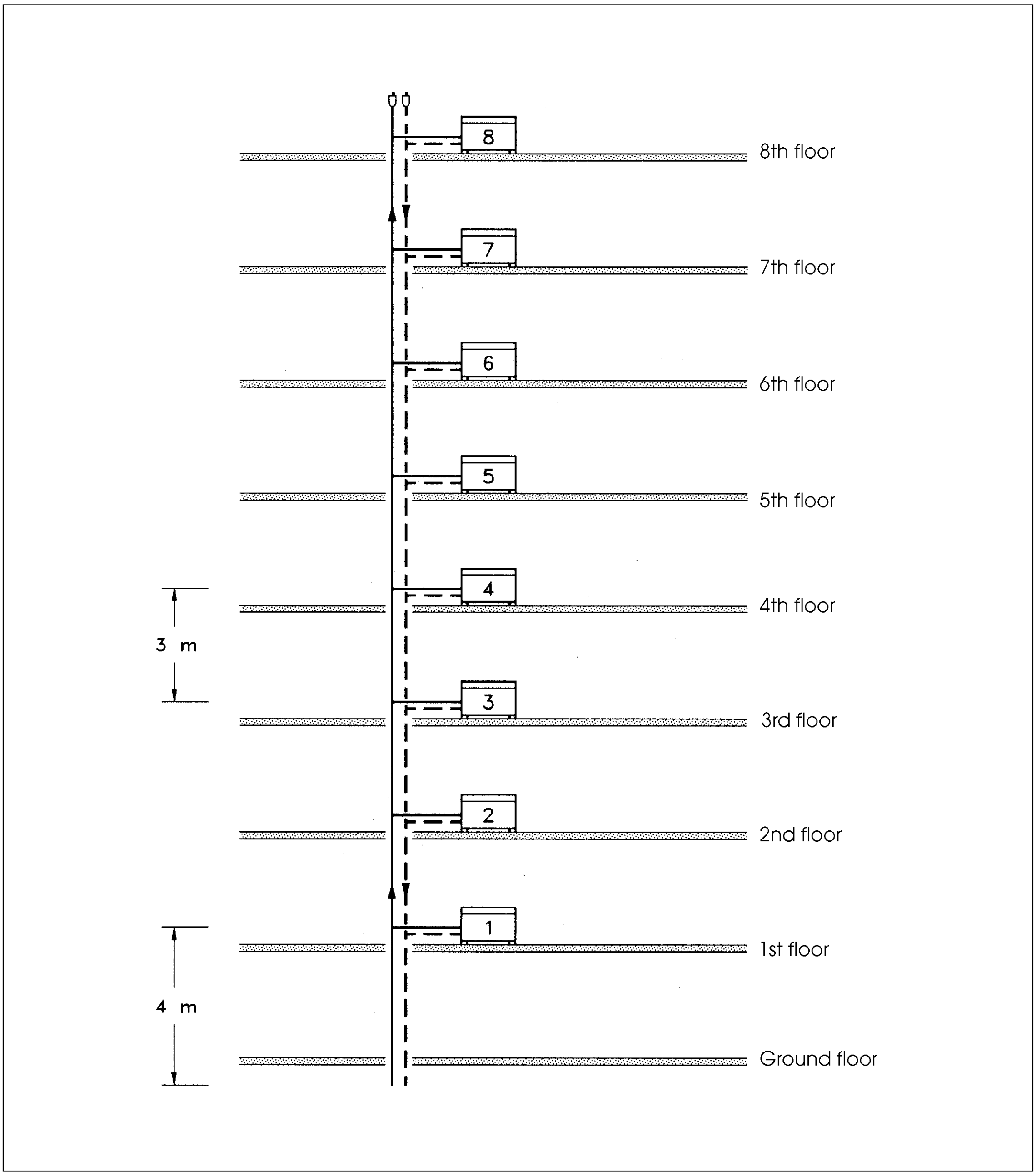
   – Proceed as indicated in 3 above **for the secondary circuits.**

   – Proceed as indicated in 4 above for the sections **of the main circuit.**

Example 1 - Calculation of a simple circuit

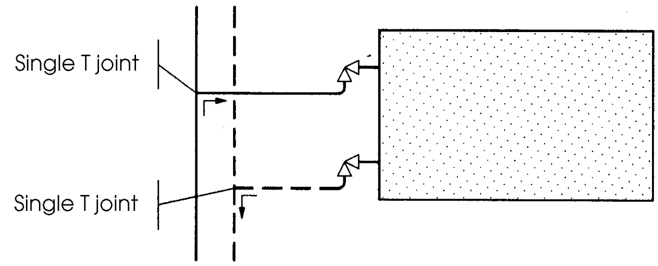
Using the method of calculation shown on the previous page determine the dimensions of the simple circuit illustrated bellow. The following factors are taken into account:

- G = 330 l/h nominal flow rate through each fan coil unit,
- ΔP = 150 mm w.g. pressure loss corresponding to the nominal flow rate,
- l = 4 m lenght of pipework between the riser and the fan coil unit,
- n = 2 (at 90°) number of bends in the pipework.



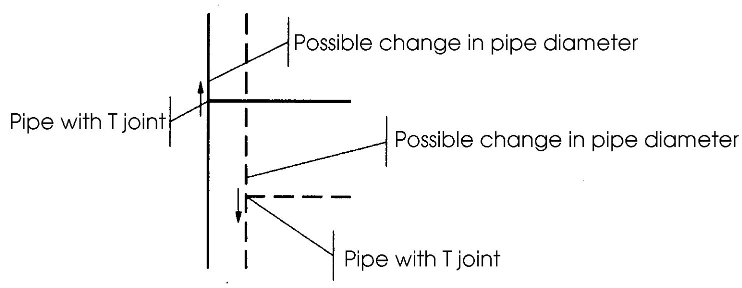
Solution:

— Pressure losses through fittings and valves in the fan coil - riser connecting circuit. Determination of ξ coefficient (localised pressure loss coefficient):



- 1 single T joint	1,0	=	1,0				1,0	
- 1 single T joint	1,0	=	1,0				1,0	
- 2 normal 90° bends	2 · 1,5	=	3,0	(ø = 3/8“, 1/2”)	2 · 1,0	=	2,0	(ø = 3/4“, 1”)
- 1 angle valve (mean value)	4,0	=	4,0				4,0	
- 1 lockshield (mean value)	1,0	=	1,0				1,0	
	Total Σξ	=	10,0	(ø = 3/8“, 1/2”)	Total Σξ	=	9,0	(ø = 3/4“, 1”)

— Determining the ξ values relating to the pipework between floors:



- 1 pipe with a single T joint	1,0
- 1 pipe with a single T joint	1,0
	Total Σξ = 2,0 (sum of ξ values in the case of constant pipe ø)
- 1 pipe diameter increase	1,0
- 1 pipe diameter decrease	0,5
	Total Σξ = 3,5 (sum of ξ values in the case of change in pipe ø)

— To calculate the pipework size assume that: r = 10 mm w.g./m (See INTRODUCTORY NOTES)

— Reference tables (collected at the end of this book):
TABLE 2 - localised pressure losses (1st. Manual LOCALISED PRESSURE LOSSES section)
TABLE 5 - Pressure losses in steel tubes (1st. Manual STEEL TUBES section)

Secondary circuit of the fan coil unit - floor 8

It is necessary to ensure that fan coil unit 8 has its nominal flow rate: G_8 = 330 l/h. This flow rate can be obtained using 1/2“ connections which give a fluid velocity (0,44 m/s) of less than 0.7 m/s, i.e. the limit recommended in TABLE. 1 under the VELOCITY heading (1st Manual).

— Calculation of the pressure loss (H_8) of the secondary circuit:

- Pressure loss distributed along the branches. These are calculated using the formula: $h = l \cdot r$.
  Where: l = 4 m (lenght of branches)
  and r = 20,5 mm w.g./m (TAB. 5, for ø = 1/2” and G = 330 l/h)
  giving: h = 82,0 mm w.g.

- Localised pressure loss of the branches. These are calculated using TAB. 2.
  Where: $\Sigma\xi$ = 10,0
  and v = 0,44 m/s (TAB. 5, for ø = 1/2” and G = 330 l/h)
  giving z = 96 mm w.g. (TAB. 2)

- Pressure loss of the fan coil unit (k):
  k = 150 mm w.g. at the nominal flow rate of the fan coil unit.

  Therefore: $H_8 = h + z + k = 82 + 96 + 150 = 328$ mm w.g.

Pipework between floors 8 and 7

— Flow rate of pipework: $G_{8\text{-}7} = G_8$ = 330 l/h

— ø selected: = 3/4” (the commercially available diameter nearest the guide value: r = 10 mm w.g./m)

— Calculation of the head loss ($\Delta P_{8\text{-}7}$) of the pipework:

- Continuous head losses. These are calculated using the formula: $h = l \cdot r$.
  Where: l = 6 m (length of the pipework)
  and r = 5 mm w.g./m (TAB. 5, for ø = 3/4” and G = 330 l/h)
  giving: h = 30 mm w.g.

- Localised head losses. These are calculated using TAB. 2.
  Where: $\Sigma\xi$ = 2,0
  v = 0,25 m/s (TAB. 5, for ø = 3/4“ and G = 330 l/h)
  giving z = 6 mm w.g. (TAB. 2)

  Therefore: $\Delta P_{8\text{-}7} = h + z = 30 + 6 = 36$ mm w.g.

Secondary circuit of fan coil unit - floor 7

With the same flow rate (nominal flow rate) and the same diameter (3/8" would give an excessive velocity), this circuit is equal to that of fan coil unit 8. Consequently the effective flow rate of the circuit being considered can be determined by balancing G_8 and H_8 at the effective head (H_7) available at its connections (See 1st Manual - BALANCING FLOW RATE).

— Head at the connections of the circuit under examination:
$H_7 = H_8 + \Delta P_{8\text{-}7} = 328 + 36 = 364$ mm w.g.

— Effective flow rate of the circuit under consideration:
$G_7 = G_8 \cdot (H_7 / H_8)^{0,525} = 330 \cdot (364 / 328)^{0,525} = 349$ l/h $v_7 = 0{,}47$ m/s

Pipework between floors 7 and 6

— Flow rate in the pipes: $G_{7\text{-}6} = G_{8\text{-}7} + G_7 = 330 + 349 = 679$ l/h   Chosen diameter ø = 3/4"
$r = 18{,}5$ mm w.g./m $v = 0{,}51$ m/s $\Sigma\xi = 2{,}0$ (no variation in diameter)

— Pressure loss: $\Delta P_{7\text{-}6} = 18{,}5 \cdot 6 + 26 = 137$ mm w.g.

Secondary circuit of fan coil unit - floor 6

— $H_6 = H_7 + \Delta P_{7\text{-}6*} = 364 + 137 = 501$ mm w.g.
— $G_6 = G_8 \cdot (H_6 / H_8)^{0,525} = 330 \cdot (501 / 328)^{0,525} = 412$ l/h; $v_6 = 0{,}55$ m/s

Pipework between floors 6 and 5

— Flow rate in the pipes: $G_{6\text{-}5} = G_{7\text{-}6} + G_6 = 679 + 412 = 1.091$ l/h   Chosen diameter ø = 1"
$r = 14{,}0$ mm w.g./m $v = 0{,}52$ m/s $\Sigma\xi = 3{,}5$ (variation in diameter)

— Pressure loss: $\Delta P_{6\text{-}5} = 14{,}0 \cdot 6 + 47 = 131$ mm w.g.

Secondary circuit of fan coil unit - floor 5

— $H_5 = H_6 + \Delta P_{6\text{-}5} = 501 + 131 = 632$ mm w.g.
— $G_5 = G_8 \cdot (H_5 / H_8)^{0,525} = 330 \cdot (632 / 328)^{0,525} = 466$ l/h $v_5 = 0{,}63$ m/s

Pipework between floors 5 and 4

— Flow rate in the pipes: $G_{5\text{-}4} = G_{6\text{-}5} + G_5 = 1.091 + 466 = 1.557$ l/h   Chosen diameter ø = 1 1/4"
$r = 7{,}0$ mm w.g./m; $v = 0{,}43$ m/s $\Sigma\xi = 3{,}5$ (variation in diameter)

— Pressure loss: $\Delta P_{5\text{-}4} = 7{,}0 \cdot 6 + 32 = 74$ mm w.g.

Secondary circuit of fan coil unit - floor 4

— $H_4 = H_5 + DP_{5\text{-}4} = 632 + 74 = 706$ mm w.g.
— $G_4 = G_8 \cdot (H_4 / H_8)^{0,525} = 330 \cdot (706 / 328)^{0,525} = 494$ l/h $v_4 = 0{,}66$ m/s

Pipework between floors 4 and 3

— Flow rate in the pipes: $G_{4\text{-}3} = G_{5\text{-}4} + G_4 = 1.557 + 494 = 2.051$ l/h   Chosen diameter ø = 1 1/4”
$r = 11{,}5$ mm w.g./m $v = 0{,}56$ m/s $\Sigma\xi = 2{,}0$ (no variation in diameter)

— Pressure loss: $\Delta P_{4\text{-}3} = 11{,}5 \cdot 6 + 31 = 100$ mm w.g.

Secondary circuit of fan coil unit - floor 3

— $H_3 = H_4 + \Delta P_{4\text{-}3} = 706 + 100 = 806$ mm w.g.

— $G_3 = G_8 \cdot (H_3 / H_8)^{0,525} = 330 \cdot (806 / 328)^{0,525} = 529$ l/h $v_3 = 0{,}71$ m/s

Pipework between floors 3 and 2

— Flow rate in the pipes: $G_{3\text{-}2} = G_{4\text{-}3} + G_3 = 2.051 + 529 = 2.580$ l/h   Chosen diameter ø = 1 1/2”
$r = 8{,}5$ mm w.g./m $v = 0{,}52$ m/s $\Sigma\xi = 3{,}5$ (variation in diameter)

— Pressure loss: $\Delta P_{3\text{-}2} = 8{,}5 \cdot 6 + 47 = 98$ mm w.g.

Secondary circuit of fan coil unit - floor 2

— $H_2 = H_3 + DP_{3\text{-}2} = 806 + 98 = 904$ mm w.g.

— $G_2 = G_8 \cdot (H_2 / H_8)^{0,525} = 330 \cdot (904 / 328)^{0,525} = 562$ l/h $v_2 = 0{,}75$ m/s

Pipework between floors 2 and 1

— Flow rate in the pipes: $G_{2\text{-}1} = G_{3\text{-}2} + G_2 = 2.580 + 562 = 3.142$ l/h   Chosen diameter ø = 1 1/2”
$r = 12{,}0$ mm w.g./m $v = 0{,}64$ m/s $\Sigma\xi = 2{,}0$ (no variation in diameter)

— Pressure loss: $\Delta P_{2\text{-}1} = 12{,}0 \cdot 6 + 41 = 113$ mm w.g.

Secondary circuit of fan coil unit - floor 1

— $H_1 = H_2 + \Delta P_{2\text{-}1} = 904 + 113 = 1.017$ mm w.g.

— $G_1 = G_8 \cdot (H_1 / H_8)^{0,525} = 330 \cdot (1.017 / 328)^{0,525} = 598$ l/h $v_1 = 0{,}80$ m/s

Pipework between the first floor and the pipework connections

— Flow rate in the pipes: $G_{1\text{-}0} = G_{2\text{-}1} + G_1 = 3.142 + 598 = 3.740$ l/h   Chosen diameter ø = 2”
$r = 5{,}0$ mm w.g./m $v = 0{,}47$ m/s $\Sigma\xi = 3{,}5$ (variation in diameter)

— Pressure loss: $\Delta P_{1\text{-}0} = 5{,}0 \cdot 8 + 38 = 78$ mm w.g.

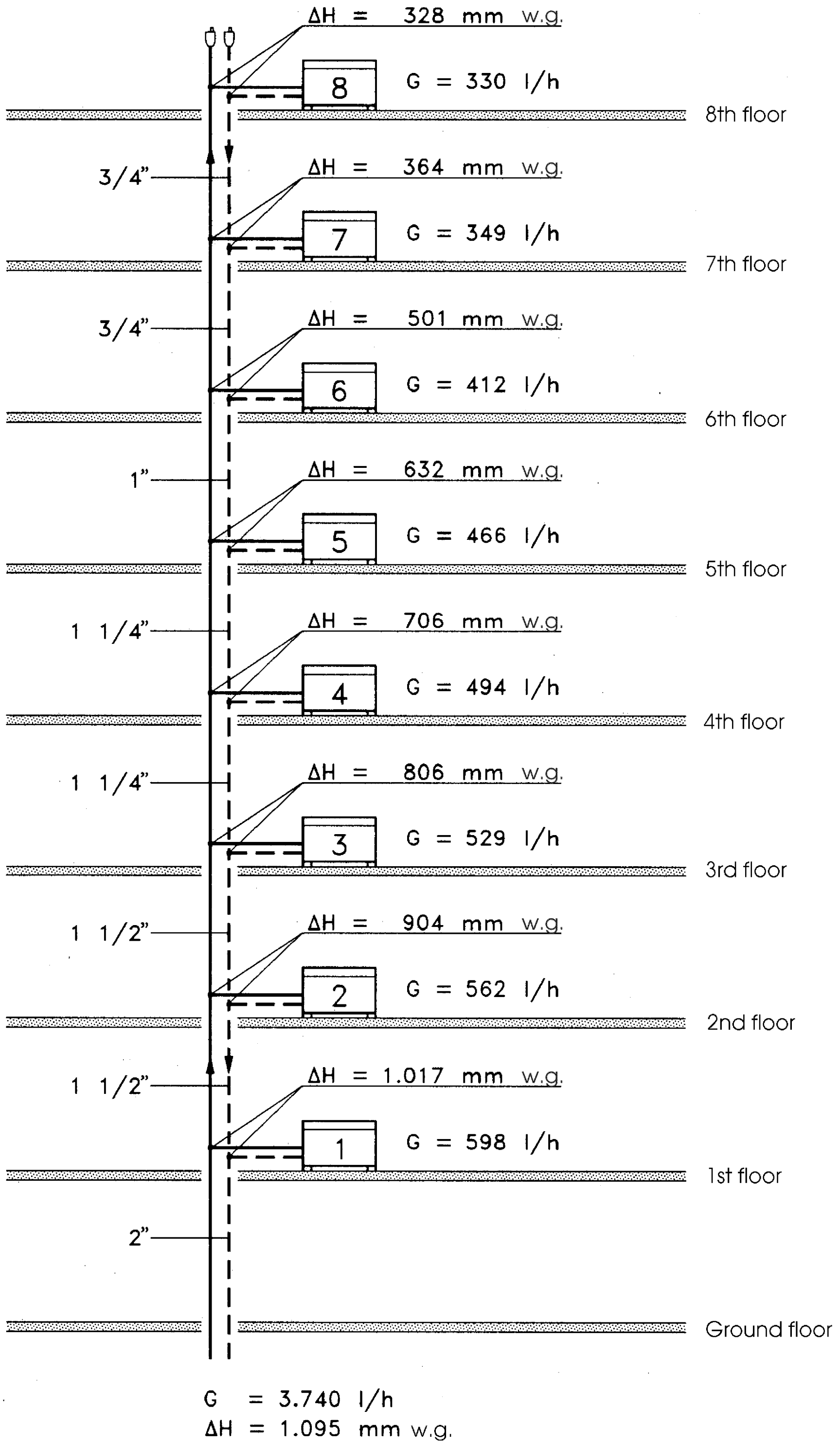
Flow rate and head at the pipework connections

— $H = 1.017 + 78 = 1.095$ mm w.g.

— $G = 3.740$ l/h

<u>**Note:**</u>

The emitters on the lower floors are supplied at excessive flow rates.
See chapter entitled “CHARACTERISTICS OF SIMPLE CIRCUITS”.


ΔH = 328 mm w.g.
8
G = 330 l/h
8th floor
3/4"
ΔH = 364 mm w.g.
7
G = 349 l/h
7th floor
3/4"
ΔH = 501 mm w.g.
6
G = 412 l/h
6th floor
1"
ΔH = 632 mm w.g.
5
G = 466 l/h
5th floor
1 1/4"
ΔH = 706 mm w.g.
4
G = 494 l/h
4th floor
1 1/4"
ΔH = 806 mm w.g.
3
G = 529 l/h
3rd floor
1 1/2"
ΔH = 904 mm w.g.
2
G = 562 l/h
2nd floor
1 1/2"
ΔH = 1.017 mm w.g.
1
G = 598 l/h
1st floor
2"
Ground floor
G = 3.740 l/h
ΔH = 1.095 mm w.g.

PRACTICAL CALCULATION OF SIMPLE CIRCUITS

There are no completely reliable general 'rule of thumb' methods for the practical design of simple circuits are there are too many variables to be taken into account: Only experience can provide knowledge of short cuts which can be used to reduce the complexities of the calculations.

A practical method of assessing the dimensions of simple circuit pipework in bildings with floor inequalities of between 2.7 and 3.3 m is shown below. The main stages of this assessment are as follows.

1. **The dimensions of the final secondary circuit are calculated, based on the required flow rate,** by determining:
   - **the diameter** and
   - **pressure loss.**

2. **The pressure losses across the other secondary circuits are calculated** using the following criteria:
   – **in the case of the penultimate secondary circuit** assess the value as follows:
     - increase the pressure loss across the final circuit by 100 mm w.g. and
     - round off this value to the nearest multiple of 100.

   – **in the case of the other secondary circuits** the value is obtained by increasing the pressure loss across the upper floor secondary circuit by 100 mm w.g.

3. **The dimensions of the other secondary circuits are calculated, on the basis of the required capacity,** by determining:
   - **the diameter** and
   - **pressure loss.**

   The flow rate and pressure losses calculated **in this way must then be balanced with the head determined in 2.**

4. **The dimensions of the pipework in the main circuit are calculated** based on:
   - **their flow rate** (this is obtained by adding the flow rates of the secondary circits served by the pipework being dimensioned) and
   - **by considering continuous pressure losses equal to 10 mm w.g./m.**

Note:

If the secondary circuits and emitters are equal the flow rate of a particular circuit can be derived from that of the upper floor by means of the relative conversion factor given in TAB. 1.

TAB. 1 - CONVERSION FACTORS FOR FLOW RATE AGAINST VARIATION IN HEAD

H_1	H_2	F	H_1	H_2	F	H_1	H_2	F
100	200	1,44	1.100	1.200	1,05	2.100	2.200	1,02
200	300	1,24	1.200	1.300	1,04	2.200	2.300	1,02
300	400	1,16	1.300	1.400	1,04	2.300	2.400	1,02
400	500	1,12	1.400	1.500	1,04	2.400	2.500	1,02
500	600	1,10	1.500	1.600	1,03	2.500	2.600	1,02
600	700	1,08	1.600	1.700	1,03	2.600	2.700	1,02
700	800	1,07	1.700	1.800	1,03	2.700	2.800	1,02
800	900	1,06	1.800	1.900	1,03	2.800	2.900	1,02
900	1.000	1,06	1.900	2.000	1,03	2.900	3.000	1,02
1.000	1.100	1,05	2.000	2.100	1,03	3.000	3.100	1,02

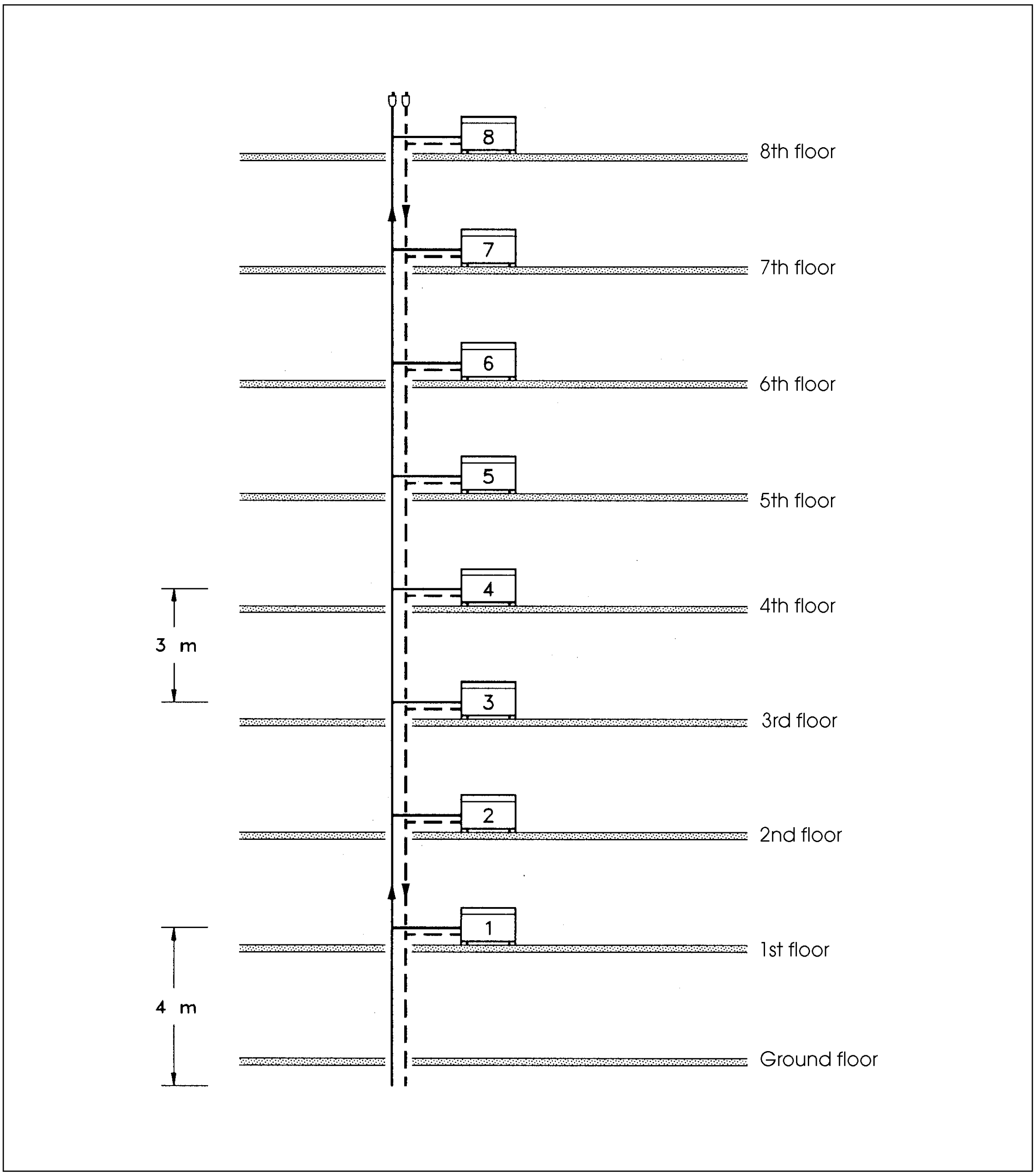
F is the conversion facor used to determine the new flow rate of a circuit which changes from a pressure H_1 (mm w.g.) **to a pressure H_2** (mm w.g.).

The values in TAB. 1 are derived from the formulae given in: BALANCING FLOW-RATE (1st Manual).

Example 2 - Practical calculation of a simple circuit

Using the method of calculation shown on the previous page determine the dimension of the simple circuit illustrated below. The following factors are taken into account:

- G = 330 l/h nominal flow rate of each fan coil unit,
- ΔP = 150 mm w.g. pressure loss corresponding to the nominal flow rate,
- l = 4 m lenght of the riser - fan coil unit connections,
- n = 2 (a 90°) bends in the riser - fan coil unit connections.



<u>Solution:</u>

— **Reference tables** (collected at the end of this book):
TABLE 2 - Localised pressure losses (1st Manual, LOCALISED PRESSURE LOSSES section)
TABLE 5 - Pressure loss in steel tubes (1st Manual STEEL TUBES section)

1 - Secondary circuit of fan coil unit 8

The dimensions of this circuit can be determined in the same way as the corresponding circuit of Example 1.

Therefore: $ø_8$ = 1/2”

G_8 = 330 l/h

H_8 = 328 mm w.g.

2 - Pressure losses in the secondary circuits

— Penultimate circuit: This is determined:
- by increasing the pressure of the final circuit by 100 mm w.g.: H = 328 + 100 = 428 mm w.g.
- by rounding off this value to the nearest multiple of 100.

This gives therefore: H_7 = 400 mm w.g.

— Other circuits: The pressure loss in these circuits is obtained by increasing that of the upper floor by 100 mm w.g.

This gives therefore: H_6 = 500 mm w.g.

H_5 = 600 mm w.g.

H_4 = 700 mm w.g.

H_3 = 800 mm w.g.

H_2 = 900 mm w.g.

H_1 = 1.000 mm w.g.

3 - Secondary circuit of fan coil unit 7

With the same flow rate (nominal flow rate) and the same diameter (3/8” would give an excessive velocity), this circuit is equal to that of fan coil unit 8. Consequently the effective flow rate of the circuit being considered can be determined by balancing G_8 and H_8 at the effective head (H_7) available at its connections (See 1st Manual - BALANCING FLOW RATE).

The following is therefore obtained: $G_7 = G_8 \cdot (H_7 / H_8)^{0,525} = 330 \cdot (400 / 328)^{0,525} = 366$ l/h

4 - Calculating the dimensions of the other secondary circuits

Given that these circuits and the corresponsing emitters are similar, che flow rate of the secondary circuit of each floor can be calculated using the balancing factors in TAB. 1.

- secondary circuit 6: H_6 = 500 mm w.g. $G_6 = G_7 \cdot F = 366 \cdot 1{,}12 = 410$ l/h
- secondary circuit 5: H_5 = 600 mm w.g. $G_5 = G_6 \cdot F = 410 \cdot 1{,}10 = 451$ l/h
- secondary circuit 4: H_4 = 700 mm w.g. $G_4 = G_5 \cdot F = 451 \cdot 1{,}08 = 487$ l/h
- secondary circuit 3: H_3 = 800 mm w.g. $G_3 = G_4 \cdot F = 487 \cdot 1{,}07 = 521$ l/h
- secondary circuit 2: H_2 = 900 mm w.g. $G_2 = G_3 \cdot F = 521 \cdot 1{,}06 = 552$ l/h
- secondary circuit 1: H_1 = 1.000 mm w.g. $G_1 = G_2 \cdot F = 552 \cdot 1{,}06 = 585$ l/h

5 - Determining the dimensions of the pipework

The dimensions of the pipework are calculated using r = 10 mm w.g./m. i.e.

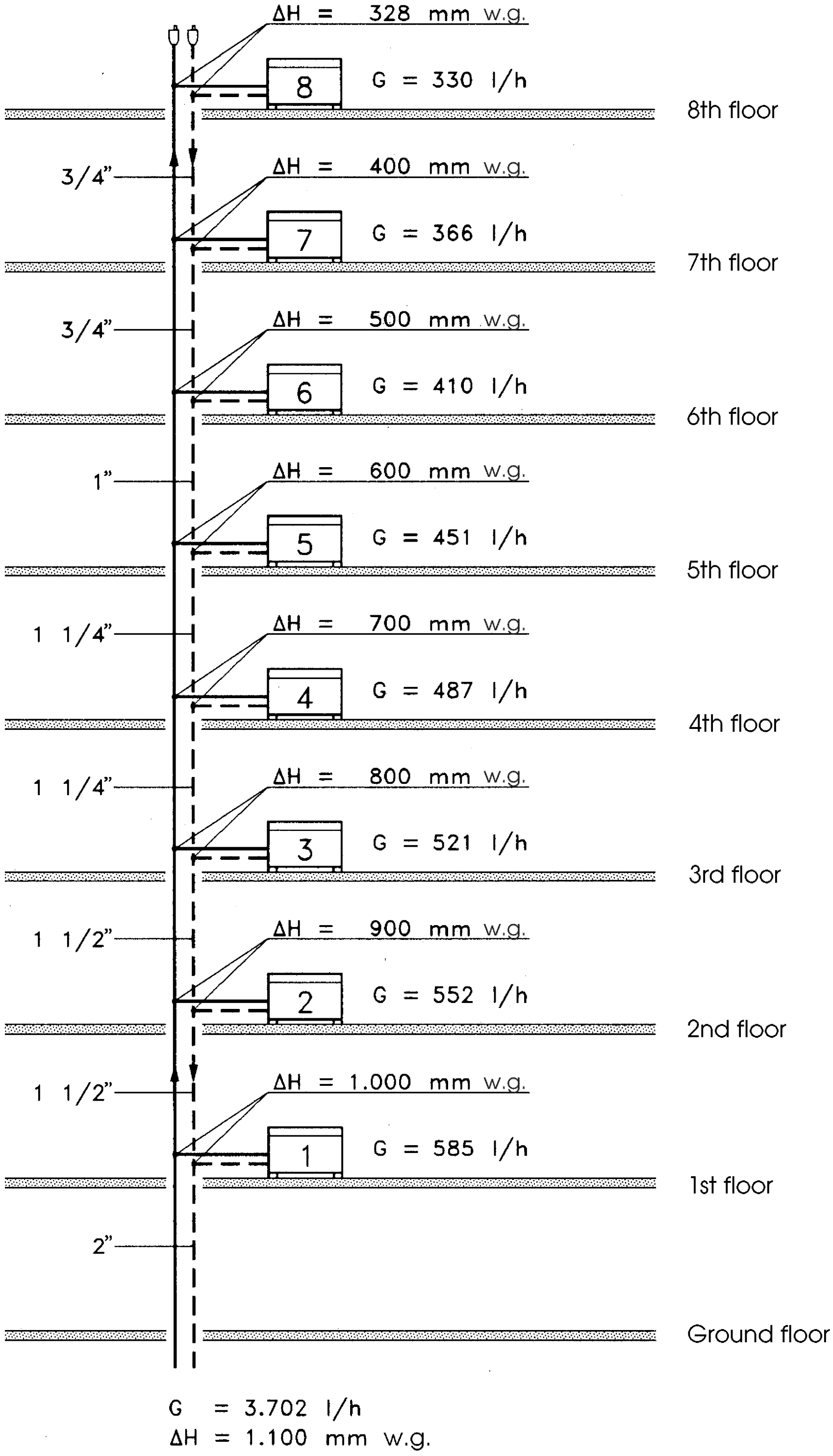
Therefore:

- pipework 8-7: $G_{8\text{-}7}$ = 330 l/h ø= 3/4”
- pipework 7-6: $G_{7\text{-}6} = G_{8\text{-}7} + G_7$ = 330 + 366 = 696 l/h ø = 3/4”
- pipework 6-5: $G_{6\text{-}5} = G_{7\text{-}6} + G_6$ = 696 + 410 = 1.106 l/h ø = 1”
- pipework 5-4: $G_{5\text{-}4} = G_{6\text{-}5} + G_5$ = 1.106 + 451 = 1.557 l/h ø = 1 1/4”
- pipework 4-3: $G_{4\text{-}3} = G_{5\text{-}4} + G_4$ = 1.557 + 487 = 2.044 l/h ø = 1 1/4”
- pipework 3-2: $G_{3\text{-}2} = G_{4\text{-}3} + G_3$ = 2.044 + 521 = 2.565 l/h ø = 1 1/2”
- pipework 2-1: $G_{2\text{-}1} = G_{3\text{-}2} + G_2$ = 2.565 + 552 = 3.117 l/h ø = 1 1/2”
- pipework 1-0: $G_{1\text{-}0} = G_{2\text{-}1} + G_1$ = 3.117 + 585 = 3.702 l/h ø = 2”

6 - Flow rate and head at the pipe connections

— H = 1.000 + 100 = 1.100 mm w.g.

— G = 3.702 l/h


ΔH = 328 mm w.g.
8
G = 330 l/h
8th floor
3/4"
ΔH = 400 mm w.g.
7
G = 366 l/h
7th floor
3/4"
ΔH = 500 mm w.g.
6
G = 410 l/h
6th floor
1"
ΔH = 600 mm w.g.
5
G = 451 l/h
5th floor
1 1/4"
ΔH = 700 mm w.g.
4
G = 487 l/h
4th floor
1 1/4"
ΔH = 800 mm w.g.
3
G = 521 l/h
3rd floor
1 1/2"
ΔH = 900 mm w.g.
2
G = 552 l/h
2nd floor
1 1/2"
ΔH = 1.000 mm w.g.
1
G = 585 l/h
1st floor
2"
Ground floor
G = 3.702 l/h
ΔH = 1.100 mm w.g.

Note:

With the following summary table the data obtained using the basic calculations and the data determined using the less complex method can be compared.

Secondary	G Theoretical calculation	G Practical calculation	ΔG	ΔG as %
8	330 l/h	330 l/h	0 l/h	0,0 %
7	349 l/h	366 l/h	+ 17 l/h	+ 4,9 %
6	412 l/h	410 l/h	- 2 l/h	- 0,5 %
5	466 l/h	451 l/h	- 15 l/h	- 3,2 %
4	494 l/h	487 l/h	- 7 l/h	- 1,4 %
3	529 l/h	521 l/h	- 8 l/h	- 1,5 %
2	562 l/h	552 l/h	- 10 l/h	- 1,8 %
1	598 l/h	585 l/h	- 13 l/h	- 2,2 %

Head at the pipe connections: H (Theoretical method) = 1.095 mm w.g.
H (Practical method) = 1.100 mm w.g.

From this comparison it will be seen that the differences between the two methods fall within the acceptable limits of design calculations for air conditioning systems (See 1st Manual under TOTAL HEAD LOSSES).

CHARACTERISTICS OF SIMPLE CIRCUITS

The previous examples show that **the circuit design will not allow the correct flow rate to pass through each branch.**

It will be necessary to increase the flow rates through the fan coil units on the lower floors in order to provide adequate circulation at the upper floors.

As has already been described in the BALANCING OF CIRCUITS section, flow rates which are greater than required can give rise to:

- **heat outputs which do not comply with the design values,**
- **emitters, operating outside design specification,**
- **excessive noise,**
- **wear and overheating of the circulating pumps,**

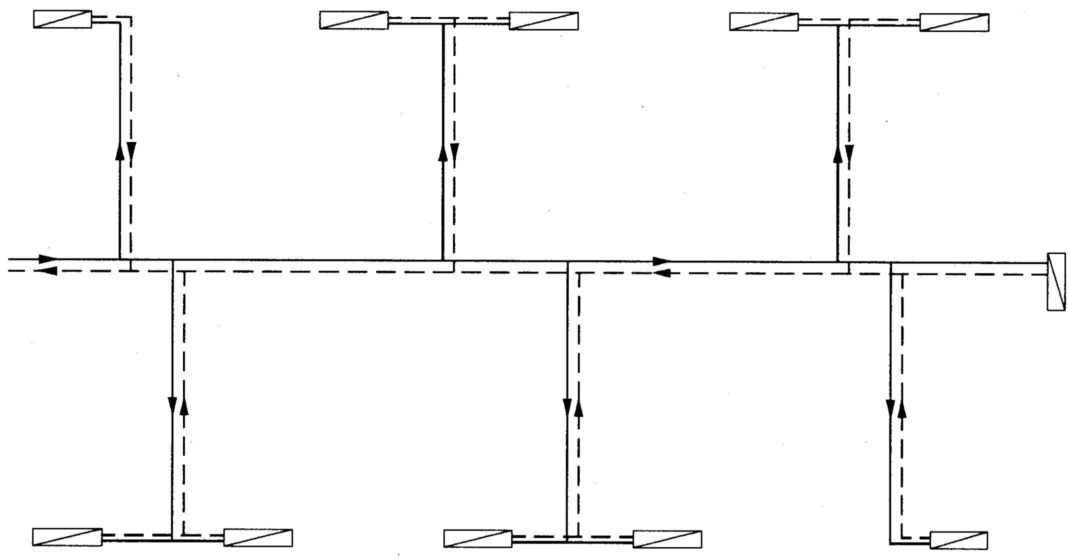
The degree of imbalance of the flow rates through circuits depends upon the number of branches served:

- **if the number of branches is small,** the differences between the required flow rates and those obtained are generally within acceptable margins;
- **if the number of branches is large,** the imbalances in the flow rates can be considerable. In such cases it is recommended to use reverse return circuits or circuits with balancing valves or autoflow units.

APPLICATIONS OF SIMPLE CIRCUITS

Small and medium sized installations use mainly simple systems

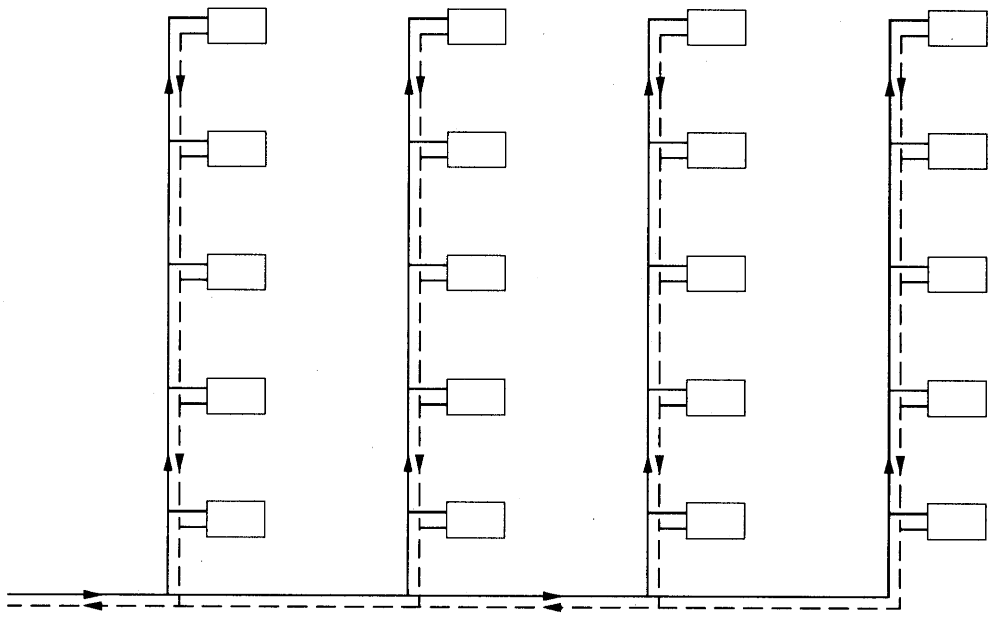
Circuits arranged horizontally



Circuits arranged horizontally

This type of circuit is used to connect a small number of emitters located on the same floor. In the case of larger distribution systems (for example in schools, hospitals or industrial buildings) it is advisable to use reverse return circuits or circuits with balancing valves and autoflow units.

Circuits arranged vertically

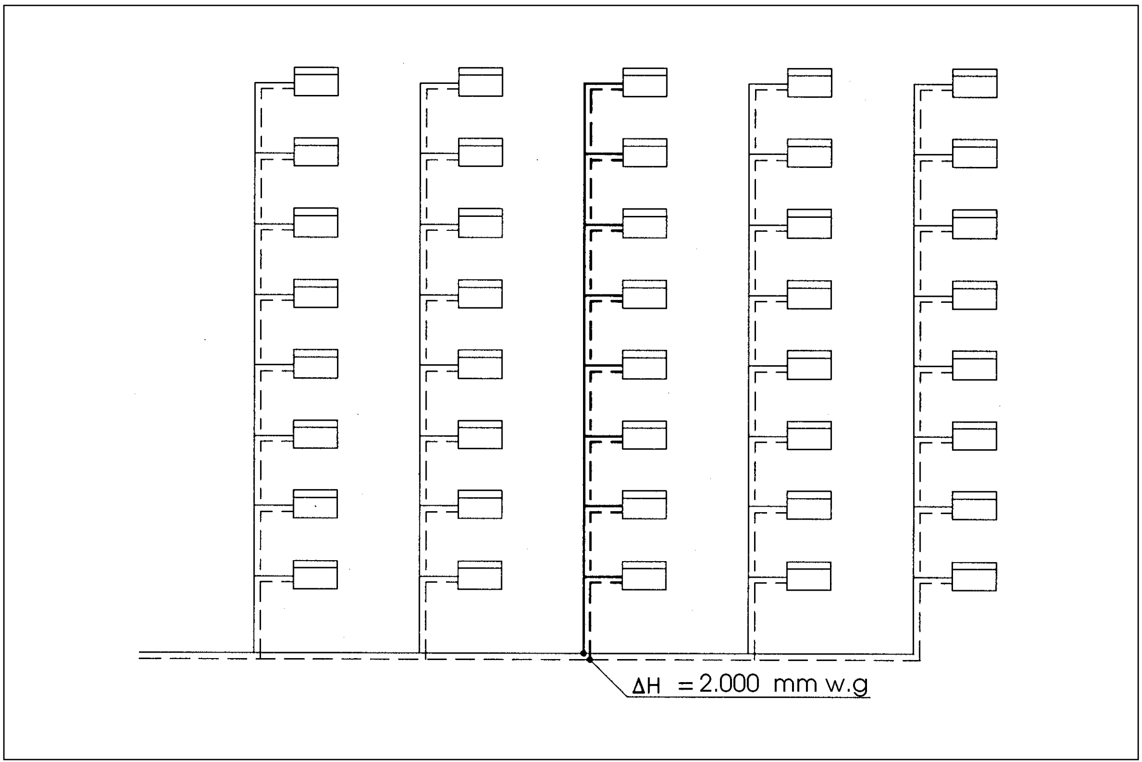


Circuits arranged vertically

This type of circuit is used to connect a small number of emitters located on different floors. On systems with extensive networks this type of circuit can result in severe imbalances in flow rate as the differential pressures of the circuits **increase not only along the vertical pipes but also along the base distribution pipes.**
As a result of this it may be important to re-calculate the flow rates in the pipework calculated in example 1 to take account of changes in differential pressure.

Example 3

Calculate the new rates at the fan coil units, using the pipework from Example 1, but take into account a differential pressure of 2000 mm w.g. at the base.



Solution:

The pipework whose dimensions were determined in Example 1 (with a differential pressure at the base of 1,095 mm w.g.) provide the following flow rates at the fan coil units:

- flow rate of fan coil unit 8: G_8 = 330 l/h
- flow rate of fan coil unit 7: G_7 = 349 l/h
- flow rate of fan coil unit 6: G_6 = 412 l/h
- flow rate of fan coil unit 5: G_5 = 466 l/h
- flow rate of fan coil unit 4: G_4 = 494 l/h
- flow rate of fan coil unit 3: G_3 = 529 l/h
- flow rate of fan coil unit 2: G_2 = 562 l/h
- flow rate of fan coil unit 1: G_1 = 598 l/h

- total flow rate of the columns: G = 3.740 l/h

The new flow rates (i.e. those corresponding to the new differential pressure) can be determined by multiplying the old values by the balancing factor F (see under BALANCING FLOW RATE in the 1st Manual).

This factor can be calculated using the following equation:

$$F = (H_1 / H)^{0,525}$$

where: H_1 = 2.000 mm w.g. new head at the foot of riser
H_1 = 1.095 mm w.g. old head at the foot of the riser
giving

$$F = (2.000 / 1.095)^{0,525} = 1,37$$

Knowing the value of F, the new flow rates in the pipework is calculated as follows:

- New flow rate of the fan coil unit 8: G_8 = 330 · 1,37 = 452 l/h
- New flow rate of the fan coil unit 7: G_7 = 349 · 1,37 = 478 l/h
- New flow rate of the fan coil unit 6: G_6 = 412 · 1,37 = 564 l/h
- New flow rate of the fan coil unit 5: G_5 = 466 · 1,37 = 638 l/h
- New flow rate of the fan coil unit 4: G_4 = 494 · 1,37 = 677 l/h
- New flow rate of the fan coil unit 3: G_3 = 529 · 1,37 = 725 l/h
- New flow rate of the fan coil unit 2: G_2 = 562 · 1,37 = 770 l/h
- New flow rate of the fan coil unit 1: G_1 = 598 · 1,37 = 819 l/h

- New total flow rate in the risers: G = 3.740 · 1,37 =5.123 l/h

<u>Note:</u>

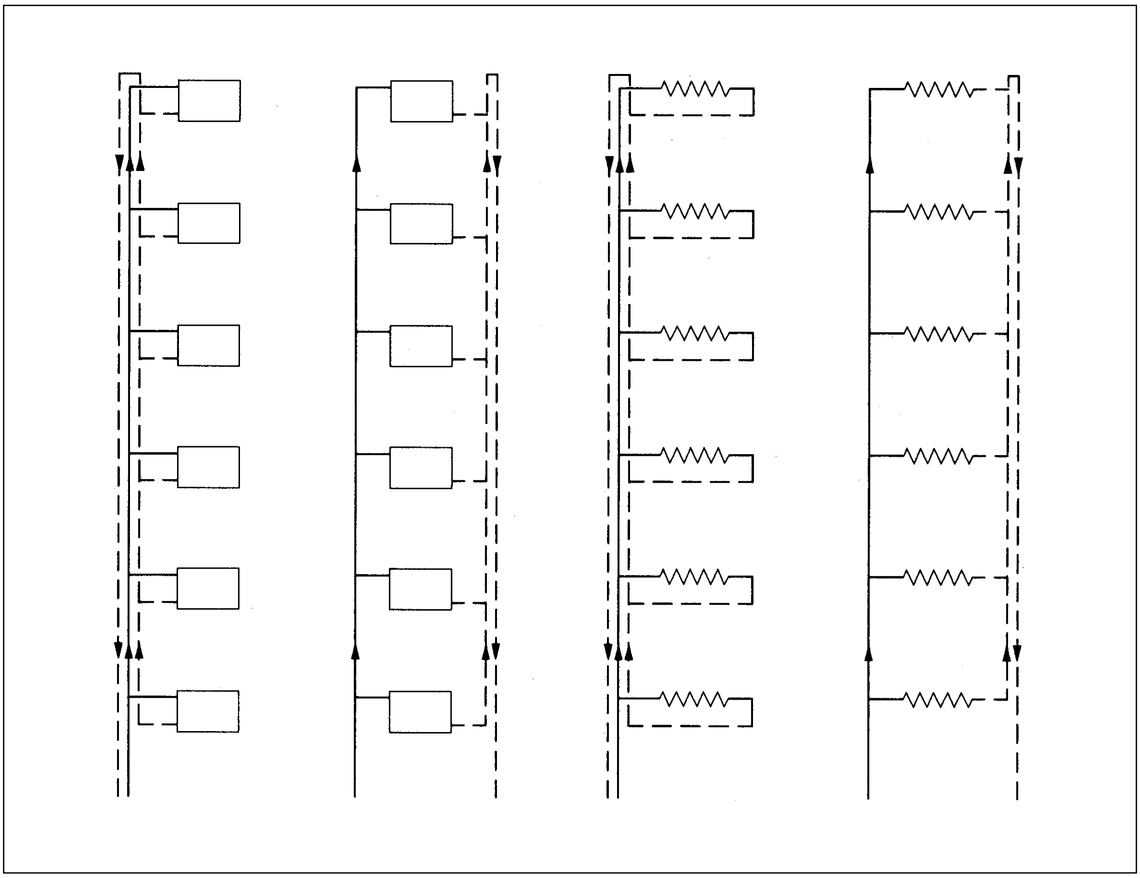
This example shows that the new differential pressure assumed at the base of the pipework provides a significant increase in the flow rates across the fan coil units. Consequently the difference between the required nominal flow rates and those which can be affectively supplied is increased to an even greater extent.

REVERSE RETURN CIRCUITS

CALCULATION OF REVERSE RETURN CIRCUITS

CHARACTERISTICS OF REVERSE RETURN CIRCUITS

APPLICATIONS OF REVERSE RETURN CIRCUITS



These are circuits wich make it possible to connect branches (risers, zones or emitters) with the same lenght of pipes. Examples are shown on the previous page.

CALCULATION OF REVERSE RETURN CIRCUITS

The dimensions of the pipes in these circuits are calculated with constant linear head losses so they are compatible with the commercial pipe sizes available. **It is possible to ensure differential pressures wich are almost equal to the branches served using the same lenghts of pipe:** all subsidiary units being connected in a "reverse return" manner. These circuits can be designed using relatively simple methods wich are both practical and reliable. **For example the following procedure can be adopted:**

1. **A reference secondary circuit is developed** (normally the last of the outgoing section or that wich requires the greatest pressure) **and its dimensions are worked out, based on the required flow rate,** by determining:
   - **the diameter,**
   - **pressure loss.**

2. **The dimensions of the other secondary circuits are calculated, based on the required flow rate,** by determining:
   - **the diameter,**
   - **pressure loss.**

   The flow rate and pressure losses determined **in this way must then be balanced against the available head at the connections of the secondary reference circuit.**

3. **The dimensions of the outgoing sections of the main circuit are determined** on the basis:
   - **of their flow rate** (this is obtained by adding the flow rates of the secondary circuits served by the sections under examination) and
   - **with constant linear pressure losses** (e.g : r = 10 mm w.g./m).

4. **The dimensions of the return sections of the main circuit are calculated using the same criteria as in 3.**

5. **The total pressure losses of the circuit are determined by adding together:**

   a) **the pressure loss of the secondary reference circuit;**

   b) **the continuous pressure losses (h) of the main circuit** calculated in the conventional manner by multiplying togheter the following values:

   r = assumed value for the linear head losses (See 3)
   l = lenght of the pipes serving the reference circuit;

   c) **the localised pressure losses (z) of the main circuit** wich are normally considered equal to a percentage of the continuous pressure losses (h).

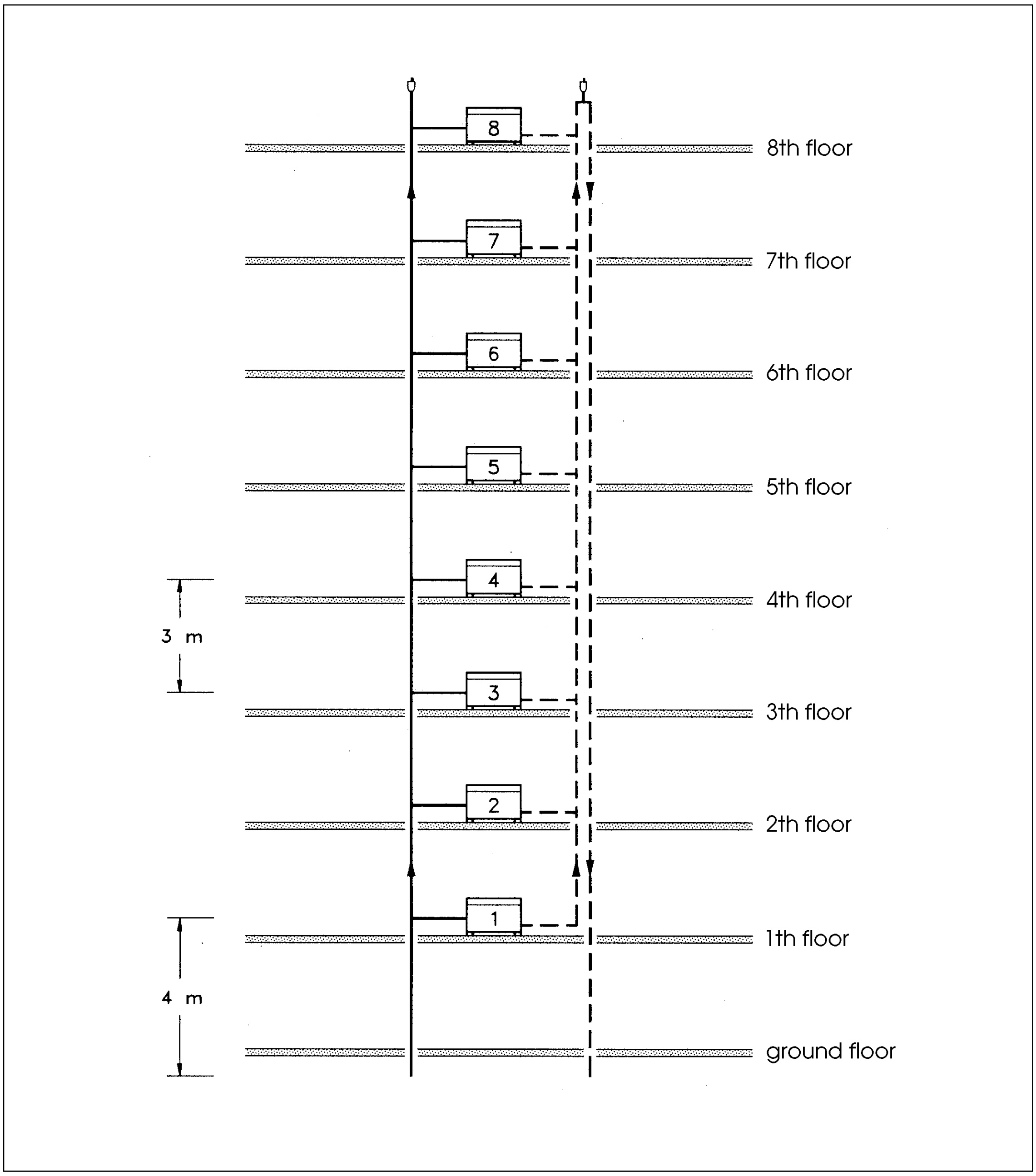
   The following are normally taken into consideration:

   $\mathbf{z = 0{,}6 \cdot h}$ for circuits with few bends,
   $\mathbf{z = 0{,}7 \cdot h}$ for circuits with several bends.

Example 1 - Calculation of a reverse return circuit

Using the method of calculation described earlier, determine the dimensions of the reverse return circuit illustrated in the diagram below. The following values are used:

- G = 330 l/h nominal flow rate of each fan coil unit,
- ΔP = 150 mm w.g. pressure losses corresponding to the nominal flow rate,
- l = 4 m length of the riser - fan coil unit connections,
- n = 2 (a 90°) bends in riser - fan coil unit connection.



Solution:

1 - Calculating the reference secondary circuit

The circuit of fan coil unit 8, the dimensions of wich can be calculated in the same way as the corresponding circuit in Example 1 "SINGLE CIRCUITS", is considered as the reference secondary circuit.

The following is obtained : $\varnothing_8$ = 1/2", G_8 = 330 l/h, H_8 = 328 mm w.g.

2 - Calculating the dimensions of other secondary circuits

With equal flow rate (nominal flow rate) and equal diameter all the secondary circuits of the intermediate floors are equal to the reference circuit and consequently also have the same pressure losses. There are not therefore any balancing operations required (See No. 2 of the method of calculation).

3 - Calculating the dimensions of the return sections of the main circuit

Calculating these sizes based on: r = 10 mm w.g./m, the following is obtained:

- outgoing section 8-7: G_{8-7} = 330 l/h ø = 3/4"
- outgoing section 7-6: $G_{7-6} = G_{8-7} + G_7$ = 330 + 330 = 660 l/h ø = 3/4"
- outgoing section 6-5: $G_{6-5} = G_{7-6} + G_6$ = 660 + 330 = 990 l/h ø = 1"
- outgoing section 5-4: $G_{5-4} = G_{6-5} + G_5$ = 990 + 330 = 1.320 l/h ø = 1 1/4"
- outgoing section 4-3: $G_{4-3} = G_{5-4} + G_4$ = 1.320 + 330 = 1.650 l/h ø = 1 1/4"
- outgoing section 3-2: $G_{3-2} = G_{4-3} + G_3$ = 1.650 + 330 = 1.980 l/h ø = 1 1/4"
- outgoing section 2-1: $G_{2-1} = G_{3-2} + G_2$ = 1.980 + 330 = 2.310 l/h ø = 1 1/4"
- outgoing section 1-0: $G_{1-0} = G_{2-1} + G_1$ = 2.310 + 330 = 2.640 l/h ø = 1 1/2"

4 - Calculating the dimensions of the return sections of the main circuit

As the flow rates of secondary circuits are equal to the each other these sections are symmetrical with the outgoing ones, i e:

- return section 1-2: G_{1-2} (return) = G_{8-7} (outgoing) = 330 l/h ø = 3/4"
- return section 2-3: G_{2-3} (return) = G_{7-6} (outgoing) = 660 l/h ø = 3/4"
- ....
- return section 8-0: G_{8-0} (return) = G_{1-0} (outgoing) = 2.640 l/h ø = 1 1/2"

5 - Total pressure losses of the circuit

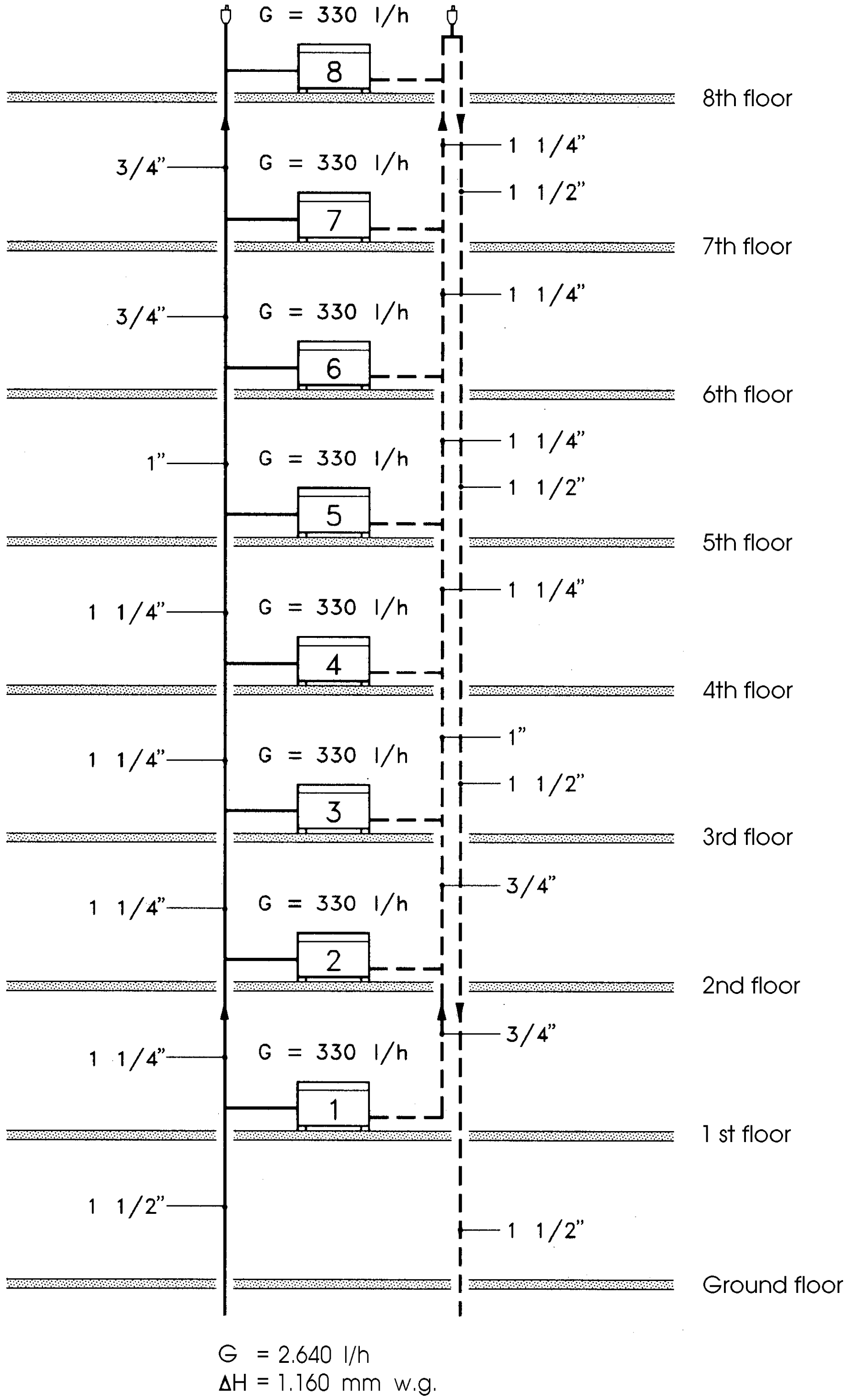
The total pressure losses of the circuit are calculated by adding together the following:

a) The total pressure losses of the reference secondary circuit: H_8 = 328 mm w.g.

b) The continuous pressure losses of the main circuit (See calculation methods)
With: r = 10 mm w.g/m
l = (7 · 3 + 4) · 2 + 2 = 52 m (the reverse return section is considered over 2 m)
therefore: h = r · l = 10 · 52 = 520 mm w.g.

c) The localised pressure losses of the main circuit (See calculation methods)
z = 0,6 · 520 = 312 mm w.g. (the circuit has few bends)

Therefore: H = H_8 + h + z = 328 + 520 + 312 = 1.160 mm w.g.



G = 330 l/h
8
8th floor
1 1/4"
3/4"
G = 330 l/h
1 1/2"
7
7th floor
1 1/4"
3/4"
G = 330 l/h
6
6th floor
1 1/4"
1"
G = 330 l/h
1 1/2"
5
5th floor
1 1/4"
1 1/4"
G = 330 l/h
4
4th floor
1"
1 1/4"
G = 330 l/h
1 1/2"
3
3rd floor
3/4"
1 1/4"
G = 330 l/h
2
2nd floor
3/4"
1 1/4"
G = 330 l/h
1
1 st floor
1 1/2"
1 1/2"
Ground floor
G = 2.640 l/h
ΔH = 1.160 mm w.g.

CHARACTERISTICS OF REVERSE RETURN CIRCUITS

As has already been pointed out **if their dimensions are determined with constant linear pressure losses these circuits are capable of ensuring almost equal differential pressures at the connections of their branches. This makes it possible to produce balanced arrangements with uniform sub-circuits:** i.e. with driveded circuits which require (as in the example) equal differential pressures or which do not vary very much.

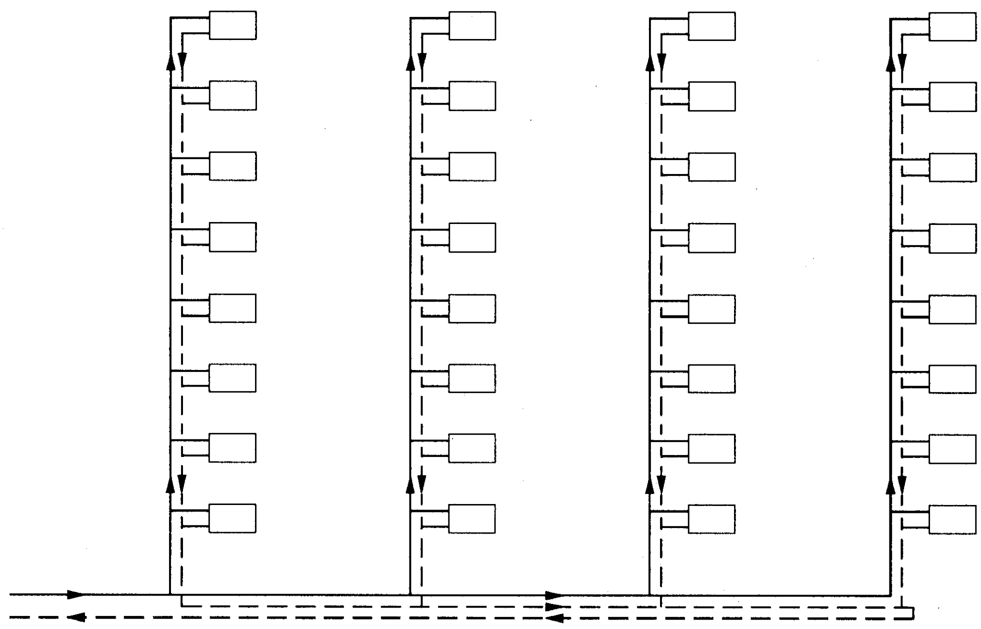
On the other hand balanced distribution cannot be provided if the sub-circuits are not uniform. For example it is not possible to obtain a balanced distribution if the reverse circuit has to serve heat emitter circuits which have substantially different differential pressures. **The term "balanced" which is sometimes used to characterise these circuits is not therefore correct.**

It should also be borne in mind that the reverse return circuits are much heavier and take up much more space than those with two pipes.

APPLICTIONS OF REVERSE RETOURN CIRCUITS

In heating systems the reverse return circuits are used to provide the same differential pressure at the connections of risers and at the zone subsidiary units or heat emitters.

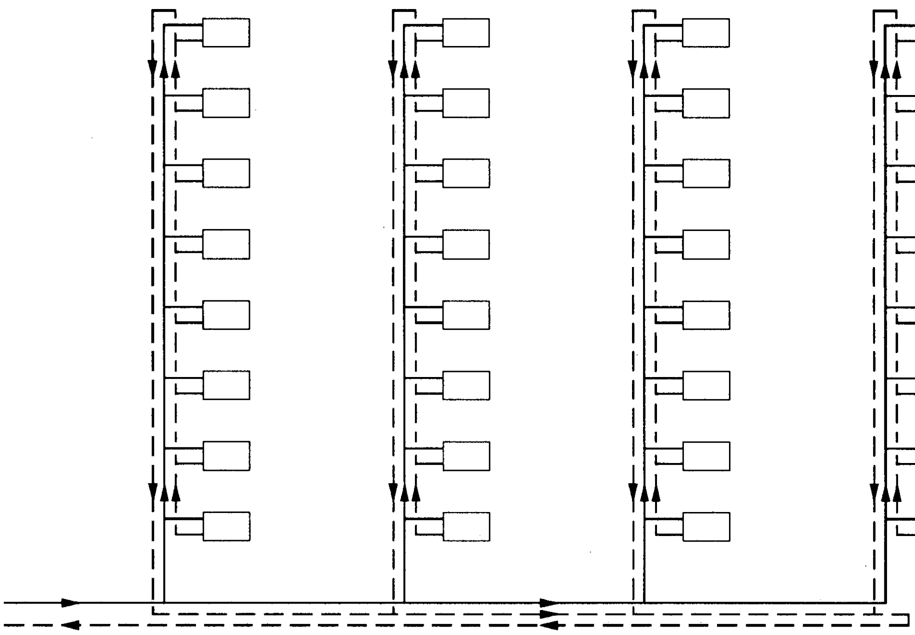
Circuit with risers served by a reverse return base circuit



Circuits with risers by a reverse return base circuit

They are used to ensure equal differential pressures at the connections of the vertical pipework. However, calculations may show some imbalance at higer levels for the reasons given in the section relating to simple circuits. These circuits are therefore normally used in buildings which do not have more than 5 or 6 floors.

Circuits with heat emitters served by reverse return base and riser circuits



Circuits with heat emitters served by reverse return base and riser circuits

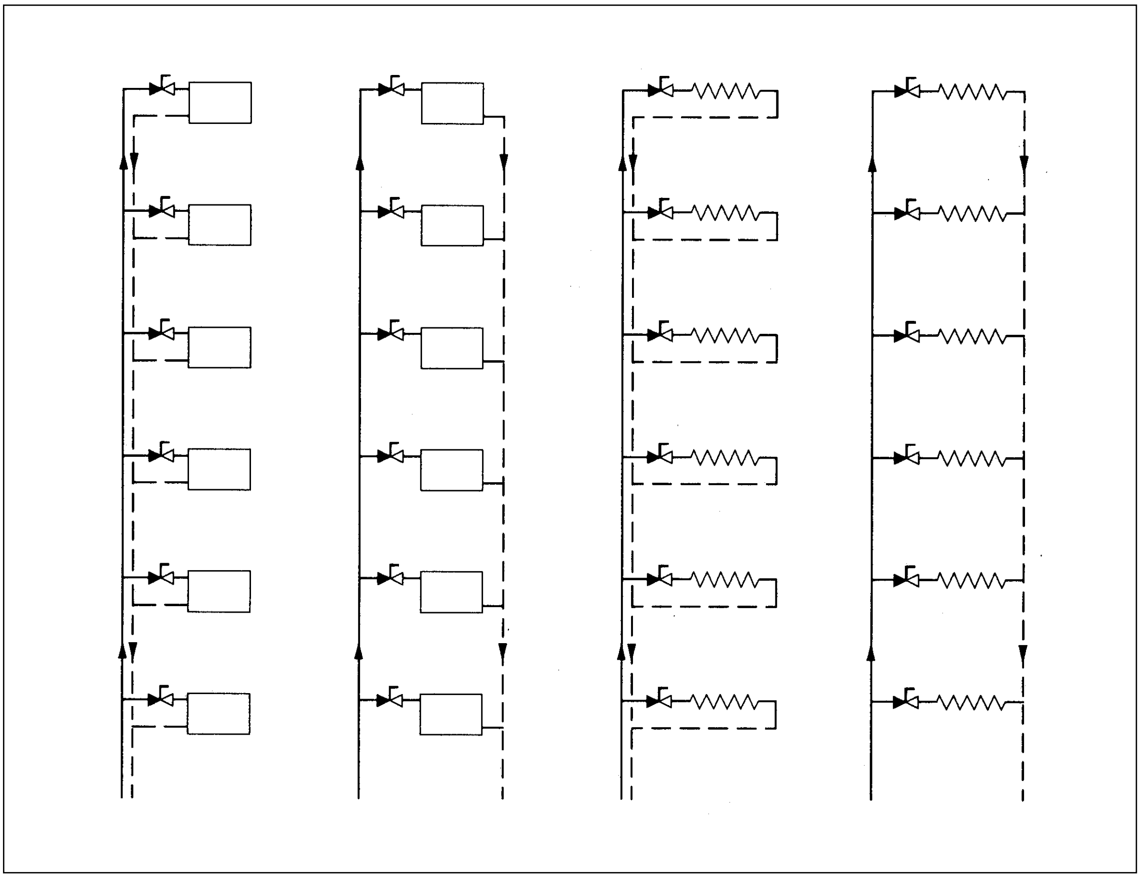
They are capable of ensuring equal differential pressures at each emitter in the circuit.

CIRCUITS WITH BALANCING VALVES

CALCULATION OF CIRCUITS WITH BALANCING VALVES

CHARACTERISTICS OF CIRCUITS WITH BALANCING VALVES

APPLICATIONS OF CIRCUITS WITH BALANCING VALVES



These are circuits with subsidiary units (risers, zones or emitters) **fitted with balancing valves:** i.e valves with wich the required pressure loss is achieved by adjustment of the fluid flow. These circuits are shown in the diagrams on the preceding page:

CALCULATION OF CIRCUITS WITH BALANCING VALVES

Generally speaking the dimensions of these circuits are calculated using a method which allows the correct calibration of the valves to be defined. **The procedure can be summarised as follows:**

1. **The dimensions of the final secondary circuit are calculated, based on the required flow rate,** by determining:
   - **the diameter** and
   - **pressure loss.**

2. **The dimensions of the final sections of the main circuit** (i.e. those between the final and penultimate secondary circuit) are calculate by determining:
   - **the flow rate** (equal to that of the final secondary circuit),
   - **the diameter** and
   - **pressure loss.**

3. **The dimensions of the penultimate secondary circuit are calculated based on the required flow rate** by determining:
   - **the diameter** and
   - **pressure loss** (on required head).

   If the required head is less than the available head this difference will be compensated by the corrisponding balancing valve.
   If, on the other hand, the required head exceeds the available head, this difference is compensated by the initial adjustment to the final secondary circuit and the new pressures distribution is then calculated.

4. **The dimensions of the penultimate sections of the main circuit are worked out** by determining:
   - **the flow rate** (This is obtained by adding the flow rates of the secondary circuits served by the sections under examination),
   - **the diameter,**
   - **pressure loss.**

5. **The dimensions of the other secondary circuits and the other sections of the main circuit:**

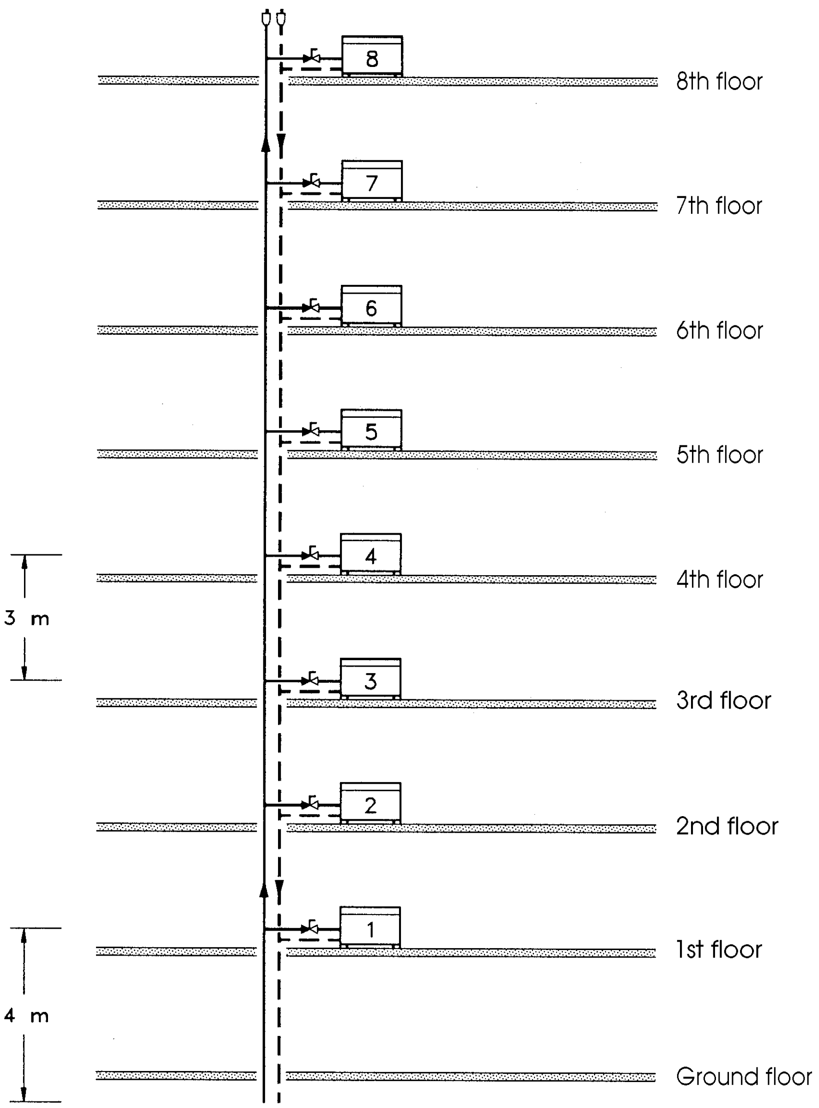
   – **In the case of the secondary circuits** the procedure as indicated in **Section** 3. is adopted.

   – **In the case of sections of the main circuit** the procedure as indicated in **Section** 4 is adopted.

Example 1 - Calculation of a circuit with balancing valves

Using the method of calculation given on the previous page determine the dimensions of the circuit with balancing valves as shown below. The following values are:

- G	=	330 l/h	nominal flow rate of each fan coil unit,
- ΔP	=	150 mm w.g.	pressure loss corresponding to the nominal flow rate,
- l	=	4 m	length of the riser - fan coil unit connections,
- n	=	2 (a 90°)	bends from the riser - fan coil unit connections.



<u>Solution:</u>

— Determining the ξ values (localised loss coefficients) relating to the riser - fan coil unit connection. (See Example 1 "SIMPLE CIRCUITS" section):

$\Sigma\xi$ = 10,0 (ø = 3/8", 1/2").

— Determining the ξ values (localised loss coefficients) relating to the riser sections between the floors (See Example 1 "SIMPLE CIRCUITS" section):

$\Sigma\xi$ = 2,0 (no variations in dia.),
$\Sigma\xi$ = 3,5 (ariations in dia.).

— To calculate the riser sections assume : r = 10 mm w.g./m (see "INTRODUCTORY NOTES").

— Reference tables :(collected at the end of this book)

TAB. 2 - Localised pressure losses (ved. 1st Manual LOCALISED PRESSURE LOSSES section)
TAB. 5 - Pressure losses in steel tubes (1st Manual STEEL TUBES section)

Secondary circuit of fan coil unit 8

The dimensions of this circuit can be calculated in the same way as the corresponding circuit in Example 1 of "SIMPLE CIRCUITS". The following is obtained:

— $ø_8$ = 1/2", G_8 = 330 l/h,

— H_8 = (328 + Zv) mm w.g., where Zv represents the pressure loss of the "open" balancing valve.

By assuming (for a flow rate of 330 l/h) Zv = 150 mm w.g. the following is obtained:

— H_8 = (328 + 150) = 478 mm w.g..

Riser section between floors 8 and 7

— Flow rate of the sections: $G_{8\text{-}7} = G_8$ = 330 l/h

—Chosen dia ø = 3/4" (The commercial diameter nearest the guide value: r = 10 mm w.g./m)

— Calculating of the pressure loss ($\Delta P_{8\text{-}7}$) of the riser sections:

• Distributed pressure losses. They are calculated using the formula: h = l · r.

Where:	l	= 6 m	(length of the riser sections)
	r	= 5 mm w.g./m	(TAB. 5, for dia. ø = 3/4" and G = 330 l/h)
giving:	h	= 30 mm w.g.	

• Localised pressure losses. They are calculated using TAB. 2.

Where:	$\Sigma\xi$	= 2,0	
	v	= 0,25 m/s	(TAB. 5, for dia. ø = 3/4" and G = 330 l/h)
giving:	z	= 6 mm w.g.	(TAB. 2)

Therefore: $\Delta P_{8\text{-}7}$ = h + z = 30 + 6 = 36 mm w.g.

Secondary circuit of the fan coil unit 7

Assuming dia. $\varnothing_7$ = 1/2", the secondary circuit 7 is equal to circuit 8 and therefore requires the same net differential pressure (H = 328 mm w.g.), i.e the same pressure net of the pressure loss induced by the balancing valve. The following statements may therefore be made:

— Head at the connections of the secondary circuit: $H_7 = H_8 + \Delta P_{8\text{-}7} = 478 + 36 = 514$ mm w.g.

— Pressure loss to be compensated for with calibration of the valve: $\Delta H_7 = 514 - 328 = 186$ mm w.g.

Riser sections between floors 7 and 6

— Flow rate of sections: $G_{7\text{-}6} = G_{8\text{-}7} + G_7 = 330 + 330 = 660$ l/h   chosen dia $\varnothing$ = 3/4"

$r = 18{,}0$ mm w.g./m $v = 0{,}50$ m/s $\Sigma\xi = 2{,}0$ (no dia. variations)

— Pressure loss: $\Delta P_{7\text{-}6} = 18{,}0 \cdot 6 + 25 = 133$ mm w.g.

Secondary circuit of the fan coil unit 6

— Head at the connections of the secondary circuit: $H_6 = H_7 + \Delta P_{7\text{-}6} = 514 + 133 = 647$ mm w.g.

— Pressure loss to be compensated for with calibration of the valve: $\Delta H_6 = 647 - 328 = 319$ mm w.g.

Riser sections between floors 6 and 5

— Flow rate of sections: $G_{6\text{-}5} = G_{7\text{-}6} + G_6 = 660 + 330 = 990$ l/h   chosen dia. $\varnothing$ = 1"

$r = 12{,}0$ mm w.g./m $v = 0{,}47$ m/s $\Sigma\xi = 3{,}5$ (dia. variations)

— Pressure loss: $\Delta P_{6\text{-}5} = 12{,}0 \cdot 6 + 38 = 110$ mm w.g.

Secondary circuit of the fan coil unit 5

— Head at the connections of the secondary circuit: $H_5 = H_6 + \Delta P_{6\text{-}5} = 647 + 110 = 757$ mm w.g.

— Pressure loss to be compensated for with calibration of the valve: $\Delta H_5 = 757 - 328 = 429$ mm w.g.

Riser sections between floors 5 and 4

— Flow rate of sections: $G_{5\text{-}4} = G_{6\text{-}5} + G_5 = 990 + 330 = 1.320$ l/h   chosen dia. $\varnothing$ = 1 1/4"

$r = 5{,}0$ mm w.g./m; $v = 0{,}36$ m/s $\Sigma\xi = 3{,}5$ (dia. variations)

— Pressure loss: $\Delta P_{5\text{-}4} = 5{,}0 \cdot 6 + 22 = 52$ mm w.g.

Secondary circuit of the fan coil unit 4

— Head at the connections of the secondary circuit: $H_4 = H_5 + \Delta P_{5\text{-}4} = 757 + 52 = 809$ mm w.g.

— Pressure loss to be compensated for with calibration of the valve: $\Delta H_4 = 809 - 328 = 481$ mm w.g.

Riser sections between floors 4 and 3

— Flow rate of sections: $G_{4\text{-}3} = G_{5\text{-}4} + G_4 = 1.320 + 330 = 1.650$ l/h   chosen dia. $\varnothing$ = 1 1/4"

$r = 7{,}5$ mm w.g./m $v = 0{,}45$ m/s $\Sigma\xi = 2{,}0$ (no dia. variations)

— Pressure loss: $\Delta P_{4\text{-}3} = 7{,}5 \cdot 6 + 20 = 65$ mm w.g.

Secondary circuit of fan coil unit 3

— Head at the connections of the secondary circuit: $H_3 = H_4 + \Delta P_{4\text{-}3} = 809 + 65 = 874$ mm w.g.

— Pressure loss to be compensated for with calibration of the valve: $\Delta H_3 = 874 - 328 = 546$ mm w.g.

Riser sections between floors 3 and 2

— Flow rate of sections: $G_{3\text{-}2} = G_{4\text{-}3} + G_3 = 1.650 + 330 = 1.980$ l/h chosen dia. ø = 1 1/4”
$r = 11{,}0$ mm w.g./m $v = 0{,}54$ m/s $\Sigma\xi = 2{,}0$ (no dia. variations)

— Pressure loss: $\Delta P_{3\text{-}2} = 11{,}0 \cdot 6 + 29 = 95$ mm w.g.

Secondary circuit of fan coil unit 2

— Head at the connections of the secondary circuit: $H_2 = H_3 + \Delta P_{3\text{-}2} = 874 + 95 = 969$ mm w.g.

— Pressure loss to be compensated for with calibration of the valve: $\Delta H_2 = 969 - 328 = 641$ mm w.g.

Riser sections between floors 2 and 1

— Flow rate of sections: $G_{2\text{-}1} = G_{3\text{-}2} + G_2 = 1.980 + 330 = 2.310$ l/h chosen dia. ø = 1 1/4”
$r = 14{,}5$ mm w.g./m $v = 0{,}63$ m/s $\Sigma\xi = 2{,}0$ (no dia variations)

— Pressure loss: $\Delta P_{2\text{-}1} = 14{,}5 \cdot 6 + 39 = 126$ mm w.g.

Secondary circuit of fan coil unit 1

— Head at the connections of the secondary circuit: $H_1 = H_2 + DP_{2\text{-}1} = 969 + 126 = 1.095$ mm w.g.

— Pressure loss to be compensated for with calibration of the valve: $DH_1 = 1.095 - 328 = 767$ mm w.g.

Riser sections between the first floor and the riser connections

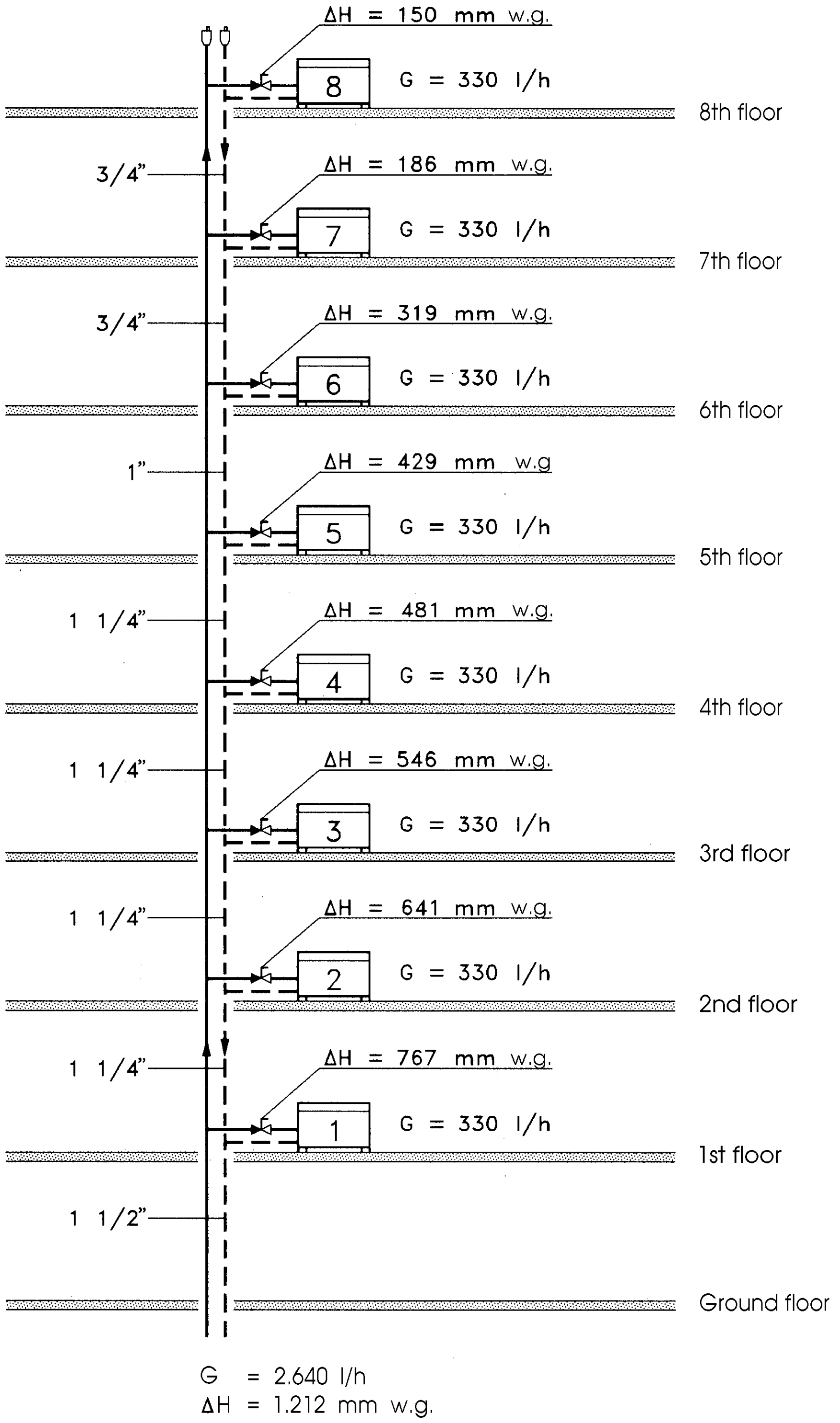
— Flow rate of sections: $G_{1\text{-}0} = G_{2\text{-}1} + G_1 = 2.310 + 330 = 2.640$ l/h chosen dia. ø = 1 1/2”
$r = 8{,}5$ mm w.g./m $v = 0{,}53$ m/s $\Sigma\xi = 3{,}5$ (dia. variations)

— Pressure loss: $\Delta P_{1\text{-}0} = 8{,}5 \cdot 8 + 49 = 117$ mm w.g.

Flow rate and head at the riser connections

— $H = 1.095 + 117 = 1.212$ mm w.g.

— $G = 2.640$ l/h


ΔH = 150 mm w.g.
G = 330 l/h
8
8th floor
3/4"
ΔH = 186 mm w.g.
G = 330 l/h
7
7th floor
3/4"
ΔH = 319 mm w.g.
G = 330 l/h
6
6th floor
1"
ΔH = 429 mm w.g
G = 330 l/h
5
5th floor
1 1/4"
ΔH = 481 mm w.g.
G = 330 l/h
4
4th floor
1 1/4"
ΔH = 546 mm w.g.
G = 330 l/h
3
3rd floor
1 1/4"
ΔH = 641 mm w.g.
G = 330 l/h
2
2nd floor
1 1/4"
ΔH = 767 mm w.g.
G = 330 l/h
1
1st floor
1 1/2"
Ground floor
G = 2.640 l/h
ΔH = 1.212 mm w.g.

CHARACTERISTICS OF CIRCUITS WITH BALANCING VALVES

The correct application of balancing valves will ensure adequate control of flow rates in subsidiary circuits and ensure even water distribution in the most complex systems.

The following are however considered essential requirements for the correct operation of these circuits:

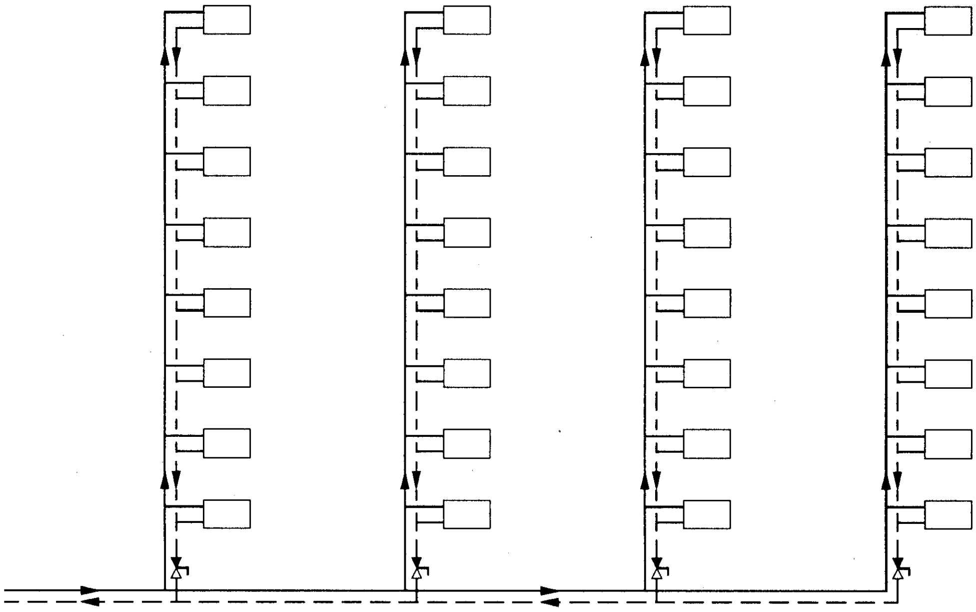
- **Thorough design work,** as the correct setting for each valve must be calculated;
- **A careful, and frequently laborious, performance of the balancing operations;**
- **An efficient maintenance service** which is capable of carrying out checks and, if necessary, new balancing operations.
  In practice these circuits can be put out of balance either by operations which have been incorrectly carried out or by operations designed to change the output of the emitters, particularly in cases where the thermal energy is measured indirectly.

It must also be borne in mind that when there are changes circuits must be recalculated and re-balanced. In fact only static type regulation i.e. with a fixed piston, can be carried out with balancing valves and regulation of this type cannot be automatically adapted to new operating conditions.

APPLICATIONS FOR WHICH CIRCUITS WITH BALANCING VALVES ARE USED

Circuits with balancing valves are used in air conditioning systems primarily to provide the desired differential pressures at the riser connections, at the zone subsidiary units or at the emitters.

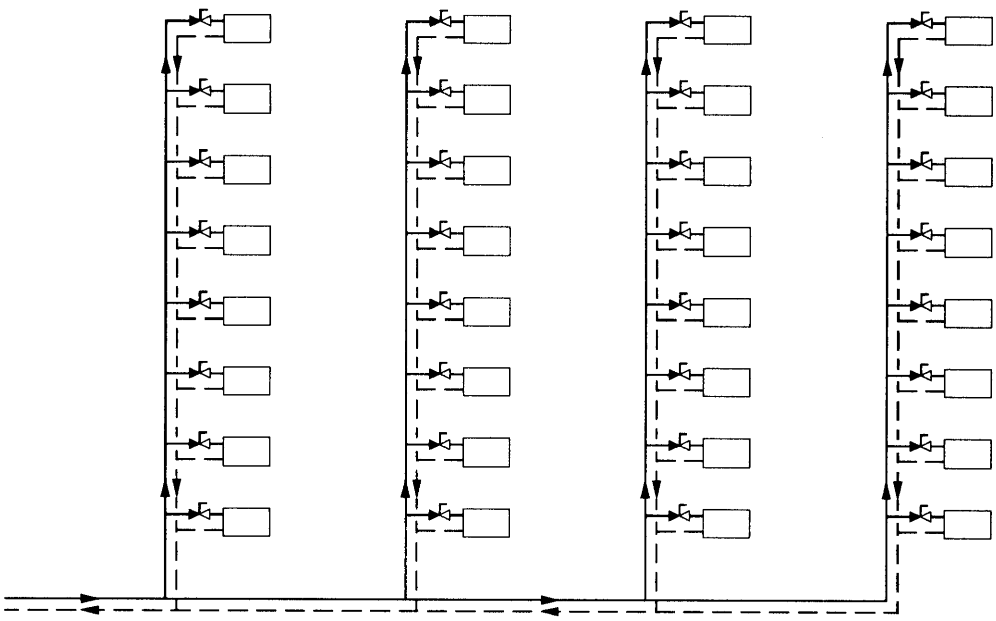
Circuit with balancing valve at the foot of the riser



Circuit with balancing valve at the foot of the riser

They provide desired differential pressures at the riser connections.
Imbalances in the flow rate distribution can occur along the risers for the reasons given in the section relating to simple circuits. For this reason these circuits are normally used in building which do not exceed 5 or 6 floors.

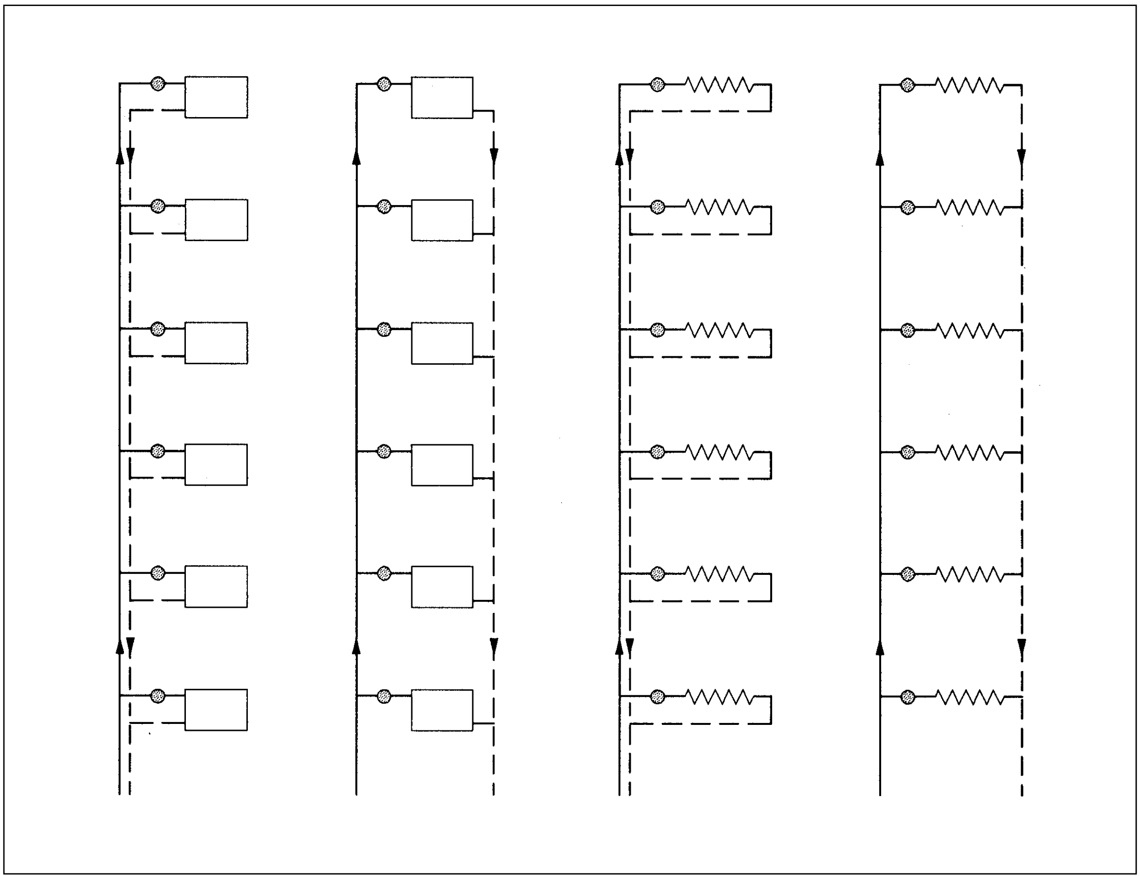
Circuits with balancing valves at each amitter



Circuits with balancing valves at each emitter

These circuits will provide the desired differential pressures at each emitter.

CIRCUITS WITH AUTOFLOW



These are circuits with subsidiary units (risers, zones or emitters) **equipped with an autoflow regulator,** i.e. an automatic device which maintains the flow rate passing through it at a constant level. Examples are shown on the previous page.

CALCULATION OF CIRCUITS WITH AUTOFLOW

The use of autoflow units means that simple, reliable and practical calculation methods can be used. For example the following procedure can be adopted:

1. **The autoflow device for each secondary circuit is chosen in relation to the required flow rate.**

2. **The dimensions of the pipework of the secondary and primary circuits are calculated** based **on their flow rate and constant linear pressure losses** (e.g.r = 10 mm w.g./m)

3. **The total pressure loss of the circuit is calculated** by adding together:

   a) **The pressure loss of the final terminal;**

   b) **The minimum differential pressure of the autoflow devices;**

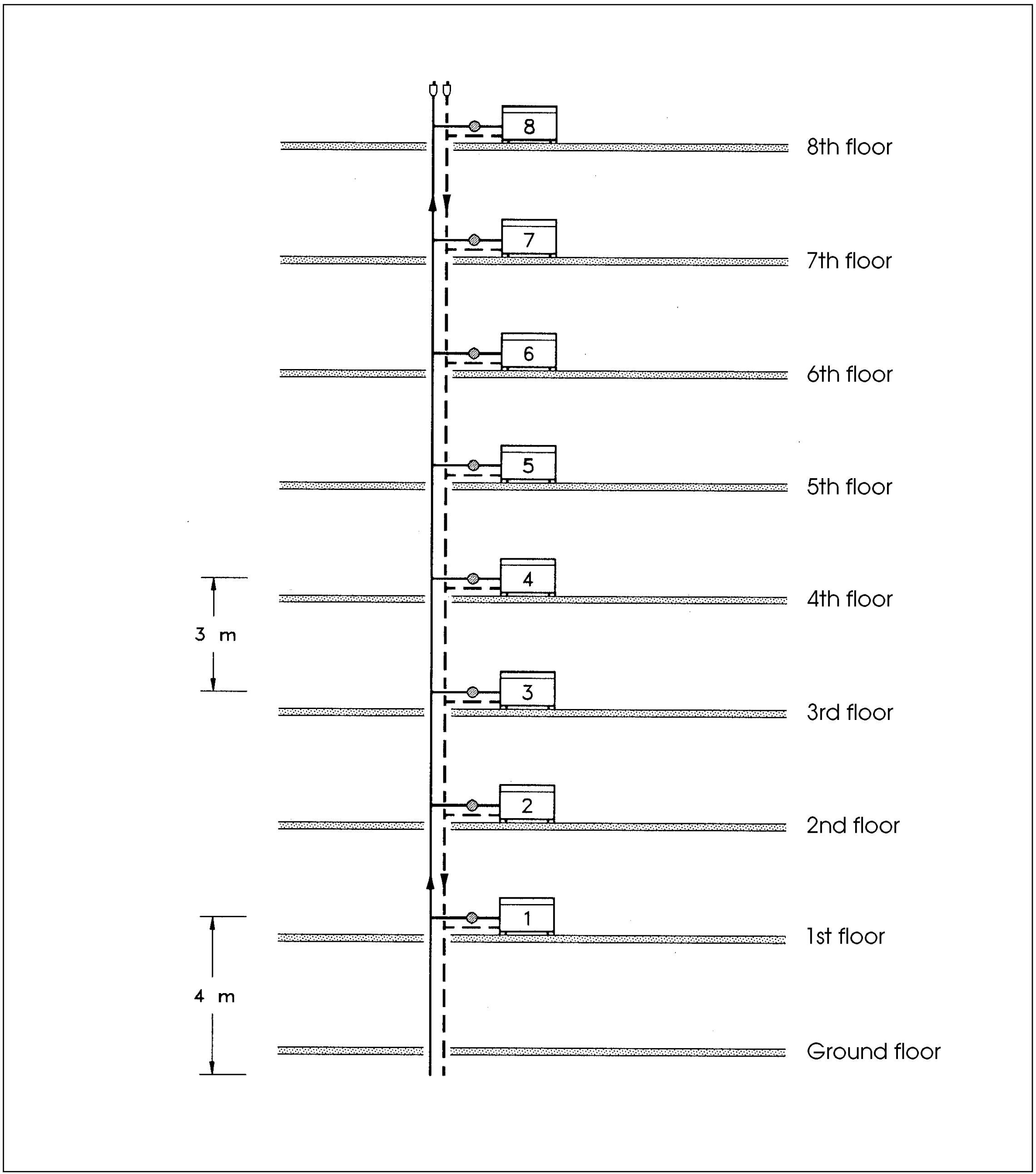
   c) **The continuous pressure loss (h) of the circuit** calculated in the conventional manner by multiplying the following values together:
   r = assumed value for the linear head losses (See 2) and
   l = lenght of pipes (of the main and secondary circuits) serving the final terminal;

   d) **The localised pressure loss (z) of the circuit,** conventionally considered equal to a percentage of the continuous pressure loss (h).
   The following values are normally considered:
   $\mathbf{z = 0,6 \cdot h}$ for runs with a few bends,
   $\mathbf{z = 0,7 \cdot h}$ for runs with several bends.

Example 1: Calculation of a circuit with autoflow

Determine the dimensions of the autoflow circuit illustrated below using the method of calculation detailed on the previous page. The following values are used:

- G = 330 l/h nominal flow rate of each fan coil unit,
- ΔP = 150 mm w.g. pressure loss corresponding to the nominal flow rate,
- ΔK = 1.250 mm w.g. value assumed as the minimum differential pressure of the autoflow device,
- l = 4 m lenght of the riser - fan coil unit connections,
- n = 2 (a 90°) bends in the riser - fan coil unit connections.



Procedure:

1 - Determining the flow rates and choosing the autoflow regulator

Each terminal requires a flow rates: G = 330 l/h.
It is assumed that autoflow regulator capable of providing these flow rates are available commercially.

2 - Calculating the dimensions of the pipes

For calculating the pipes it is assumed : r = 10 mm w.g./m (See INTRODUCTORY NOTES)

— Secondary circuits:

on the basis of the flow rate required by the terminals (330 l/h) and the assumed r value, their diameter is equal to 1/2”

— Sections of the main circuit:

on the basis of their flow rate and the r value defined above, the diameter of these sections is:

•Riser sections 8-7:	G_{8-7} = 330 l/h		dia ø = 3/4”
•Riser sections 7-6:	$G_{7-6} = G_{8-7} + G_7$ =	330 + 330 = 660 l/h	dia ø = 3/4”
•Riser sections 6-5:	$G_{6-5} = G_{7-6} + G_6$ =	660 + 330 = 990 l/h	dia ø = 1”
•Riser sections 5-4:	$G_{5-4} = G_{6-5} + G_5$ =	990 + 330 = 1.320 l/h	dia ø = 1 1/4”
•Riser sections 4-3:	$G_{4-3} = G_{5-4} + G_4$ =	1.320 + 330 = 1.650 l/h	dia ø = 1 1/4”
•Riser sections 3-2:	$G_{3-2} = G_{4-3} + G_3$ =	1.650 + 330 = 1.980 l/h	dia ø = 1 1/4”
•Riser sections 2-1:	$G_{2-1} = G_{3-2} + G_2$ =	1.980 + 330 = 2.310 l/h	dia ø = 1 1/4”
•Riser sections 1-0:	$G_{1-0} = G_{2-1} + G_1$ =	2.310 + 330 = 2.640 l/h	dia ø = 1 1/2”

3 - Total pressure loss of the circuit

The total pressure loss of the circuit is calculated by adding together:

a) The pressure loss of the final terminal: ΔP = 150 mm w.g.

b) The minimum differential pressure of the autoflow regulator: ΔK = 1.250 mm w.g.

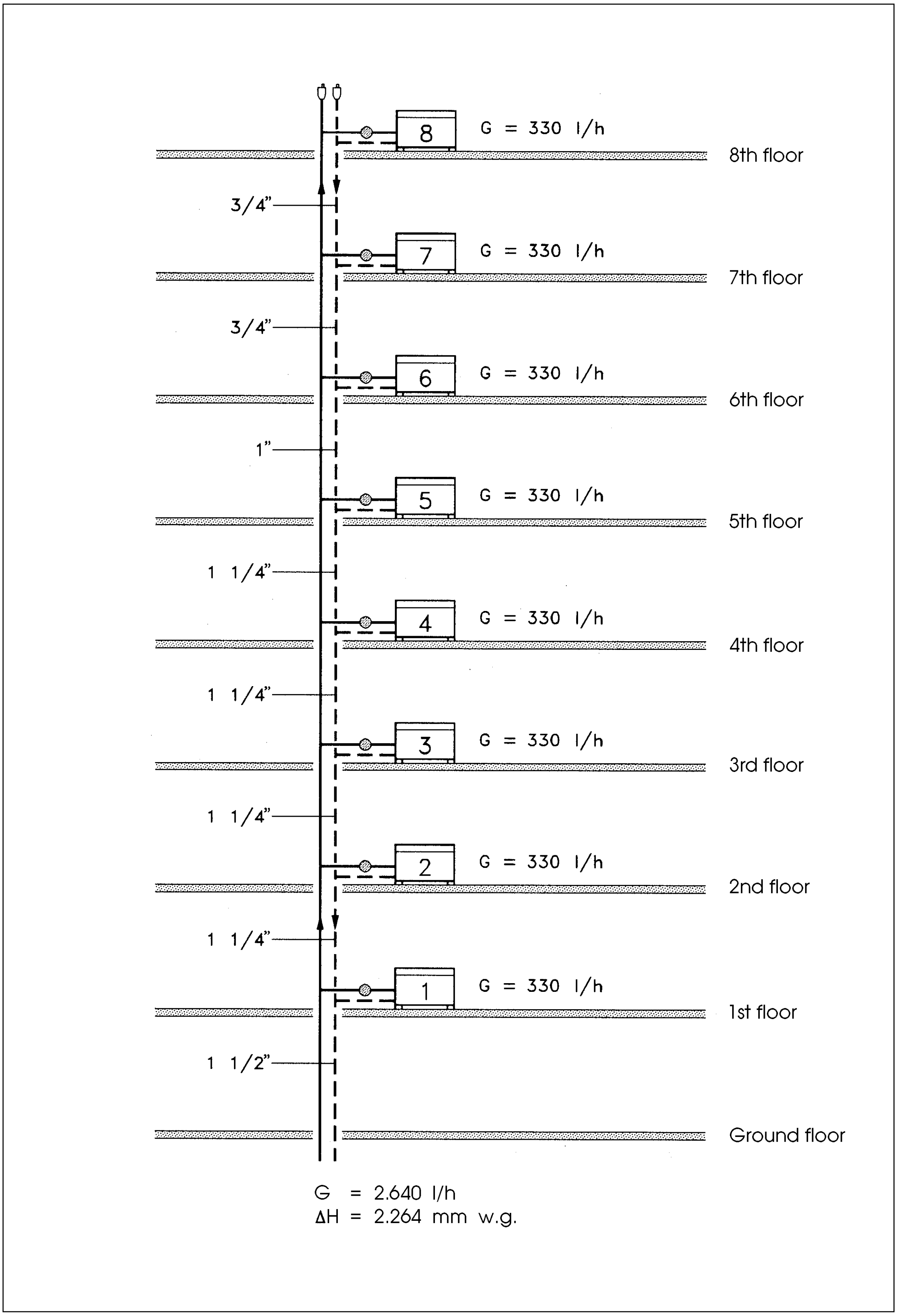
c) The continuous pressure loss of the circuit (See calculation conventions) with:
Where: r = 10 mm w.g/m
l = (7 · 3 + 4) · 2 + 4 = 54 m
giving: h = 10 · 54 = 540 mm w.g.

d) The localised pressure loss of the circuit (See calculation conventions)
z = 0,6 · 540 = 324 mm w.g. (the circuit has few bends)

The following result is obtained therefore:
H = ΔP + ΔK + h + z = 150 + 1.250 + 540 + 324 = 2.264 mm w.g.

Note:

The minimum differential pressure required by autoflow regulators causes a significant increase in the head required at the base of the risers. It will be found however that this increase is less on a percentage basis if it is assessed in the general context of the system.


8
G = 330 l/h
8th floor
3/4"
7
G = 330 l/h
7th floor
3/4"
6
G = 330 l/h
6th floor
1"
5
G = 330 l/h
5th floor
1 1/4"
4
G = 330 l/h
4th floor
1 1/4"
3
G = 330 l/h
3rd floor
1 1/4"
2
G = 330 l/h
2nd floor
1 1/4"
1
G = 330 l/h
1st floor
1 1/2"
Ground floor
G = 2.640 l/h
ΔH = 2.264 mm w.g.

CHARACTERISTICS OF CIRCUITS WITH AUTOFLOW

With these circuits the flow rate passing through subsidiary units can be automatically regulated by the action of suitable devices. Automatic regulation to maintain design flow rate will provide balanced distribution in both routine and complex systems.

Unlike circuits with balancing valves, circuits with autoflow:

- **are simple to calculate;**
- **do not require calibration operations;**
- **are not subject to the dangers of out of calibration operations.**

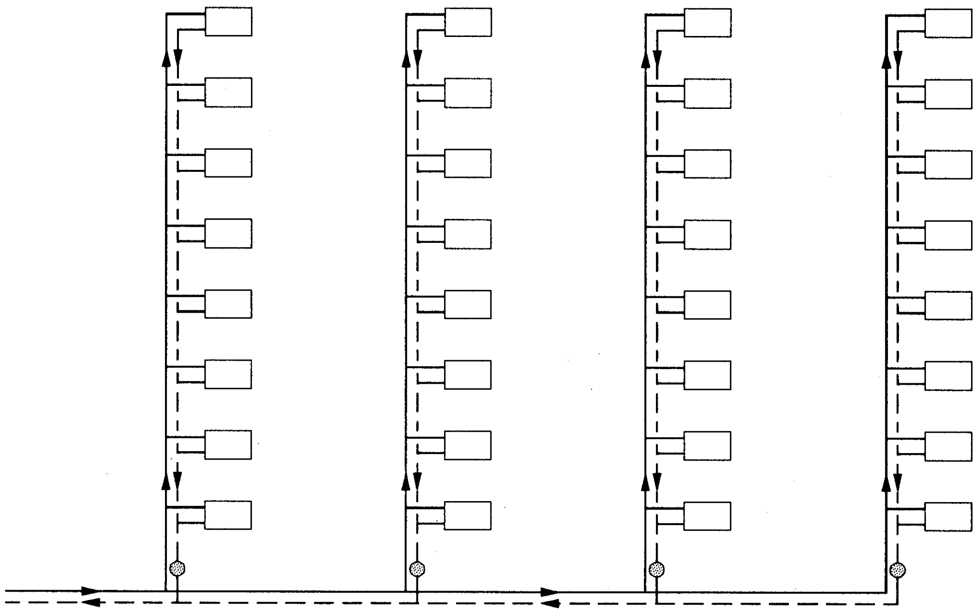
Another major advantage is that when design changes are necessary these circuits can easily be adapted to the new operating conditions. Autoflow devices with dynamic type regulation i.e. with a mobile piston, are capable of maintaining constant flow rates at the emitters over a wide range of differential pressures

The minimum differential pressure required by autoflow systems must be compared to the requirements when using manual balancing valves. Typically proportionally balanced systems require manual balancing valves on the rise, branch and terminal.These wide open pressure drops are often equal to or greater than the autoflow minimum requirement.

APPLICATIONS FOR WHICH AUTOFLOW CIRCUITS CAN BE USED

Circuits with autoflow are used in air conditioning systems primarily to provide the required flow rates at the risers, the zone subsidiary units or the emitters.

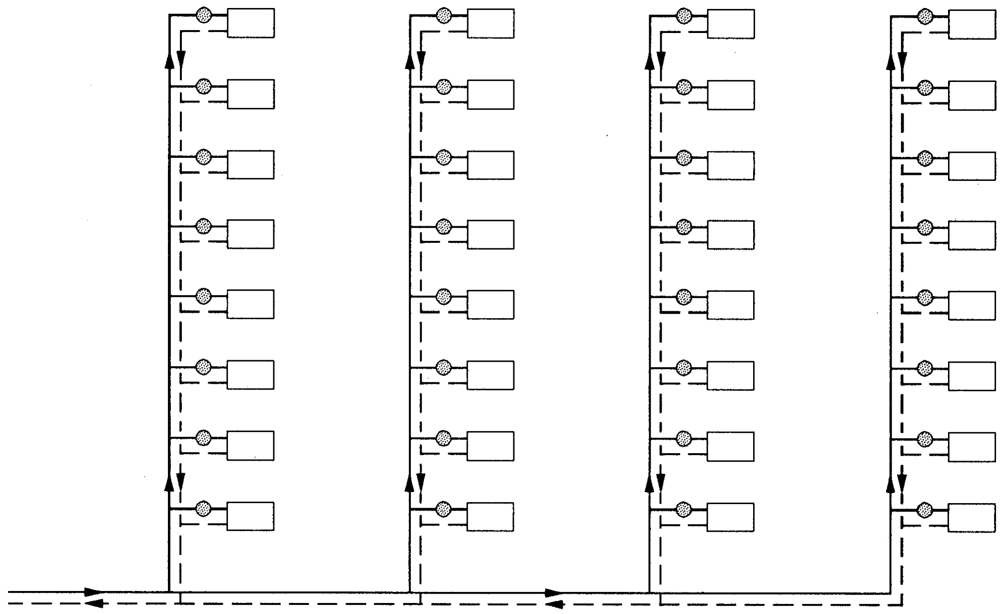
Circuit with autoflow at the foot of a column



Circuit with autoflow at the foot of the riser

They provide the required flow rates at the risers.
Imbalances in flow rate distribution can however occur along the risers for the reasons given in the section relating to simple circuits. For this reason these circuits are normally used in buildings with no more than 5 or 6 floors.

Circuits with autoflow at each terminal

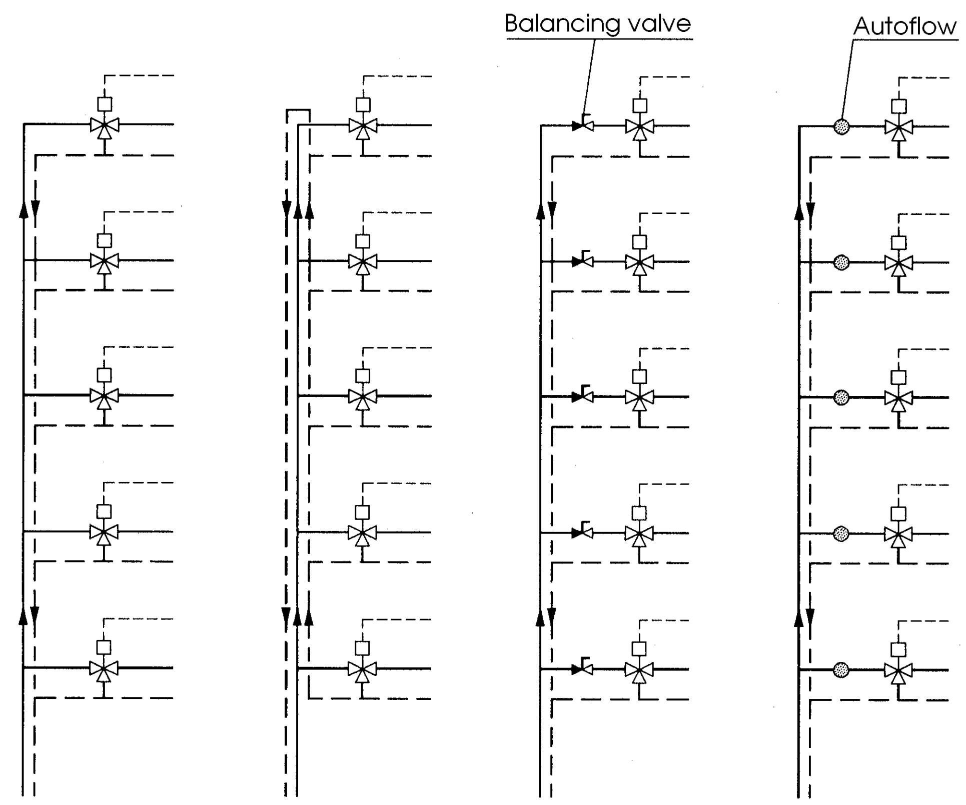


Circuits with autoflow at each emitter

These circuits are capable of guaranteeing the required water flow through each emitter

CIRCUITS WITH THREEWAY CONTROL VALVES

In circuits using this type of control the flow rate or temperature of the fluid passing through the emitters can be varied by the action of the automatic three-way valves. They can be used with simple systems, with compensated return, with balancing valves or with autoflow.



Type diagrams of circuits with three-way control valves

DETERMINING THE DIMENSIONS OF CIRCUITS WITH THREEWAY CONTROL VALVES

The dimensions of these circuits are determined with open valves adopting the same criteria used for circuits without regulating valves (See "SIMPLE CIRCUITS, REVERSE RETURN CIRCUITS, CIRCUITS WITH BALANCING VALVES AND CIRCUITS WITH AUTOFLOW").
The imbalances associated with the closing of the valves should then be analysed (i.e. at the opening of their bypasses) and, if necessary, the arrangements for keeping these imbalances under control should then be specified.

From a practical point of view the following procedure can be adopted when considering the following cases:

1. **zone systems,**
2. **modulation of water flow rate through systems with fan coil units,**
3. **circuits with temperature control and coil control.**

On zone systems it is always necessary to balance the bypasses of the valves (with calibrated disks, balancing valves or autoflow). If this balancing is not carried out then even a limited number of closed valves may, because of the relatively low pressure loss through their uncalibrated open bypasses, reduce water flow through some heat emitters, still with their valves in the open position.

Fan coil units with modulating valves may overflow by passes if not balanced. This may results in flow shortages to remote units (terminals).
Autoflow has the additional advantage of maintaining constant flow at all positions of modulation further preventing overflows.

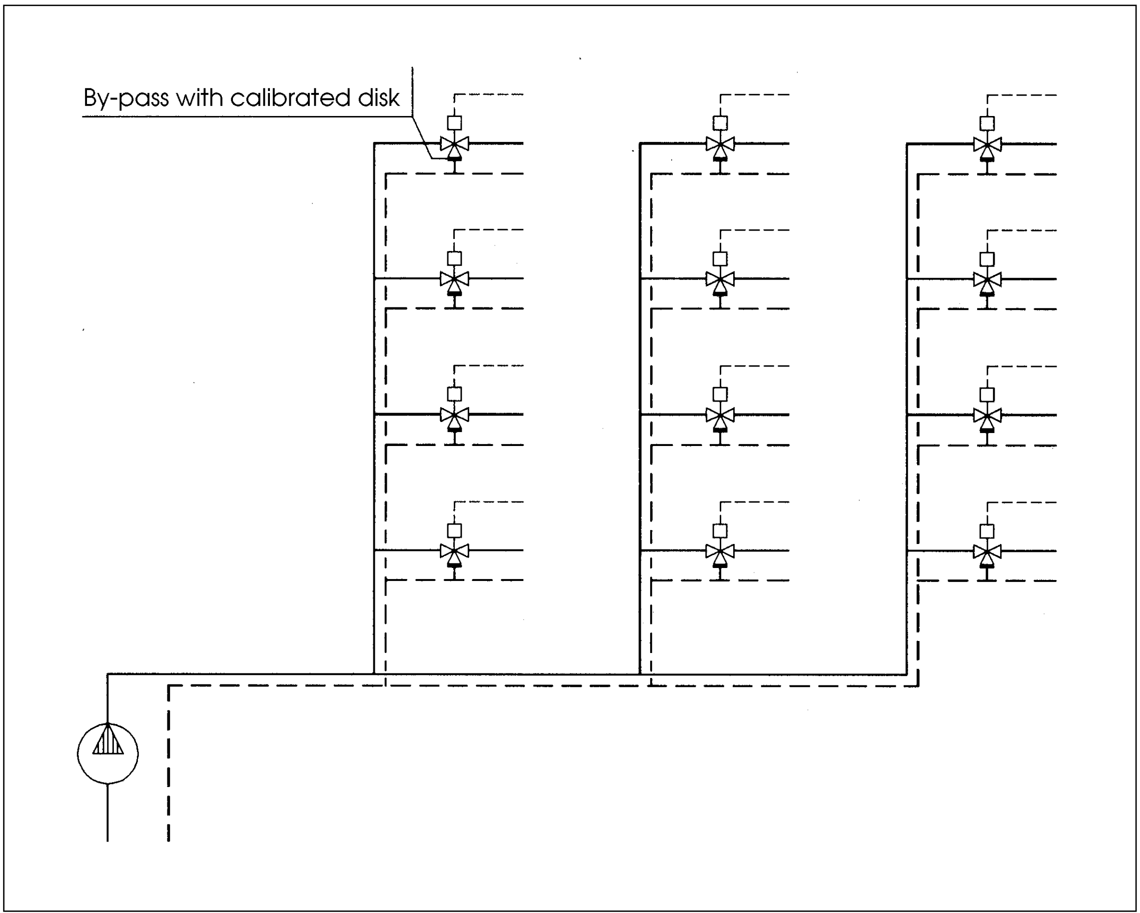
In the case of the temperature control or coil control circuits the need or not to balance the bypasses essentially depends on the type of circuit, on the valves used and on the emitters served. If the baypasses are not balanced a check will have to be made from time to time to ensure that the closing of the valves will not lead, in these circuits, to:
– **excessive velocities** (causing noise and abrasion) or
– **circulation feed back** which would have a harmful effect on the functioning of the emitters.

The functional diagrams illustrating these cases and the relative balancing systems are given below.

Note:
The three-way valves which are slaved to an on/off control can operate as required:
- either as **diverters** (1 inlet and 2 outlets) or
- as **mixers** (2 inlets e 1 outlet).

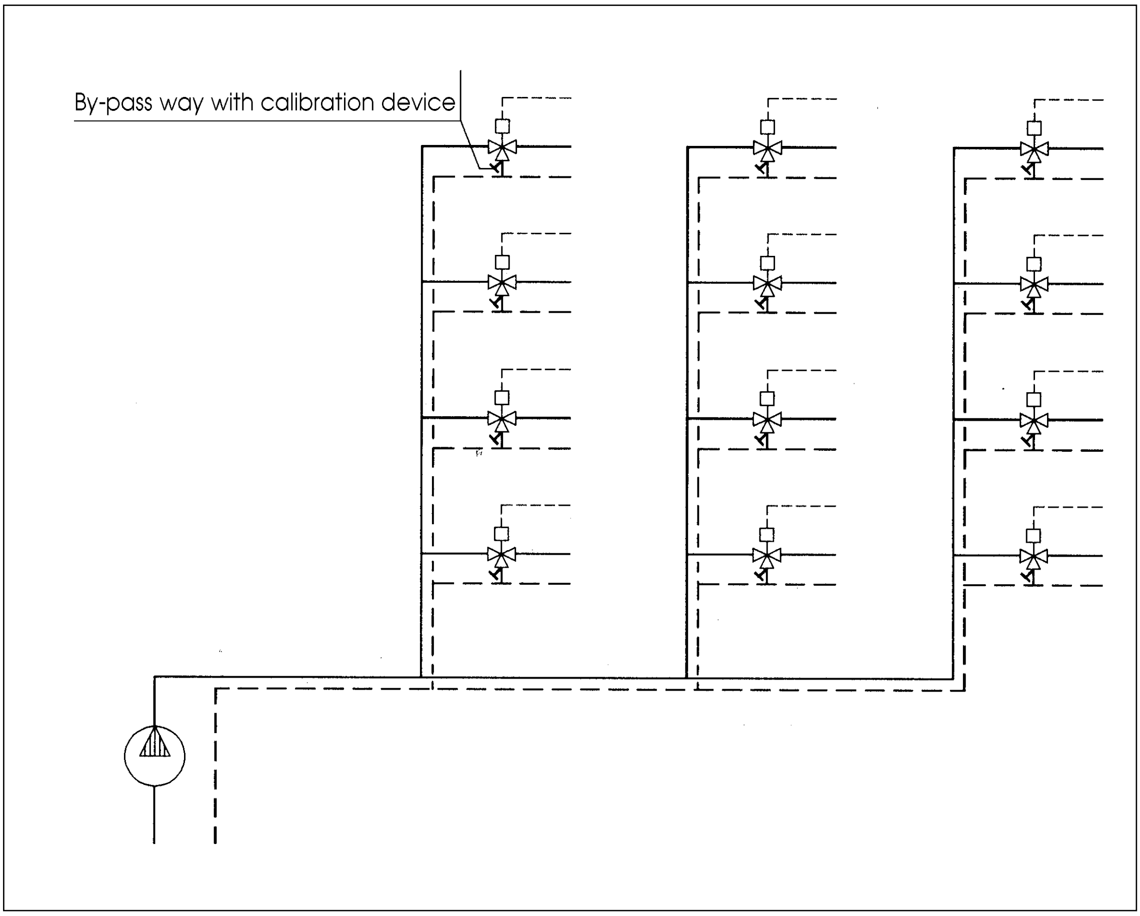
On the other hand three-way valves which are slaved to a modulating control operate more efficiently as mixers.

Zone systems: balancing with calibrated disks



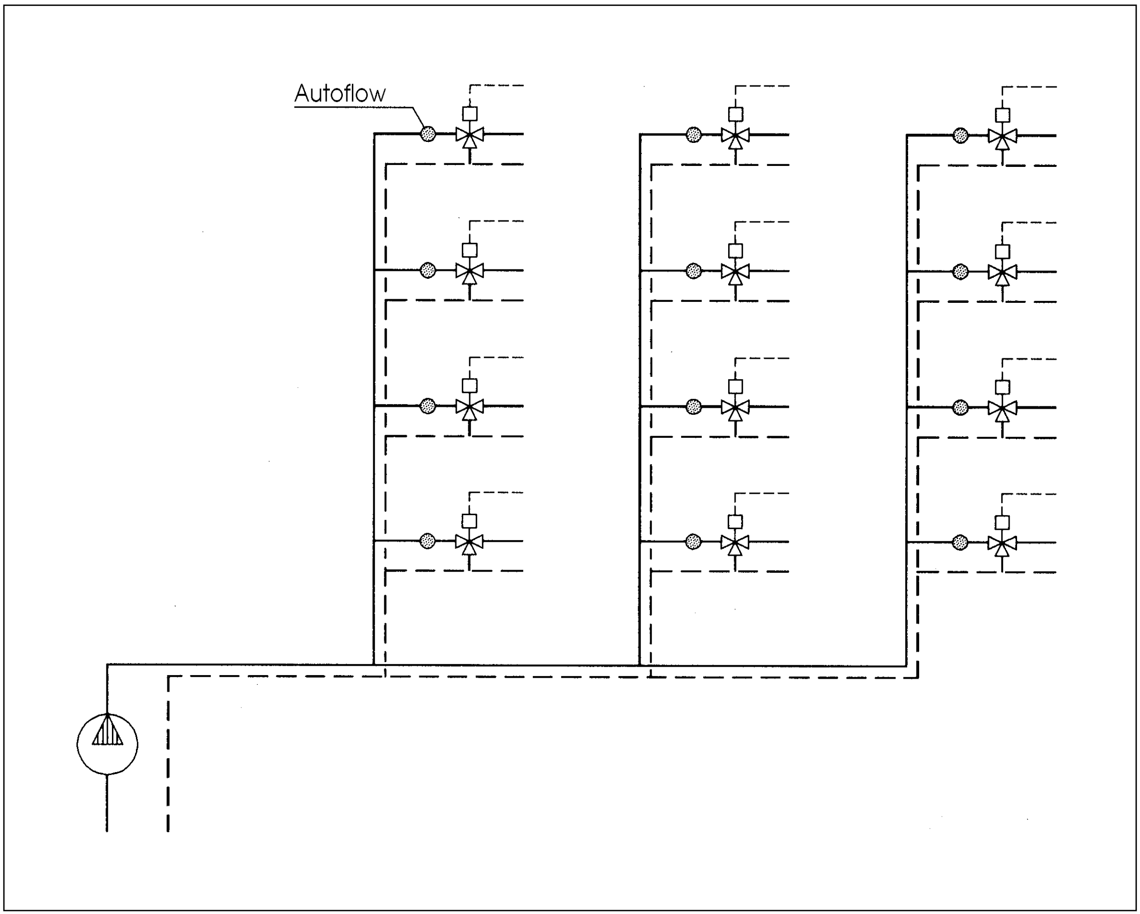
The calibrated disks (located on the bypasses) **must be selected such that the design water flow passing through the bypass** (when the zone outlet is closed) **sees a resistance equal to that of the zone** (when the bypass is closed).
This method of balancing, which is a compromise due to the impractically large number of different diameter disks needed. If undertaken with care it will be sufficiently reliable however and has the advantage of not falling out of calibration.

Zone systems: balancing with balancing valves



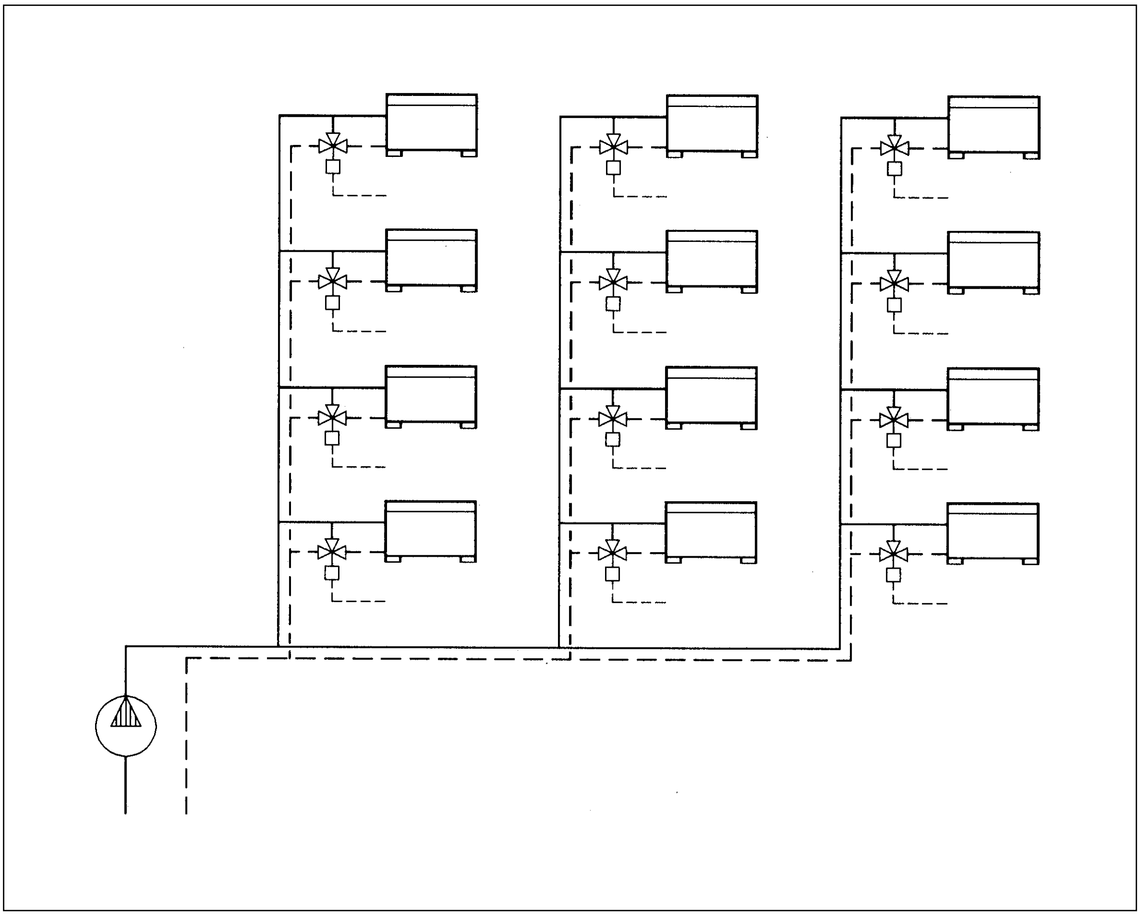
The balancing valves (located on bypasses) **must be adjusted in such a way as to oppose the same pressure drop as the zone circuit served.**

Zone systems: balancing with autoflow devices

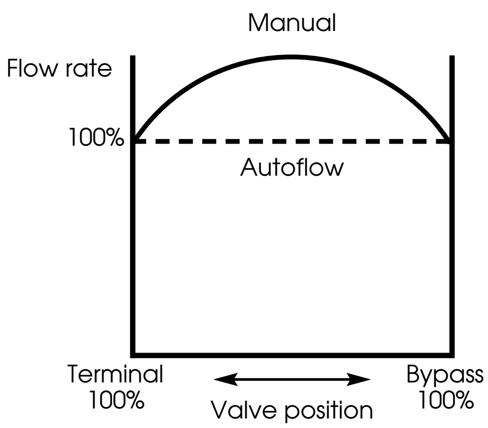


With autoflow devices (located on the return section of the zone) **the flow rate of each zone can be kept at a constant level either with open valves or closed valves.**

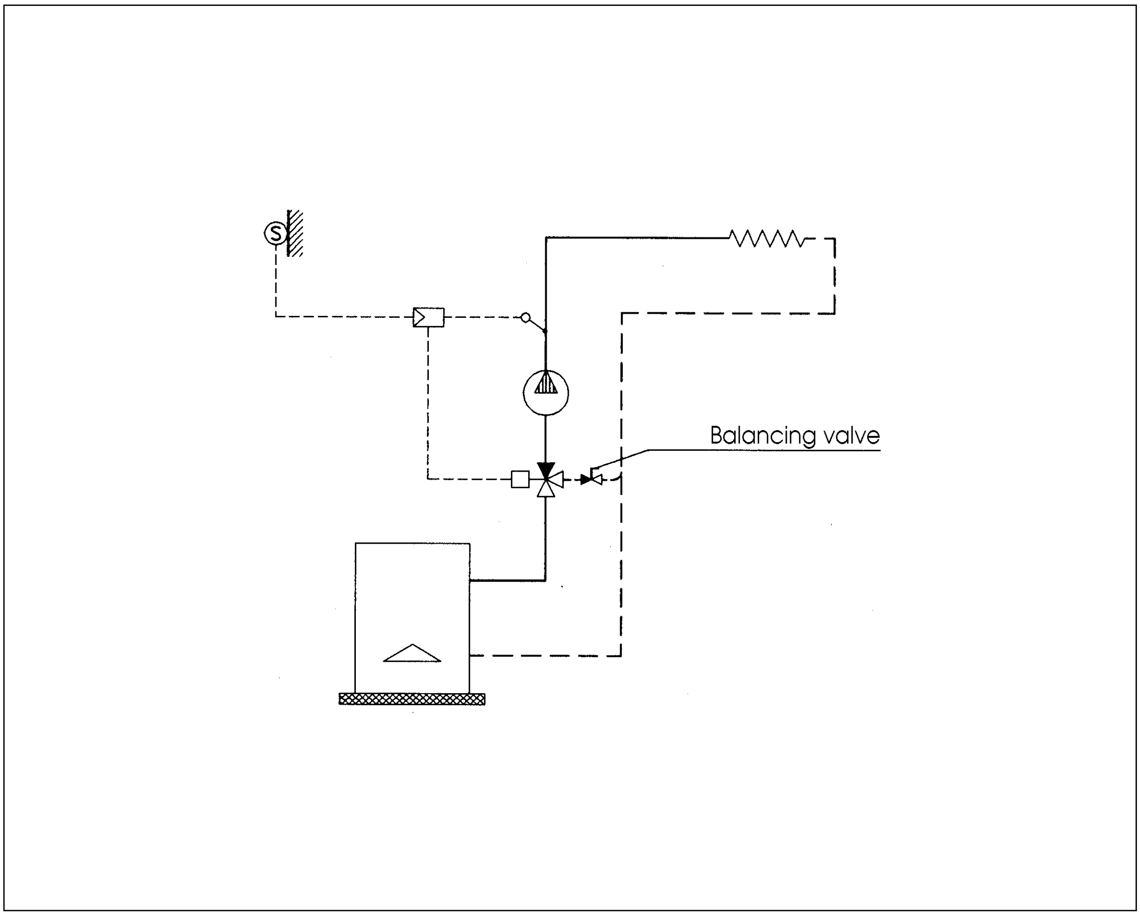
Systems with fan coil units and modulating control



Fan coil units **with modulating valves** (especially on cooling) **need balancing of both terminal and by pass to prevent overflows. Autoflow has additional benefit of preventing overflows at all points of operation.**

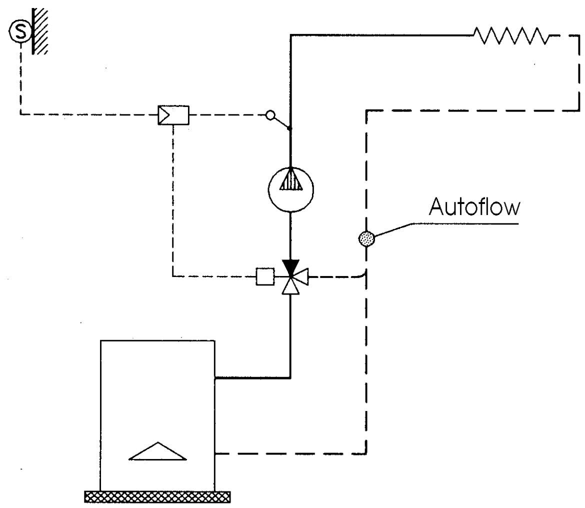


Temperature control: balancing with balancing valve



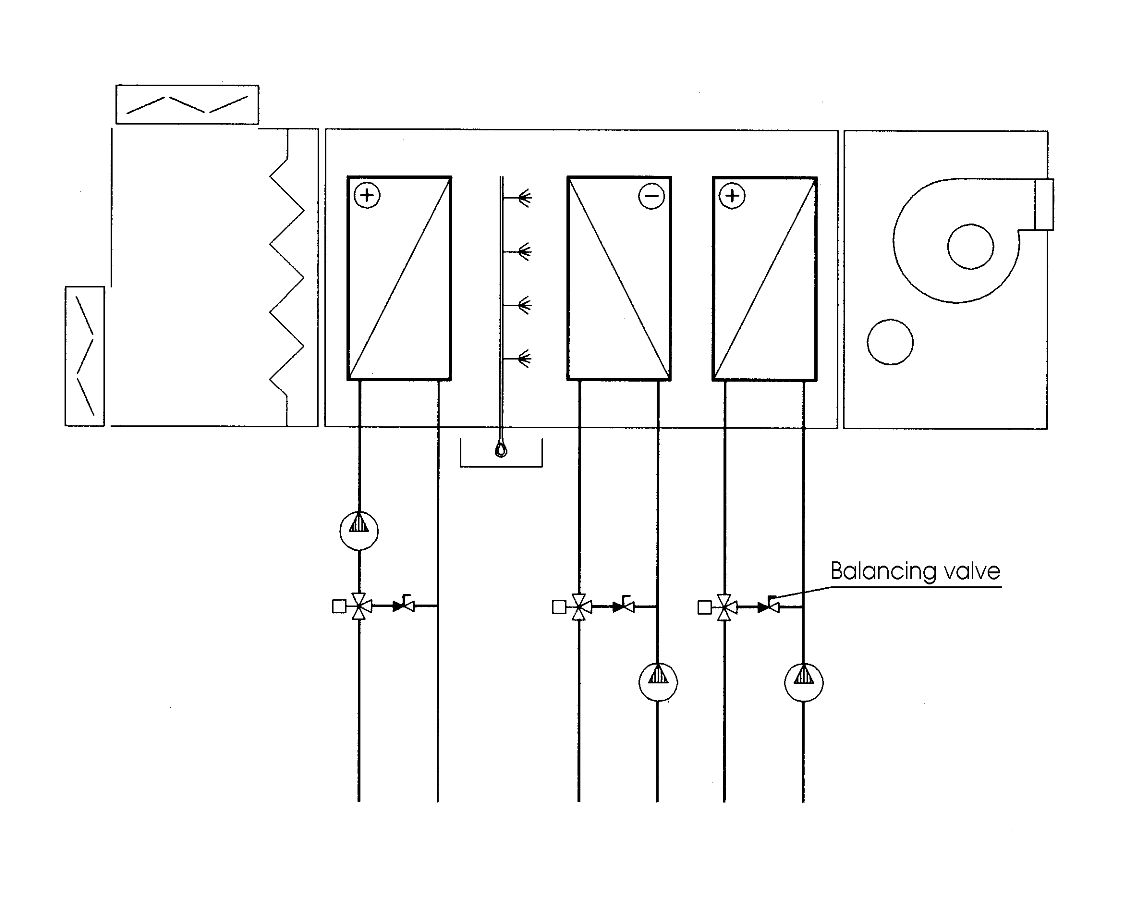
The balancing valve (located on bypass) **must be adjusted so that the pressure loss through the bypass at full design flow rate, is equal to the circuit pressure loss, circuit closed by the control valve: the boiler circuit in this case.**

Temperature control: balancing with autoflow



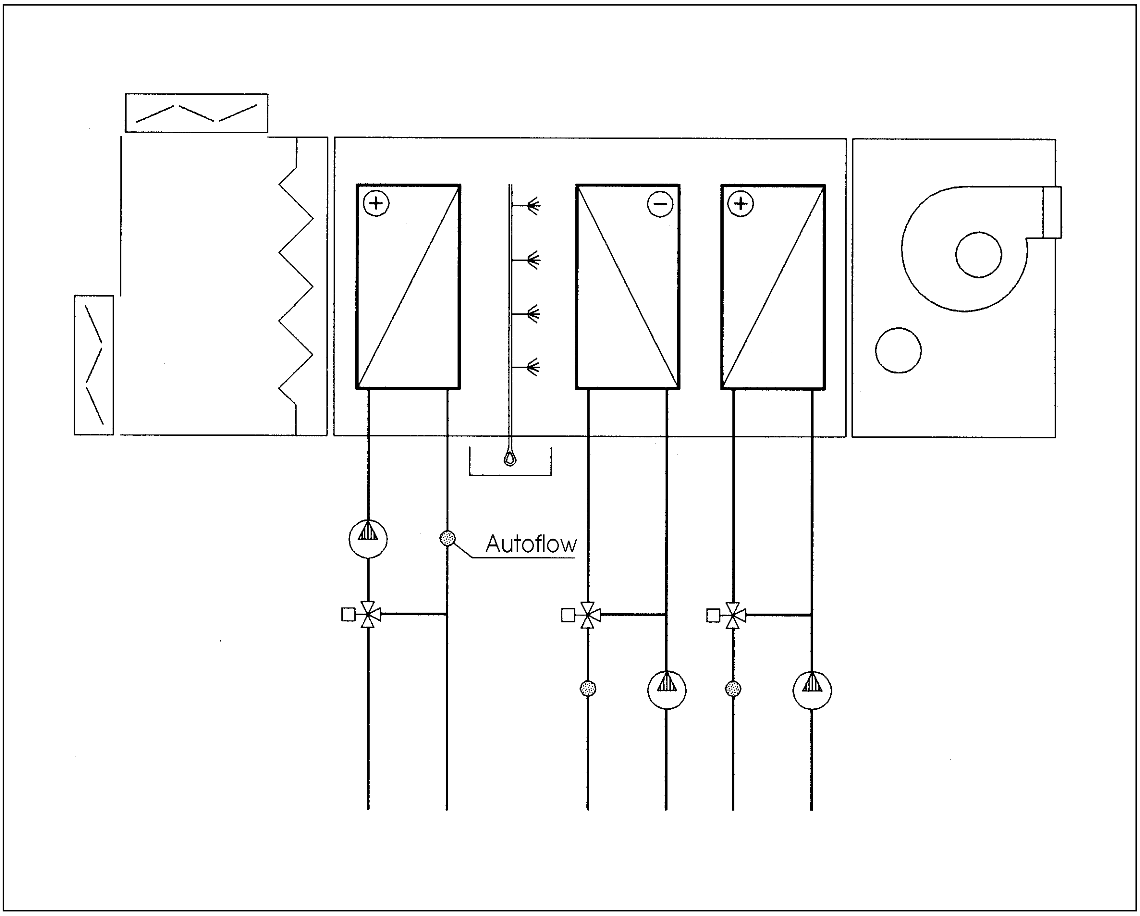
With an autoflow device the flow rate in the circuit can be kept at a constant level either with the valve open or with the valve closed.

Control of coils: balancing with balancing valves



The balancing valves (located on the bypass) **must be adjusted so that the pressure loss through the bypass at full design flow rate, is equal to the circuit pressure loss, when the bypass is closed.**

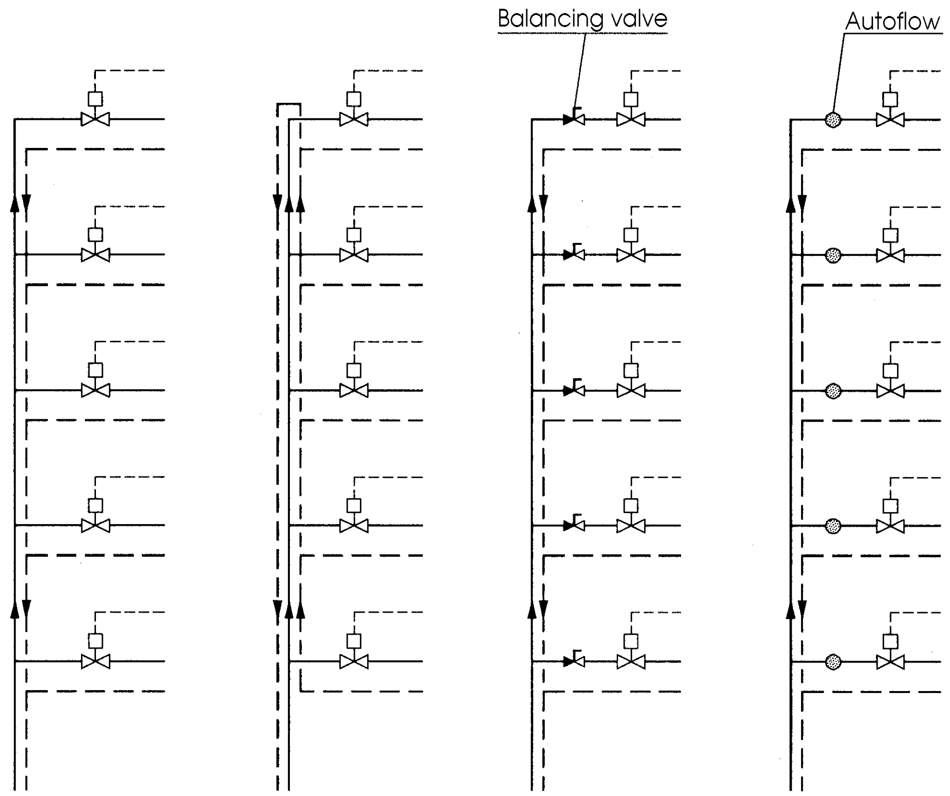
Control of coils: balancing with autoflow



With the autoflow devices the flow rates in the distribution circuits can be kept at a constant level either with the valves open or closed.

CIRCUITS WITH TWO-WAY CONTROL VALVES

With these circuits the flow rate of water passing through the emitters can be varied by means of automatic two-way valves. They can be used with simple systems, compensated return, with balancing valves or with autoflow devices.



Diagrams of circuits with two-way control valves

DETERMING THE DIMENSIONS OF CIRCUITS WITH TWO-WAY CONTROL VALVES

The dimensions of these circuits are calculated with valves open adopting the same criteria used for circuits without control valves (See “SIMPLE CIRCUITS, REVERSE RETURN CIRCUITS, CIRCUITS WITH BALANCING VALVES AND CIRCUITS WITH AUTOFLOW”).
The imbalances associated with the closing of the valves should then be analysed and, if necessary, controls added which are capable of preventing this closing from causing excessive differential pressures along the circuit and consequently:
– noise,
– erosion,
– incomplete closing of the thermostatic valves,
– operation outside the performance curve of the circulating pumps.

A reasonably accurate and practical method can be used to consider: **small systems** (i.e. systems for single accomodation) **and medium to large systems.**

The increase in the differential pressures **within small systems** can be controlled simply by installing a pressure limiting device at the base of the circuit.

On the other hand the increase in the differential pressures **on medium to large systems** must be controlled not onlt at the base but also along the circuit. **Both partial control arrangements** (with pressure limiting device located at the base of the risers) **or total control arrangements** (with pressure limiting device located on every derived circuit) can be used.

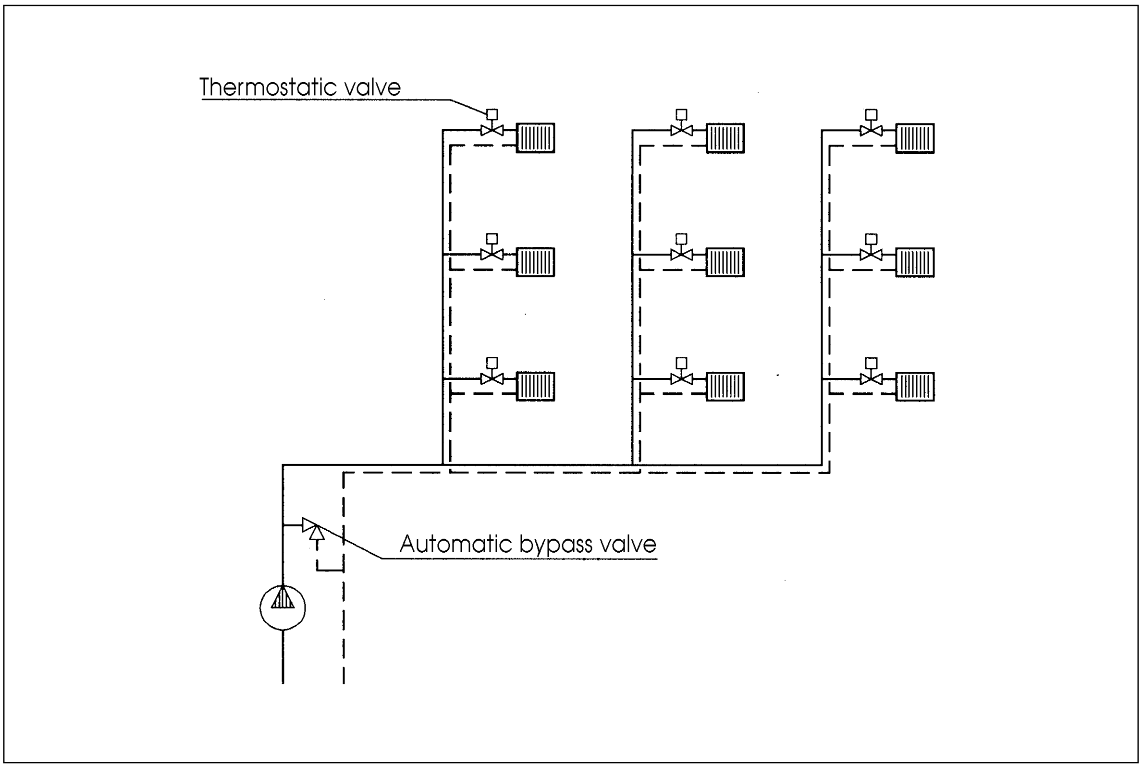
Another solution is variable speed pumps to prevent excessive differential pressure increase. they can also save energy while solving previously mentioned problems.

In the case of systems with very extensive network it is advisable to make a thorough analysis of all possible arrangements taking into account:
– the variation in the differential pressures when the valves are closed,
– the characteristics of the emitters,
– the degree of reliability of the maintenance service,
– the cost of the various designs which could be adopted and
– the costs of the operation of the circulating pumps.

For the development of this type of analysis, which is too complex to be dealt with in this publication, reference should be made to the works of P.Fridmann and ASHRAE listed in the bibliography.

The functional diagrams illustrating the arrangements mentioned above are given below.

Automatic bypass valve located at the base of the circuit

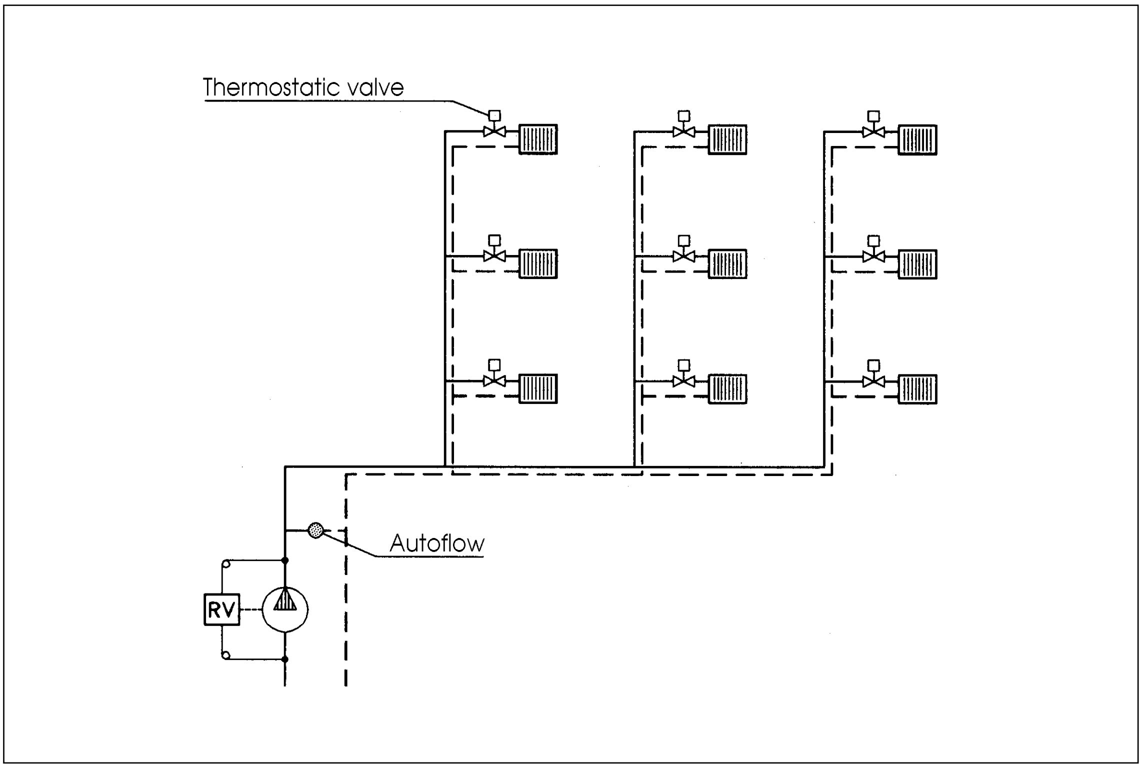


This arrangement is used mainly in small systems using radiators fitted with thermostatic valves.

- **Discharge rate of the automatic bypass valve (Gv):**
  Its value depends on the total flow rate of systems with open valves **(G)** and on the type of control used. The following is normally assumed:
  Gv = 0,6 · G on systems with temperature control and
  Gv = G on sistems without temperature control.

- **Opening pressure of the automatic bypass valves:**
  It is recommended that this value is approximately 10% greater than the differential pressures between the points of the circuit connected by the bypass when all valves are open.

Differential pressure limiting device located at the base of the circuit



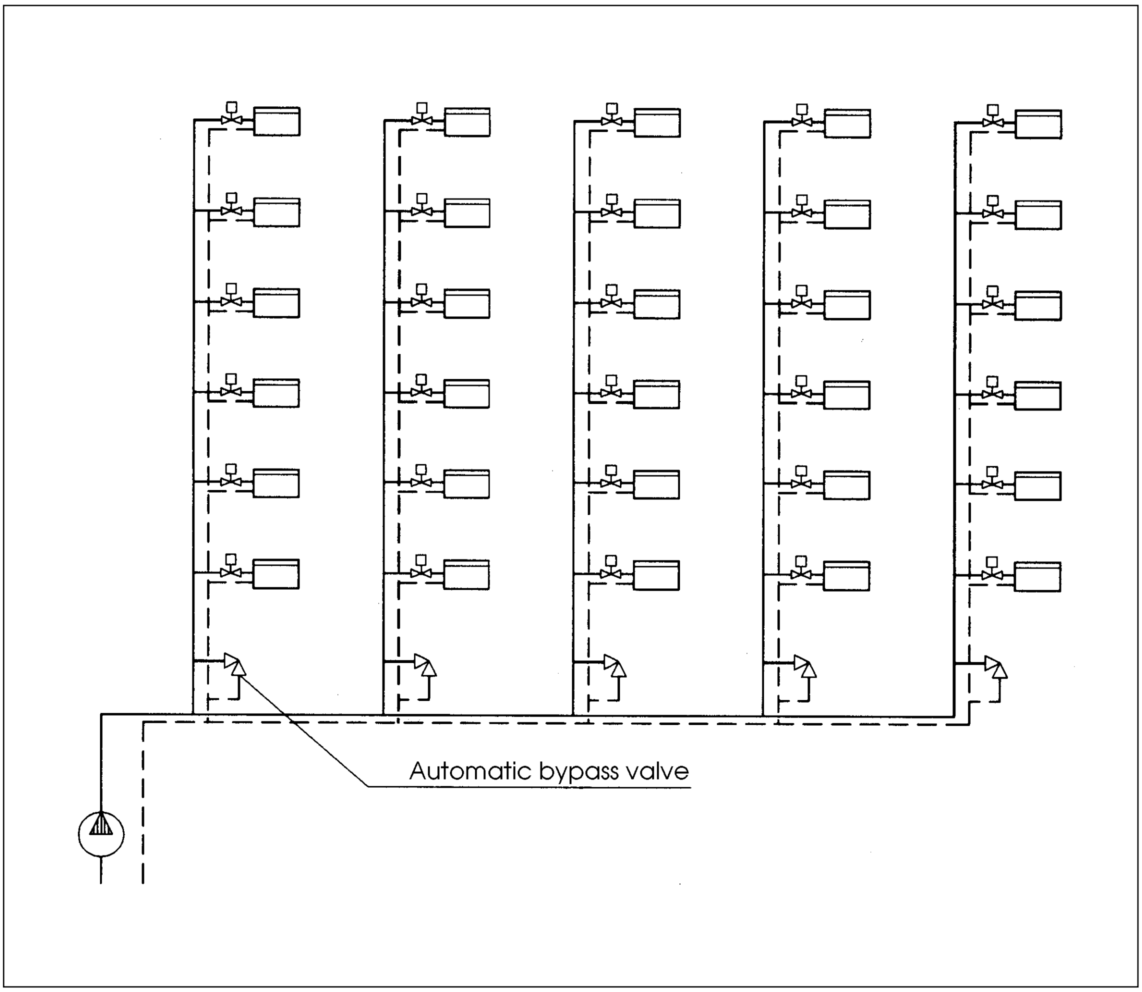
This arrangement is used (as was the arrangement of the previous page) **mainly on small systems using radiators fitted with thermostatic valves.**

- **Setting pressure of the limiting device:**
  It is recommended that this value is approximately 10% greater than the differential pressures between the points of the circuit in which are fitted the pressure ports of the limiting device when all valves are open.

Note:

The bypass with autoflow device is used to guarantee a minimum flow rate even the thermostatic valves are closed. This flow rate is required to prevent the thermal inertia of the heat generator (boiler) causing localised overheating of the fluid and operation of the overheat thermostat or other safety devices such as the fuel valve or temperature controlled safety valves.

Automatic bypass valves positioned at the base of the risers

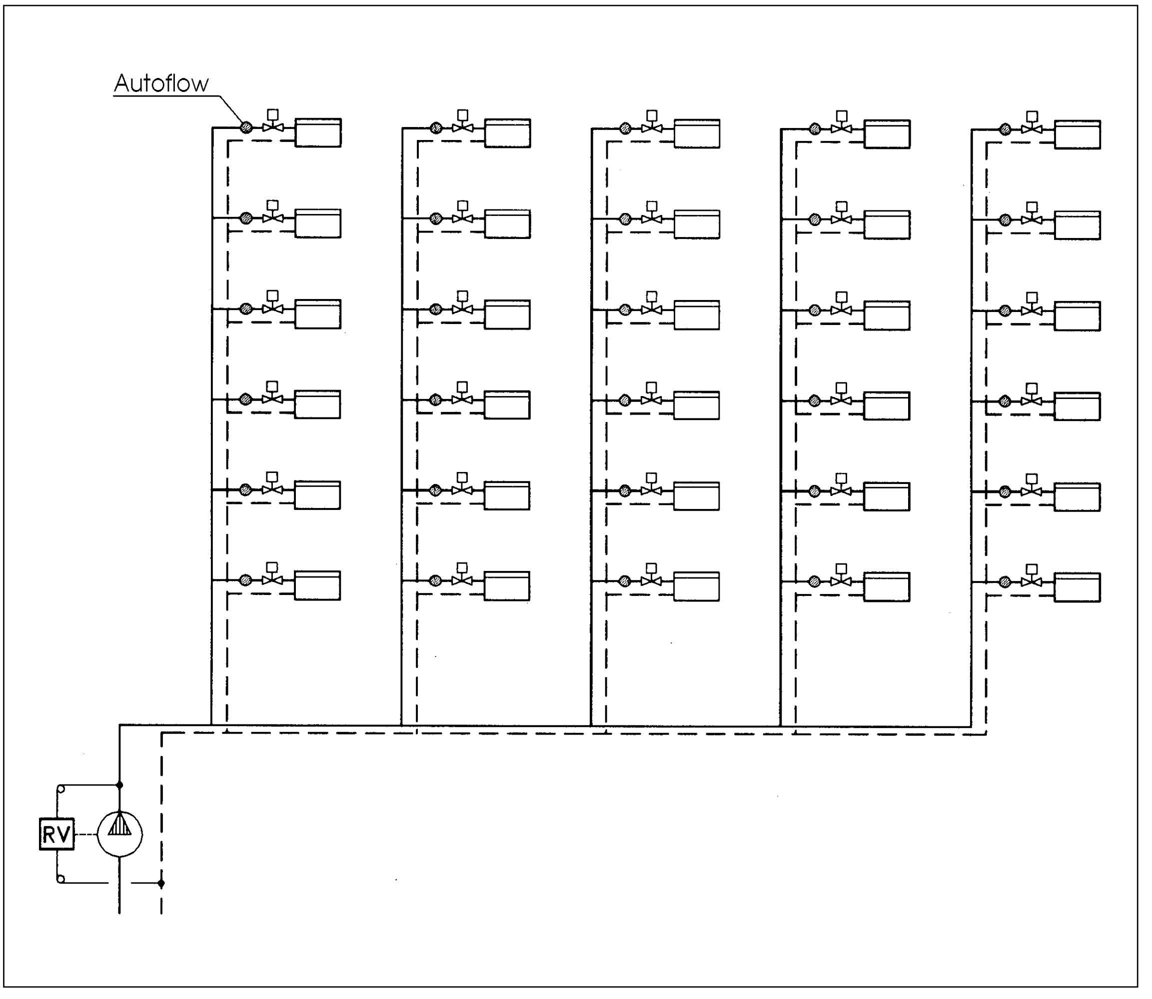


This arrangement is normally used in radiator systems fitted with thermostatic valves. It is not advisable to install it in buildings with more than 6 or 7 floors.

- **Discharge rates of the automatic bypass valves:**
  The same flow rates as for the risers with all valves open are assumed;

- **Opening pressure of the automatic bypass valves:**
  It is recommended that this value is approximately 10% greater than the differential pressures between the points of the circuit connected by the bypass when all valves are open.

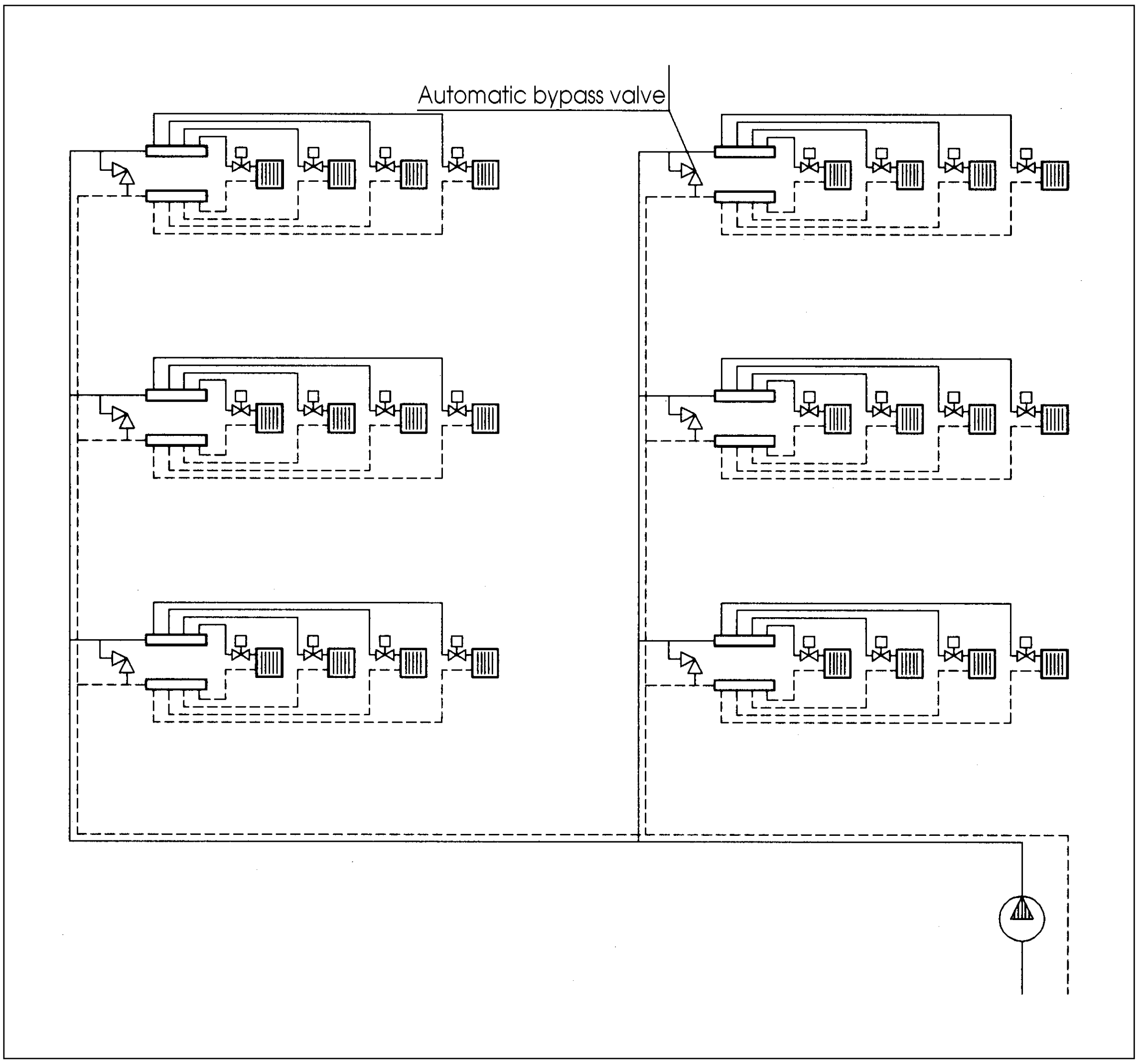
Differential pressure limiting device installed at the base of the circuit and terminals with autoflow



This arrangement is used mainly in systems with fan coil units and modulating control valves
The autoflow devices are used to stabilise the amount of fluid passing through the terminals.

- **Setting pressure of the limiting device:**
  It is recommended that this value is approximately 10% greater than the differential pressures between the points of the circuit in which are fitted the pressure ports of the limiting device when all valves are open.

Automatic bypass valves located on each zone



This arrangement is normally used in zone systems fitted with thermostatic valves. Using automatic bypass valves the increases in differential pressure at each zone branching point can be controlled.

- **Discharge rates of the automatic bypass valves:**
  The same flow rates as those achivied with the thermostatic valves open are assumed for the bypass valves;

- **Opening pressure of the automatic bypass valves:**
  It is recommended that this value is approximately 10% greater than the differential pressures between the points of the circuit connected by the bypass when all valves are open.

HEATING AND AIR CONDITIONING SYSTEM TERMINAL UNITS

Contents

UNIT HEARTERS

- CLASSIFICATION
  - HORIZONTAL UNIT HEATERS
  - VERTICAL UNIT HEATERS
- CHOICE OF UNIT HEATERS
  - TYPE AND ARRANGEMENT OF THE UNIT HEATERS
  - TEMPERATURE OF THE AIR LEAVING THE UNIT HEATERS
  - PERMISSIBLE SOUND LEVEL
- MAINTENANCE OF UNIT HEATERS
- NOMINAL HEAT OUTPUT OF A UNIT HEATER
- DESIGN TEMPERATURE OF THE HEATING FLUID
- EFFECTIVE HEAT OUTPUT OF A UNIT HEATER
  - CORRECTION FACTOR FOR THE DIFFERENT TEMPERATURE OF THE FLUIDS
  - CORRECTION FACTOR FOR THE ALTITUDE EFFECT
  - CORRECTION FACTOR DEPENDING ON THE VELOCITY OF THE HEATING FLUID

Unit heaters are heat emitters which give off heat by means of forced convection. Basically they consist of:
– a finned heat exchanger,
– a fan and,
– a casing.

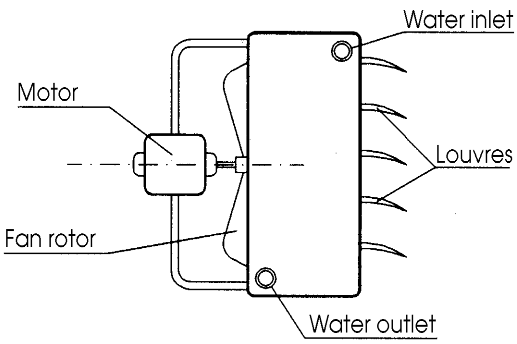
They are used to heat gymnasia, swimming pools, supermarkets, garages, laboratories, workshops etc.

CLASSIFICATION

Unit heaters can be divided into two categories based on the direction of their air flow, **i.e. horizontal or vertical.**

HORIZONTAL UNIT HEATERS

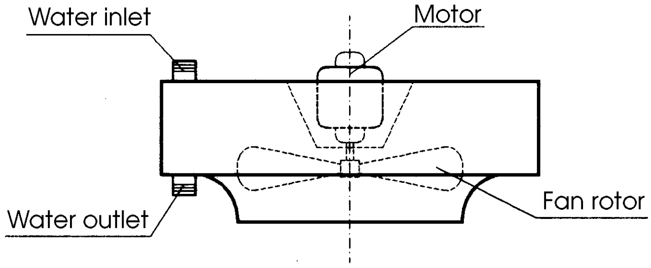
These are also called "wall" heaters and are used for heating areas which are not very high. They are provided with horizontal or vertical adjustable louvres to control the air flow.



Horizontal unit heater

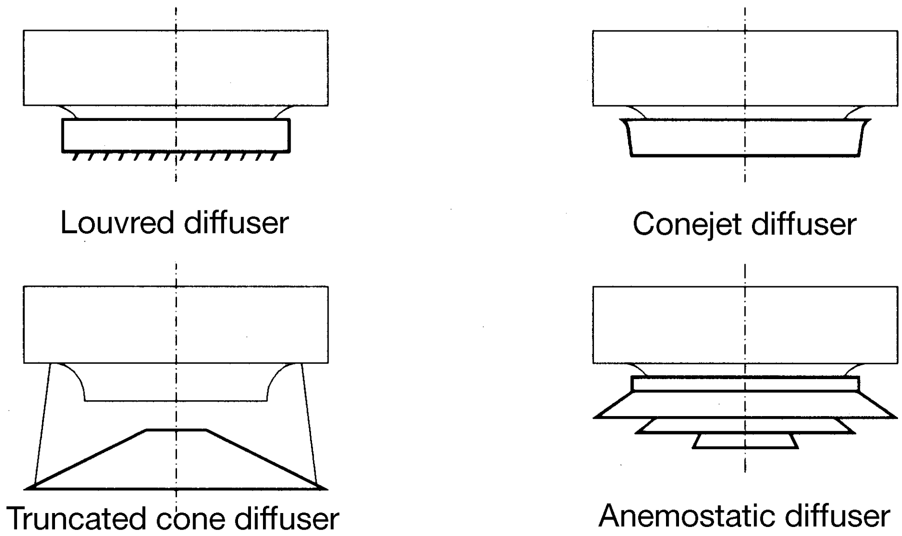
VERTICAL UNIT HEATERS

Also called “suspended” heaters and are used to heat areas with heights of up to about 20 to 25 metres.



Vertical unit heater

The air flow of these unit heaters can be adjusted **using louvred diffusers, conejet diffusers, truncated cone diffusers or anemostatic diffusers.**



CHOISE OF UNIT HEATER

The following factors must be considered in order to make the correct choice from these types of emitter:
– **installation of the unit heaters,**
– **outlet temperature of the air and**
– **noise level.**

INSTALLATION OF THE UNIT HEATERS

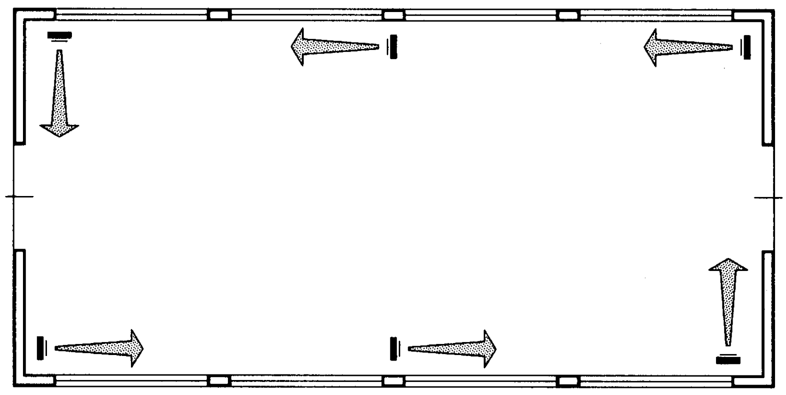
The type of unit heater should be chosen so that the positioning of these emitters can be such as to avoid the formation of excessively hot or cold areas. To achieve this the following general instructions should be followed:

- **install at least two emitters in each area;**
- **check that the sum of the hourly flow rates of the fans is not less than 3.5 times the volume of the area to be heated;**
- **arrange horizontal unit heaters with their air outlets facing external walls.** The most efficient arrangements are those with the air flow forming a continuous circuit tangential to the walls (See example 1);
- **arrange vertical unit heaters with air outlets which overlap each other;**
- **direct the flow of hot air against wide glazed zones or against large doors;**
- **avoid disturbing the flow of air with columns, machines or other obstacles.**

Some examples of effective installations are given below.

Example 1

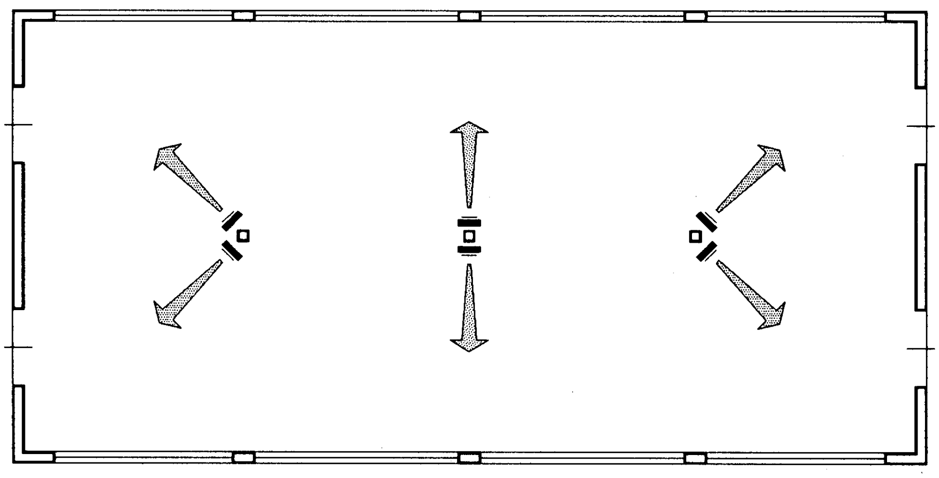
The horizontal unit heaters are arranged in such a way as to provide a continuous movement of air along the external walls. This arrangement should be used in low, regularly shaped areas.



Example 1

Example 2

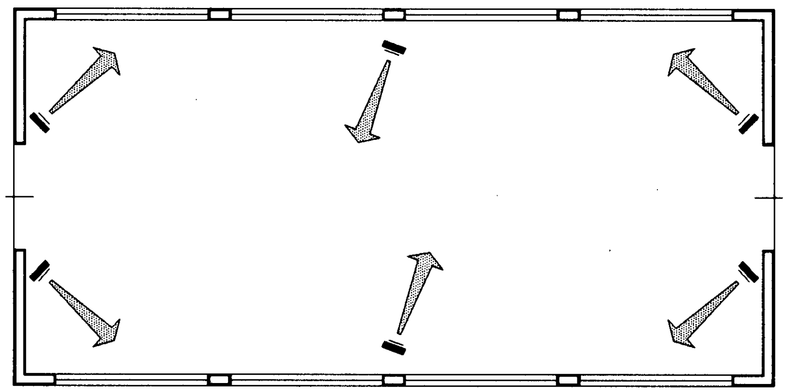
The horizontal unit heaters are installed along the main axis of symmetry of the area and their air outlets are directed towards the external walls. This arrangement should be used to heat particularly wide premises.



Example 2

Example 3

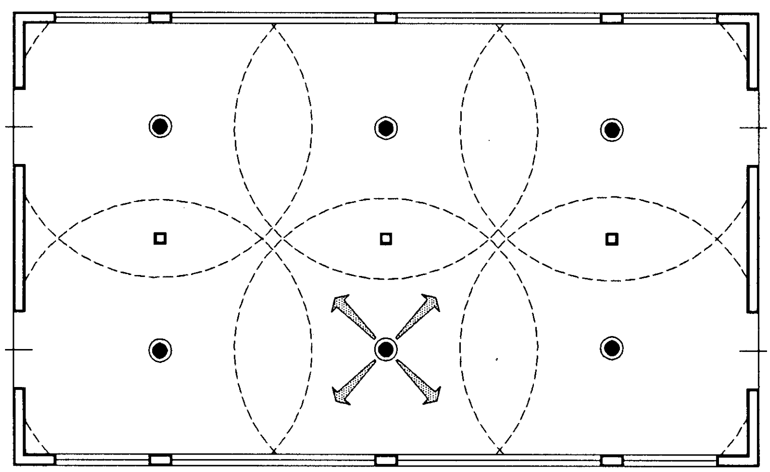
The horizontal unit heaters are installed on the external walls and their air outlets are directed towards the adjacent or opposite walls. This arrangement should be used in premises which are not very wide.



Example 3

Example 4

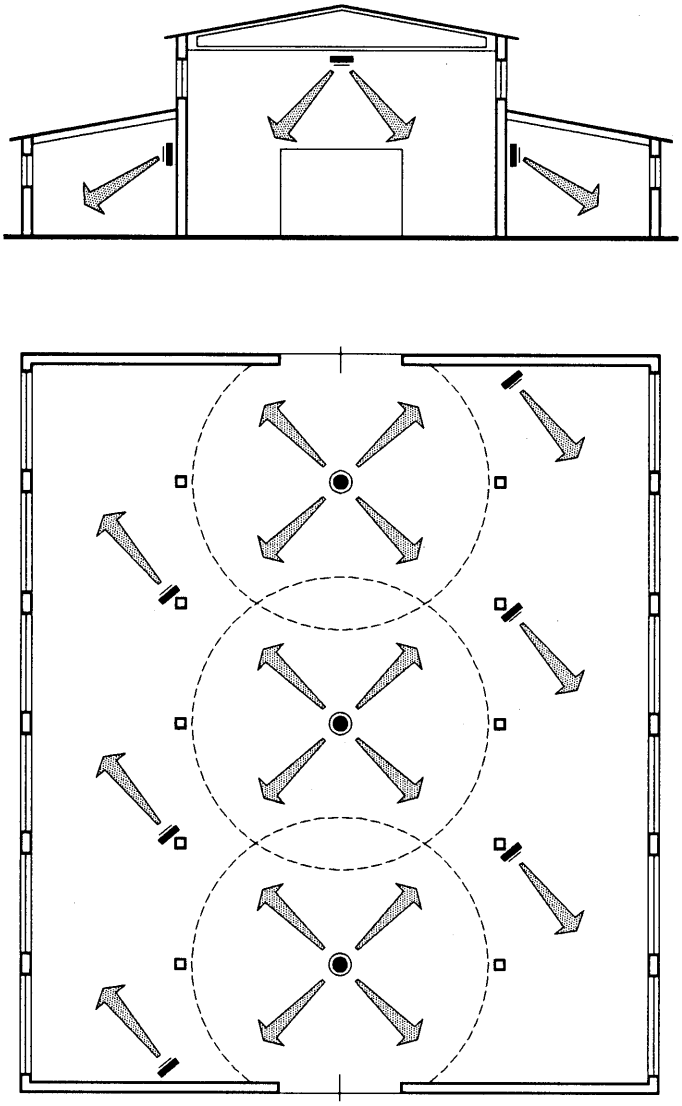
The vertical unit heaters are arranged with their air flows overlapping each other. This arrangement should be used on premises with heights of more than 4 to 5 metres.



Example 4

Example 5

Vertical unit heaters are installed in the central area above the overhead crane whilst horizontal unit heaters are arranged in the side areas with a low ceiling. This arrangement can be used in buildings with areas of varying height.



Example 5

UNIT HEATER AIR OUTLET TEMPERATURE

It is desirable that the temperature of the air at the outlet of unit heaters should be:
- between 40 and 45°C for **horizontal unit heaters** and
- between 30 and 45°C or **vertical unit heaters.**

With these values a satisfactory compromise can be reached between the two following requirements:

- **preventing the air currents** generated by the unit heaters **causing cold sensations** and
- **preventing significant air stratification.**

The temperature of the air at the outlet of the unit heaters is normally given in the manufactures' technical specifications. If not it can be calculated using the following formulae:

$$\boxed{t_{au} = t_{ae} + \frac{(273 + t_{ae}) \cdot Q}{84{,}6 \cdot G}} \qquad (1)$$

$$\boxed{t_{au} = t_{ae} + \frac{(273 + t_{ae}) \cdot Q}{84{,}6 \cdot G - Q}} \qquad (2)$$

where: t_{au} = temperature of the air at the outlet from the unit heater in °C
t_{ae} = temperature of the air at the inlet to the unit heater in °C
Q = heat output provided under standard conditions in kcal/h
G = air flow rate of the fan related to 15°C, m^3/h

Formula (1) applies to horizontal unit heaters, i.e. unit heaters where the fan blows cold air to the heat exchanger.

Formula (2) applies to vertical unit heaters, i.e. unit heaters where the fan sucks in hot air from the heat exchanger.

The temperature of the air at the inlet to the unit heater (t_{ae}) is considered to be:

– **equal to ambient temperature** when there is a total recirculation of internal air;

– **equal to the external temperature** when all the air passing through the unit heater is taken from te outside;

– **equal to the temperature of the mixed air** when the air which passes through the unit heater is taken partly from the inside and partly from the outside.
The temperature of the mixed air can be determined with the following formula:

$$\boxed{t_{am} = \frac{G_{est} \cdot t_{est} + G_{int} \cdot t_{int}}{G_{est} + G_{int}}} \qquad (3)$$

where: t_{am} = temperature of the mixed air in °C
t_{est} = temperature of the external air in °C
t_{int} = temperature of the internal air in °C
G_{est} = flow rate of external air in m^3/h
G_{int} = flow rate of internal air in m^3/h

PERMISSIBLE SOUND LEVEL

The noise level produced by unit heaters, which is normally given in their technical specifications, **should not exceed the ambient permissible sound level.**
This value depends on what the premises are used for and can be estimated based on values recommended in technical literature.

Table 1 gives the sound levels which are normally acceptable in civil and industrial environments (source: ASHRAE).

UNIT HEATER MAINTENANCE

Proper maintenance of unit heaters requires the following:

- **Cleaning the heat exchange battery** with a brush or compressed air. In some cases, particularly in the presence of fatty deposits, the fins need be washed with soap and water taking care not to wet the electrical equipment.
  The frequency cleaning will depend on the environment and on the type of heat exchanger.

- **Checking the tightness of the motor** and casing mountings at least once a year or when irregular noise and vibrations occur during operation.

- **Check the electrical consumption** in the case of the frequent overload cutout.

TAB. 1 - ACCEPTABLE LEVELS OF AMBIENT NOISE

Reading rooms Concert halls Theatres	25 to 30 dB(A)
Apartments Classrooms Offices Meeting rooms Hotel rooms Cinemas Libraries Churches	35 to 42 dB(A)
Entrance halls and corridors of hotels Entrance halls and corridors of hospitals School laboratories Hospital laboratories Halls of public offices Data processing centres Shops Restaurants	40 to 45 dB(A)
Gymnasia kitchens Laundries Entrance halls and corridors of schools Recreation halls Canteens Booking offices Stud farms etc. Garages Large departmental stores Covered swimming pools	45 to 50 dB(A)
Carpenter's shops Light engineering workshops Manufacturing works	50 to 55 dB(A)
Medium to light engineering workshops Printing works	55 to 60 dB(A)
Heavy engineering workshops Engine testing shops Car repair workshops above Textile works Pressing shops	60 dB(A)

NOMINAL HEAT OUTPUT OF A UNIT HEATER

This is the heat output under test conditions. These conditions, taken from various National Standards may be summarized as follows:

- **measuring equipment and instrumentation:** as specified in the above Standard;
- **temperatures of the fluids:**
  – t_e = **80°C,** inlet temperature of the heating fluid,
  – t_u = 70°C, outlet temperature of the heating fluid,
  – t_{ae} = 15°C, temperature of the air at the inlet to the unit heater;
- **maximum rotational speed of the fan;**
- **static pressure difference between the inlet and outlet air:** zero
- **test pressure:** 101,3 kPa.

DESIGN TEMPERATURE OF THE HEATING FLUID

In domestic systems this temperature should be between 60 and 90°C. Higher values however (up to 150, 160°C) can be used in industrial systems.

In all cases, to avoid draughts and a significant stratification of the air, **the design temperature of the heating fluid should be such that the figures specified under the** UNIT HEATER AIR OUTLET TEMPERATURE Section can be met.

EFFECTIVE HEAT OUTPUT OF A UNIT HEATER

This is the heat given up by a unit heater to the external environment under operating conditions. It can be calculated using the following formula:

$$\boxed{Q_{eff} = Q_{nom} \cdot F} \qquad (4)$$

where: Q_{eff} = the effective heat output in W or kcal/h
Q_{nom} = the nominal heat output in W or kcal/h
F = nondimensional correction factor

The correction factor **F** can be calculated using the following equation:

$$\boxed{F = F_t \cdot F_{al} \cdot F_v} \qquad (5)$$

where: F_t = correction factor for the **different temperature**
F_{al} = correction factor for the effect of **altitude**
F_v = correction factor depending on the **velocity of the heating fluid**

These correction factors are calculated below on the basis of the test conditions specified above. In addition the following formula is used to determine the factor F_t:

$$\boxed{Q' = B \cdot (t_m - t_{ae})} \qquad (6)$$

where: Q' = heat output of the unit heater in W or kcal/h
B = characteristics constant of the unit heater in W/°C or kcal/(h·°C)
t_m = mean temperature of the heating fluid in °C
t_{ae} = temperature of the air at the inlet of the unit heater in °C

Note:
Formula (6) **can be considered valid** (with good approximation) **for heating fluid mean temperatures of between 60 and 100°C.**

CORRECTION FACTOR FOR FLUID TEMPERATURE VARIATION

This is the factor used to determine the heat output of a unit heater when the temperature of the air at the inlet (t_{ae}) and the mean temperature of the heating fluid (t_m) differ from the test values. By definition its value is given by the following equation:

$$F_t = \frac{Q'_{eff}}{Q'_{nom}} \qquad (7)$$

Using formula (6) Q'_{eff} e Q'_{nom} may be expressed as follows:

$$Q'_{eff} = B \cdot (t_m - t_{ae}) \qquad (8)$$

$$Q'_{nom} = B \cdot (t_m - t_{ae}) = B \cdot (75 - 15) \qquad (9)$$

The following is obtained:

$$F_t = \frac{Q'_{eff}}{Q'_{nom}} = \frac{B \cdot (t_m - t_{ae})}{B \cdot (75 - 15)} \qquad (10)$$

By simplifying the terms of this equation:

$$\boxed{F_t = \frac{t_m - t_{ae}}{60}} \qquad (11)$$

The values of factor F_t - obtained from formula (11) **- are given in Table 2.**

It may also be necessary to calculate F_t in cases when only the mean temperature (t_m) of the heating fluid or the temperature of the air (t_{ae}) at the inlet of the unit heater varies. In these cases the following formulae can be obtained by substituting in (11) first for t_{ae} and then for t_m the corresponding test values:

– (11.1) which is valid for **variabile** t_m with t_{ae} = 15°C (test temperature)
– (11.2) which is valid for **variabile** t_{ae} with t_m = 75°C (test temperature).

$$\boxed{F_t\,(t_{ae} = 15°C) = \frac{t_m - 15}{60}} \qquad (11.1)$$

$$\boxed{F_t\,(t_m = 75°C) = \frac{75 - t_{ae}}{60}} \qquad (11.2)$$

TAB. 2 - CORRECTION FACTOR F_t FOR UNIT HEATERS

conditions for nominal heat output measurement

t_m = 75°C and t_{ae} = 15°C

mean temp. of heating fluid	temperature of the air 0°C	3°C	6°C	9°C	12°C	15°C	18°C	21°C	24°C
60°C	1,00	0,95	0,90	0,85	0,80	0,75	0,70	0,65	0,60
62°C	1,03	0,98	0,93	0,88	0,83	0,78	0,73	0,68	0,63
64°C	1,07	1,02	0,97	0,92	0,87	0,82	0,77	0,72	0,67
66°C	1,10	1,05	1,00	0,85	0,90	0,85	0,80	0,75	0,70
68°C	1,13	1,08	1,03	0,98	0,93	0,88	0,83	0,78	0,73
70°C	1,17	1,12	1,07	1,02	0,97	0,92	0,87	0,82	0,77
72°C	1,20	1,15	1,10	1,05	1,00	0,95	0,90	0,85	0,80
74°C	1,23	1,18	1,13	1,08	1,03	0,98	0,93	0,88	0,83
75°C	1,25	1,20	1,15	1,10	1,05	1,00	0,95	0,90	0,85
76°C	1,27	1,22	1,17	1,12	1,07	1,02	0,97	0,92	0,87
78°C	1,30	1,25	1,20	1,15	1,10	1,05	1,00	0,95	0,90
80°C	1,33	1,28	1,23	1,18	1,13	1,08	1,03	0,98	0,93
82°C	1,37	1,32	1,27	1,22	1,17	1,12	1,07	1,02	0,97
84°C	1,40	1,35	1,30	1,25	1,20	1,15	1,10	1,05	1,00
86°C	1,43	1,38	1,33	1,28	1,23	1,18	1,13	1,08	1,03
88°C	1,47	1,42	1,37	1,32	1,27	1,22	1,17	1,12	1,07
90°C	1,50	1,45	1,40	1,35	1,30	1,25	1,20	1,15	1,10
92°C	1,53	1,48	1,43	1,35	1,33	1,28	1,23	1,18	1,13
94°C	1,57	1,52	1,47	1,42	1,37	1,32	1,27	1,22	1,17
96°C	1,60	1,55	1,50	1,45	1,40	1,35	1,30	1,25	1,20
98°C	1,63	1,58	1,53	1,48	1,43	1,38	1,33	1,28	1,23
100°C	1,67	1,62	1,57	1,52	1,47	1,42	1,37	1,32	1,27

CORRECTION FACTOR FOR THE EFFECT OF ALTITUDE

This is he factor used to calculate the heat output of a unit heater when it is not installed at sea level. It must be borne in mind that the density of the air and thus its capacity for conveying heat is reduced progressively as the altitude increases. This factor can be calculated using the following formula:

$$\boxed{F_{al} = \frac{P_o}{1{,}5 \cdot P_o - 0{,}5 \cdot P}} \qquad (12)$$

where: P_o = atmospheric pressure at sea level in kPa
P = atmospheric pressure at the installation site in kPa

P_o is equal to 101,3 kPa, whilst the value of **P** can be calcuated usign the equation:

$$\boxed{P = 101{,}3 - 0{,}0113 \cdot H} \qquad (13)$$

where: **H** = height above sea level in m

The values of the factor F_{al} - obtained from formula (12) - **are given in Table 3.**

TAB. 3 - CORRECTION FACTOR F_{al} FOR UNIT HEATERS

altitude	Atmospheric pressure	F_{al}
750 m	92,8 kPa	0,96
1.000 m	90,0 kPa	0,95
1.250 m	87,2 kPa	0,93
1.500 m	84,4 kPa	0,92
1.750 m	81,5 kPa	0,91

CORRECTION FACTOR FOR THE VELOCITY OF THE FLUID

This is the factor used to calculate the heat output of a unit heater when the velocity of the fluid differs from the test value. Its value depends on the characteristics of the unit heater and in practice can only be determined experimentally.

Normally the manufacturers do no more than indicate the minimum speed (flow rate) **required so that the correction effect of this factor can be considered zero.**

<u>**Example:**</u>

To determine the nominal heat output of a unit heater which is required to give out 8000 kcal/h into the surrounding atmosphere. The following values are used:
- t_{ae} = 18°C, temperature of the air at the inlet of the unit heater,
- G_{eff} = 1.000 l/h, effective flow rate of the unit heater,
- t_e = 70°C, design temperature of the heating fluid,
- H = 1.000 m, above sea level,
- F_v = 1, correction factor for the velocity of the fluid.

<u>**Solution:**</u>

The required heat output can be calculated usign formulae (4) and (5): $Q_{nom} = \dfrac{Q_{eff}}{F_t \cdot F_{al} \cdot F_v}$

Determining F_t (correction factor for the different temperature of the fluid).

Its value can be determined using the formula (11) or Table 2 and depends on the following values:
– t_m (mean temperature of the heating fluid),
– t_{ae} (temperature of the air at the inlet of the unit heater).

To determine t_m, you can first of call calculate t_u (outlet temperature of the heating fluid) and then take the average between this temperature and t_e (inlet temperature of the heating fluid).

$$t_u = t_e - \frac{Q_{eff}}{G_{eff}} = 70 - \frac{8.000}{1.000} = 62°C \qquad t_m = \frac{t_e + t_u}{2} = \frac{70 + 62}{2} = 66°C$$

Having taken account of the temperatures t_m and t_{ae} using the formula or the above mentioned table: $F_t = 0.80$.

— **Determining F_{al}** (correction factor for altitude)

Its value can be determined using formula (12) or Table 3 and depends on the height of the installation (H) above sea level. In the case under examination (where H = 1.000 m) $F_{al} = 0.95$.

Having calculated the values of F_t and F_{al} the required nominal output (Q_{nom}) can therefore be calculated as follows:

$$Q_{nom} = \frac{Q_{eff}}{F_t \cdot F_{al} \cdot F_v} = \frac{8.000}{0{,}80 \cdot 0{,}95 \cdot 1{,}00} = 10.526 \text{ kcal/h.}$$

RADIATORS

- CLASSIFICATION
  - CAST IRON RADIATORS
  - STEEL RADIATORS
  - ALUMINIUM RADIATORS
- INSTALLATION OF THE RADIATORS
- NOMINAL HEAT OUTPUT OF A RADIATOR
- DESIGN TEMPERATURE OF THE HEATING FLUID
- EFFECTIVE HEAT OUTPUT OF A RADIATOR
  - CORRECTION FACTOR FOR THE DIFFERENT TEMPERATURE OF THE FLUIDS
  - CORRECTION FACTOR FOR THE EFFECT OF THE ALTITUDE
  - CORRECTION FACTOR FOR THE PROTECTION OF THE RADIATOR
  - CORRECTION FACTOR IN RELATION TO THE RADIATOR CONNECTIONS
  - CORRECTION FACTOR FOR THE EFFECT OF THE PAINTING

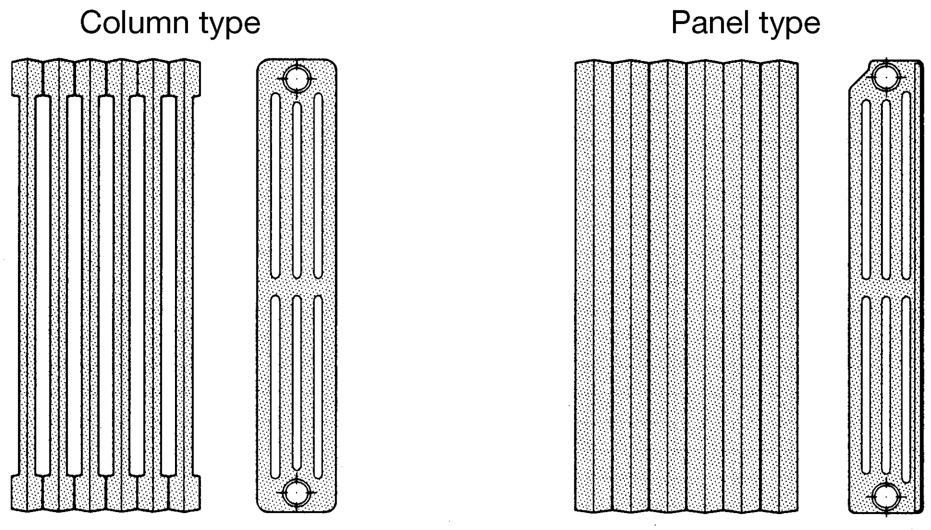
Radiators are heat emitters (made up of elements, panels, tubes or blades) **which give out heat by natural convection and radiation.**

CLASSIFICATION

Radiators can be classified according to type, i.e. **cast iron, radiators, steel radiators** and **aluminium radiators** (which pure aluminium or aluminium alloy) according to the material they are made of.

CAST IRON RADIATORS

They are made of cast elements and assembled with nipples. In addition to the conventional **column type,** the **panel type** was introduced in the 1970's. These have a wide radiant front surface and a thin section at the rear to limit passive heat exchange with the walls.



Cast iron radiators

Positive features of cast iron radiators:

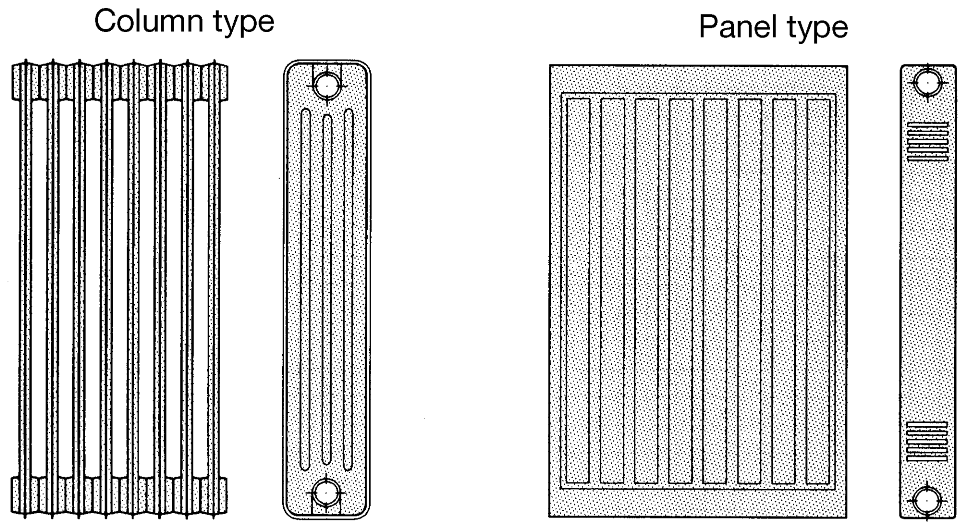
- **They are resistant to corrosion;**
- **They do not produce expansion and contraction noises;**
- **They can be manufactured in a wide variety of lenghths as they are assembled from elements.**

Negative features:

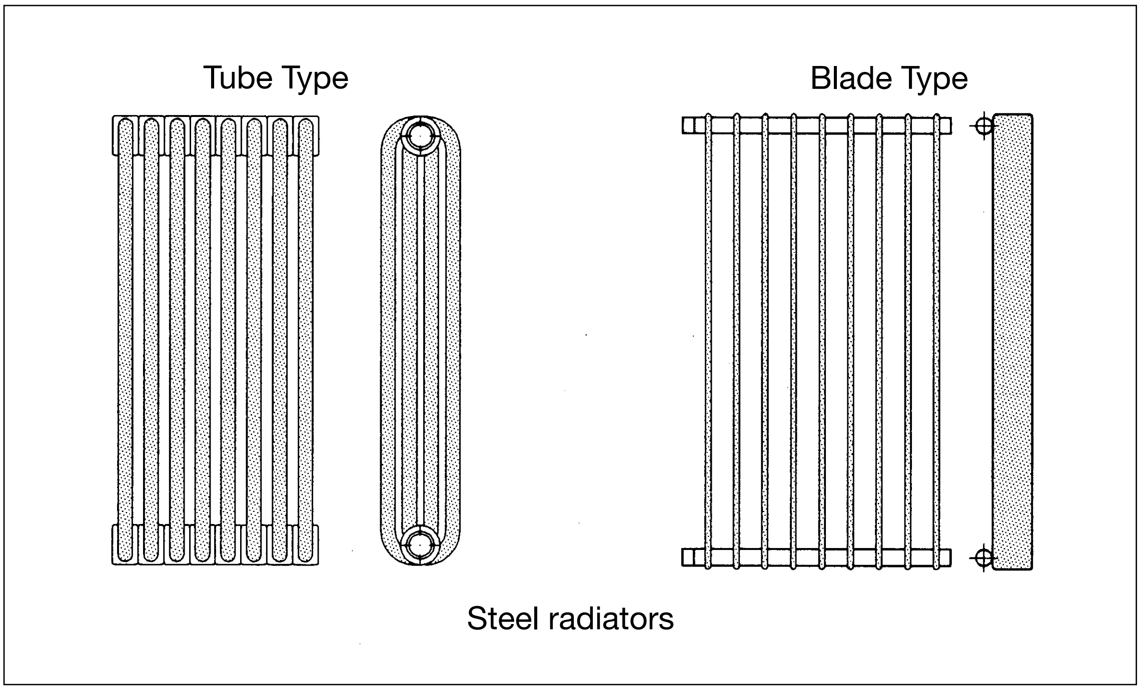
- **More expensive,** particularly in relation to panel and column steel radiators;
- **High weight** which makes istallation more difficult;
- **Brittleness** which can be the cause of fractures during assembly or installation;
- **High thermal inertia** which can make ambient temperature regulation less effective.

STELL RADIATORS

These radiators are constructed by welding stamped plates or tubes. They can be of the panel, column tube or blade type.



Steel radiators

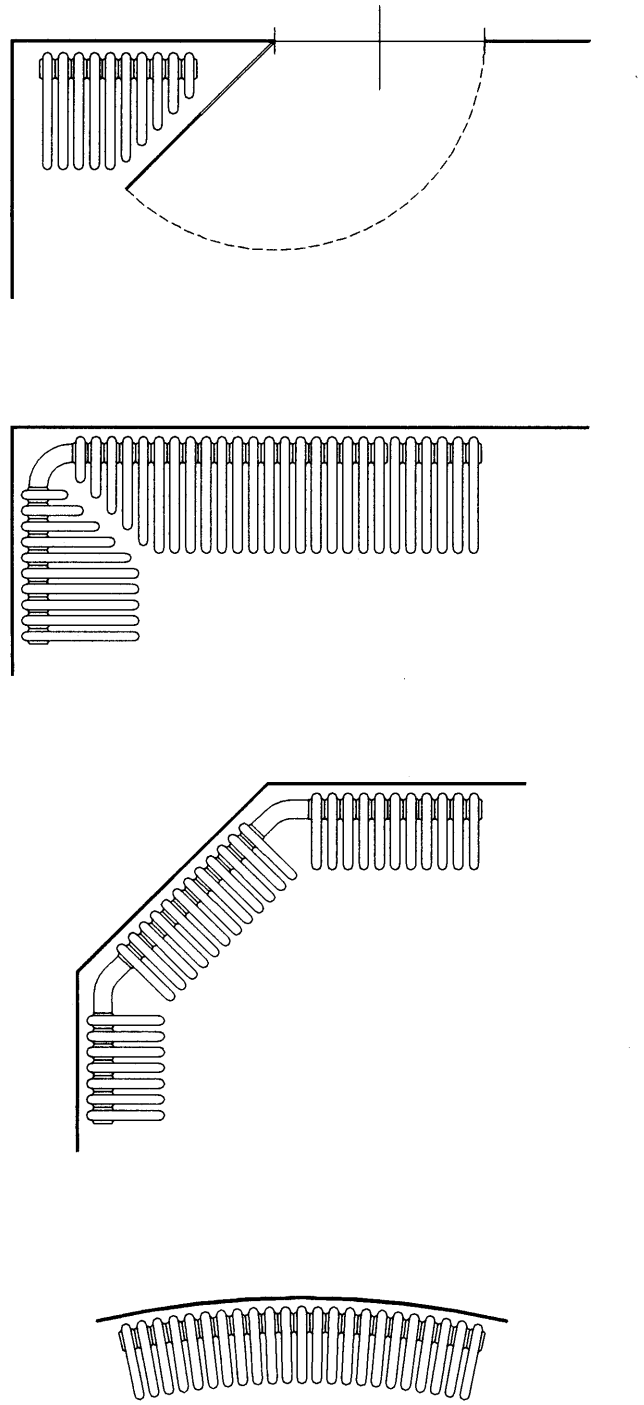


Positive features of steel radiators:

- **Low cost:** they are the most economical panel and column type radiators;
- **Low weight:** they weigh approximately 65 to 70% less than cast iron radiators with the same heat output;
- They blend in easily with their surroundings: the wide range of range of types and geometrical shapes available **can easily be integrated into their environment;**
- **Low thermal inertia on the panel types.**

Negative features:

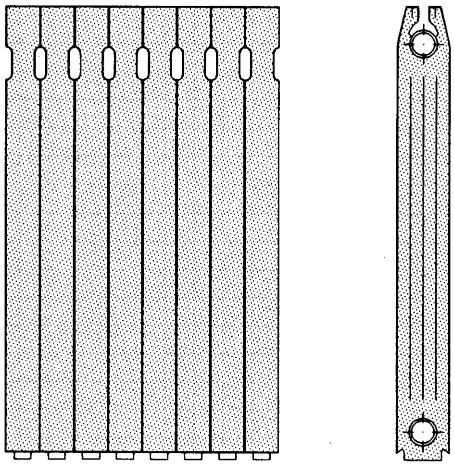
- **High thermal inertia of the column and tube types** (i.e. on high water content types). This characteristic can make ambient temperature regulation less effective;
- **The welded panel, blade and column types not assembled from sections;**
- **Possible corrosion:** these radiators are easily affected by external corrosion unless they have suitable surface coatings.



Possible arrangements with tube radiators

ALLUMINIUM RADIATOR

These radiators are assembled from components made by extrusion or pressure die casting using nipples.



Aluminium radiator

Positive features of aluminium radiators:

- **Relatively low cost.** they cost considerably less than cast iron radiators;
- **Lightweight costruction:** they weight 70 to 75% less than cast iron radiators with the same heat output;
- **They are always assembled from sections;**
- **Limited thermal inertia.**

Negative features:

- **Possible internal corrosion:** the presence of alkalis in the water promotes aluminium corrosion. For this reason it is advisable to avoid excessive water softening and the use of chemical inhibitors.

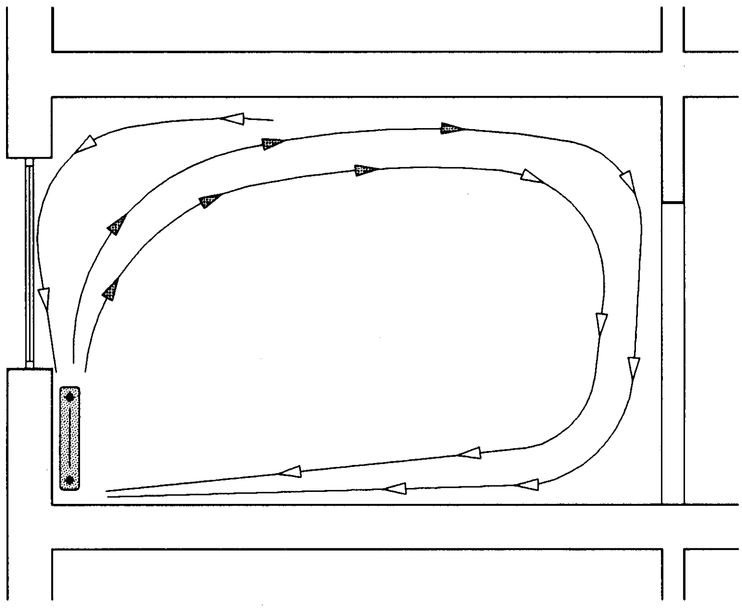
INSTALLING RADIATORS

It is advisable to install the radiators **under a window or along the external walls** as in this way:

- **The cold air currents** which form in relation to these surfaces **can be overcome more effectively;**

- **The physiological comfort conditions are improved** by limiting the radiation of the human body towards the cold zones;

- **The formation of condensation on internal surfaces is prevented or reduced** in the area surrounding the heat emitter.



Example of convective movements caused by a radiator

The following distances must be available in order that the radiators can be correctly installed:

- **distance from the floor** = 10 to 12 cm;

- **distance from the walls** = 4 to 5 cm;

- **It is advisable to stipulate "appropriate distances" of not less than 10 cm for projections above or at the side the radiator** (brackets, recesses, shelves etc.).

NOMINAL HEAT OUTPUT OF A RADIATOR

This is the heat output under test conditions. These conditions, taken from various National Standards, generally may be summarised as follows:

- **measuring equipment and instrumentation:** as specified in the Standard;
- **temperatures of the fluids:**
  – t_e = 85°C, inlet temperature of the heating fluid,
  – t_u = 75°C, outlet temperature of the heating fluid,
  – t_a = 20°C, temperature of the air;
- **installation:**
  – distance from walls = 5 cm,
  – distance from floor = 10 ÷ 12 cm;
- **radiator flow and return - top and bottom opposite ends**
- **test pressure:** 101,3 kPa.

DESIGN TEMPERATURE OF THE HEATING FLUID

This temperature is normally best kept between 65 and 75°C. Higer temperatures are not advisable as they can:

- **initiate strong convection currents** and thus contribute to the formation of areas of hot air at the ceiling and cold air at the floor;
- **cause a "baking" of atmospheric dust thereby** causing irritation of the respiratory system as well as staining of the walls behind and above the radiator.

On the other excessively low design temperatures result in a significant increase of the cost of the system and the space occupied by the radiators.

EFFECTIVE HEAT OUTPUT RADIATOR

This is the heat output of a radiator (or one of its elements) **to atmosphere under operating conditions.** It can be calculated using the following formula:

$$\boxed{Q_{eff} = Q_{nom} \cdot F} \qquad (1)$$

where: Q_{eff} = effective heat output in W or kcal/h
Q_{nom} = nominal heat output in W or kcal/h
F = nondimensional correction factor

The overall correction factor **F** can be calculated using the equation:

$$\boxed{F = F_t \cdot F_{al} \cdot F_{pr} \cdot F_{at} \cdot F_{vr}} \qquad (2)$$

where: F_t = correction factor for the **different temperature of the fluid**
F_{al} = correction factor for the effect of the **altitude**
F_{pr} = correction factor for **enclosing** the radiator
F_{at} = correction factor for the radiator connections orientation
F_{vr} = correction factor for the effect of **painting**

These correction factors are calculated below based on the test conditions defined above. In addition the following formula is used for determining the factor F_t:

$$\boxed{Q' = B \cdot (t_m - t_a)^{1,3}} \qquad (3)$$

where: Q' = the heat output of the radiator in W or kcal/h
B = the characteristic constant of the radiator in W/°C ^{1,3} or kcal/(h·°C ^{1,3})
t_m = mean temperature of the heating fluid in °C
t_a = ambient temperature °C

Note:
Formula (3) **can be considered applicable** (with acceptable accuracy) **to mean temperatures of the heating fluid from 40 and 100°C.**

CORRECTION FACTOR FOR NON STANDARD FLUID TEMPERATURES

This is the factor used to determine the heat output of a radiator when the ambient temperature (t_a) and the mean temperature of the heating fluid (t_m) differ from the test temperatures. Its value is given by the ratio:

$$F_t = \frac{Q'_{eff}}{Q'_{nom}} \tag{4}$$

Using formula (3) Q'_{eff} and Q'_{nom} may be expressed as floows:

$$Q'_{eff} = B \cdot (t_m - t_a)^{1,3} \tag{5}$$

$$Q'_{nom} = B \cdot (t_m - t_a)^{1,3} = B \cdot (80 - 20)^{1,3} \tag{6}$$

The following is therefore obtained:

$$F_t = \frac{Q'_{eff}}{Q'_{nom}} = \frac{B \cdot (t_m - t_a)^{1,3}}{B \cdot (80 - 20)^{1,3}} \tag{7}$$

By semplifying this equation the following is obtained:

$$\boxed{F_t = \left(\frac{t_m - t_a}{60}\right)^{1,3}} \tag{8}$$

The values of the factor F_t obtained from formula (8) **are given in Table 1.**

It may also be helpful to express F_t for cases whereonly the mean temperature (t_m) of the heating fluid or the temperature (t_a) of the ambient air varies. For this purpose the following formulae can be obtained by replacing first of all "t_a"and then "t_m" with the relative test values in (8):

– (8.1) valid for **variable** t_m and t_a = 20°C (test temperature)
– (8.2) valid for **variable** t_a and t_m = 80°C (test temperature).

$$F_t\ (t_a = 20°C) = \left(\frac{t_m - 20}{60}\right)^{1,3} \qquad (8.1)$$

$$F_t\ (t_m = 80°C) = \left(\frac{80 - t_a}{60}\right)^{1,3} \qquad (8.2)$$

Note:

A European standard which specifies a mean temperature of the heating fluid of 70°C is at present in the process of being approved for the thermal test ing of radiators.

On the basis of this test condition **the value of the correction factor Ft** can be determined using:

$$F_t = \left(\frac{t_m - t_a}{50}\right)^{1,3} \qquad (9)$$

The values of factor F_t obtained from formula (9) **are given in Table 2.**

TAB. 1 - CORRECTION FACTOR F_t FOR RADIATORS

conditions for measuring nominal heat output

t_m = 80°C e t_a = 20°C

Mean temp. of heating fluid	temperature of the air								
	10°C	12°C	14°C	16°C	18°C	20°C	22°C	24°C	26°C
40°C	0,41	0,37	0,34	0,30	0,27	0,24	0,21	0,18	0,15
42°C	0,44	0,41	0,37	0,34	0,30	0,27	0,24	0,21	0,18
44°C	0,48	0,44	0,41	0,37	0,34	0,30	0,27	0,24	0,21
46°C	0,51	0,48	0,44	0,41	0,37	0,34	0,30	0,27	0,24
48°C	0,55	0,51	0,48	0,44	0,41	0,37	0,34	0,30	0,27
50°C	0,59	0,55	0,51	0,48	0,44	0,41	0,37	0,34	0,30
52°C	0,63	0,59	0,55	0,51	0,48	0,44	0,41	0,37	0,34
54°C	0,67	0,63	0,59	0,55	0,51	0,48	0,44	0,41	0,37
56°C	0,71	0,67	0,63	0,59	0,55	0,51	0,48	0,44	0,41
58°C	0,75	0,71	0,67	0,63	0,59	0,55	0,51	0,48	0,44
60°C	0,79	0,75	0,71	0,67	0,63	0,59	0,55	0,51	0,48
62°C	0,83	0,79	0,75	0,71	0,67	0,63	0,59	0,55	0,51
64°C	0,87	0,83	0,79	0,75	0,71	0,67	0,63	0,59	0,55
66°C	0,91	0,87	0,83	0,79	0,75	0,71	0,67	0,63	0,59
68°C	0,96	0,91	0,87	0,83	0,79	0,75	0,71	0,67	0,63
70°C	1,00	0,96	0,91	0,87	0,83	0,79	0,75	0,71	0,67
72°C	1,04	1,00	0,96	0,91	0,87	0,83	0,79	0,75	0,71
74°C	1,09	1,04	1,00	0,96	0,91	0,87	0,83	0,79	0,75
76°C	1,13	1,09	1,04	1,00	0,96	0,91	0,87	0,83	0,79
78°C	1,18	1,13	1,09	1,04	1,00	0,96	0,91	0,87	0,83
80°C	1,22	1,18	1,13	1,09	1,04	1,00	0,96	0,91	0,87
82°C	1,27	1,22	1,18	1,13	1,09	1,04	1,00	0,96	0,91
84°C	1,31	1,27	1,22	1,18	1,13	1,09	1,04	1,00	0,96
86°C	1,36	1,31	1,27	1,22	1,18	1,13	1,09	1,04	1,00
88°C	1,41	1,36	1,31	1,27	1,22	1,18	1,13	1,09	1,04
90°C	1,45	1,41	1,36	1,31	1,27	1,22	1,18	1,13	1,09
92°C	1,50	1,45	1,41	1,36	1,31	1,27	1,22	1,18	1,13
94°C	1,55	1,50	1,45	1,41	1,36	1,31	1,27	1,22	1,18
96°C	1,60	1,55	1,50	1,45	1,41	1,36	1,31	1,27	1,22
98°C	1,65	1,60	1,55	1,50	1,45	1,41	1,36	1,31	1,27
100°C	1,69	1,65	1,60	1,55	1,50	1,45	1,41	1,36	1,31

TAB. 2 - CORRECTIVE FACTOR F_t FOR RADIATORS

conditions for measuring nominal heat output
t_m = 70°C and t_a = 20°C

Mean temp. of heating fluid	temperature of the air								
	10°C	12°C	14°C	16°C	18°C	20°C	22°C	24°C	26°C
40°C	0,51	0,47	0,43	0,39	0,34	0,30	0,26	0,23	0,19
42°C	0,56	0,51	0,47	0,43	0,39	0,34	0,30	0,26	0,23
44°C	0,61	0,56	0,51	0,47	0,43	0,39	0,34	0,30	0,26
46°C	0,65	0,61	0,56	0,51	0,47	0,43	0,39	0,34	0,30
48°C	0,70	0,65	0,61	0,56	0,51	0,47	0,43	0,39	0,34
50°C	0,75	0,70	0,65	0,61	0,56	0,51	0,47	0,43	0,39
52°C	0,80	0,75	0,70	0,65	0,61	0,56	0,51	0,47	0,43
54°C	0,85	0,80	0,75	0,70	0,65	0,61	0,56	0,51	0,47
56°C	0,90	0,85	0,80	0,75	0,70	0,65	0,61	0,56	0,51
58°C	0,95	0,90	0,85	0,80	0,75	0,70	0,65	0,61	0,56
60°C	1,00	0,95	0,90	0,85	0,80	0,75	0,70	0,65	0,61
62°C	1,05	1,00	0,95	0,90	0,85	0,80	0,75	0,70	0,65
64°C	1,11	1,05	1,00	0,95	0,90	0,85	0,80	0,75	0,70
66°C	1,16	1,11	1,05	1,00	0,95	0,90	0,85	0,80	0,75
68°C	1,21	1,16	1,11	1,05	1,00	0,95	0,90	0,85	0,80
70°C	1,27	1,21	1,16	1,11	1,05	1,00	0,95	0,90	0,85
72°C	1,32	1,27	1,21	1,16	1,11	1,05	1,00	0,95	0,90
74°C	1,38	1,32	1,27	1,21	1,16	1,11	1,05	1,00	0,95
76°C	1,43	1,38	1,32	1,27	1,21	1,16	1,11	1,05	1,00
78°C	1,49	1,43	1,38	1,32	1,27	1,21	1,16	1,11	1,05
80°C	1,55	1,49	1,43	1,38	1,32	1,27	1,21	1,16	1,11
82°C	1,61	1,55	1,49	1,43	1,38	1,32	1,27	1,21	1,16
84°C	1,66	1,61	1,55	1,49	1,43	1,38	1,32	1,27	1,21
86°C	1,72	1,66	1,61	1,55	1,49	1,43	1,38	1,32	1,27
88°C	1,78	1,72	1,66	1,61	1,55	1,49	1,43	1,38	1,32
90°C	1,84	1,78	1,72	1,66	1,61	1,55	1,49	1,43	1,38
92°C	1,90	1,84	1,78	1,72	1,66	1,61	1,55	1,49	1,43
94°C	1,96	1,90	1,84	1,78	1,72	1,66	1,61	1,55	1,49
96°C	2,02	1,96	1,90	1,84	1,78	1,72	1,66	1,61	1,55
98°C	2,09	2,02	1,96	1,90	1,84	1,78	1,72	1,66	1,61
100°C	2,15	2,09	2,02	1,96	1,90	1,84	1,78	1,72	1,66

CORRECTION FACTOR FOR ALTITUDE

This is the factor used to determine the heat output of a radiator when not installed at sea level. It must be remembered that the density of the air, and thus its capacity for conveying heat, is progressively reduced as the altitude increases. This factor can be calculated using the following formula:

$$\boxed{F_{al} = \frac{P_o}{1{,}3 \cdot P_o - 0{,}3 \cdot P}} \qquad (10)$$

where: P_o = atmospheric pressure at sea level in kPa
P = atmospheric pressure at site in kPa

The value of P_o is **101,3** kPa and the value of **P** can be calculated using the following equation:

$$\boxed{P = 101{,}3 - 0{,}0113 \cdot H} \qquad (11)$$

where: **H** = height above sea level in m

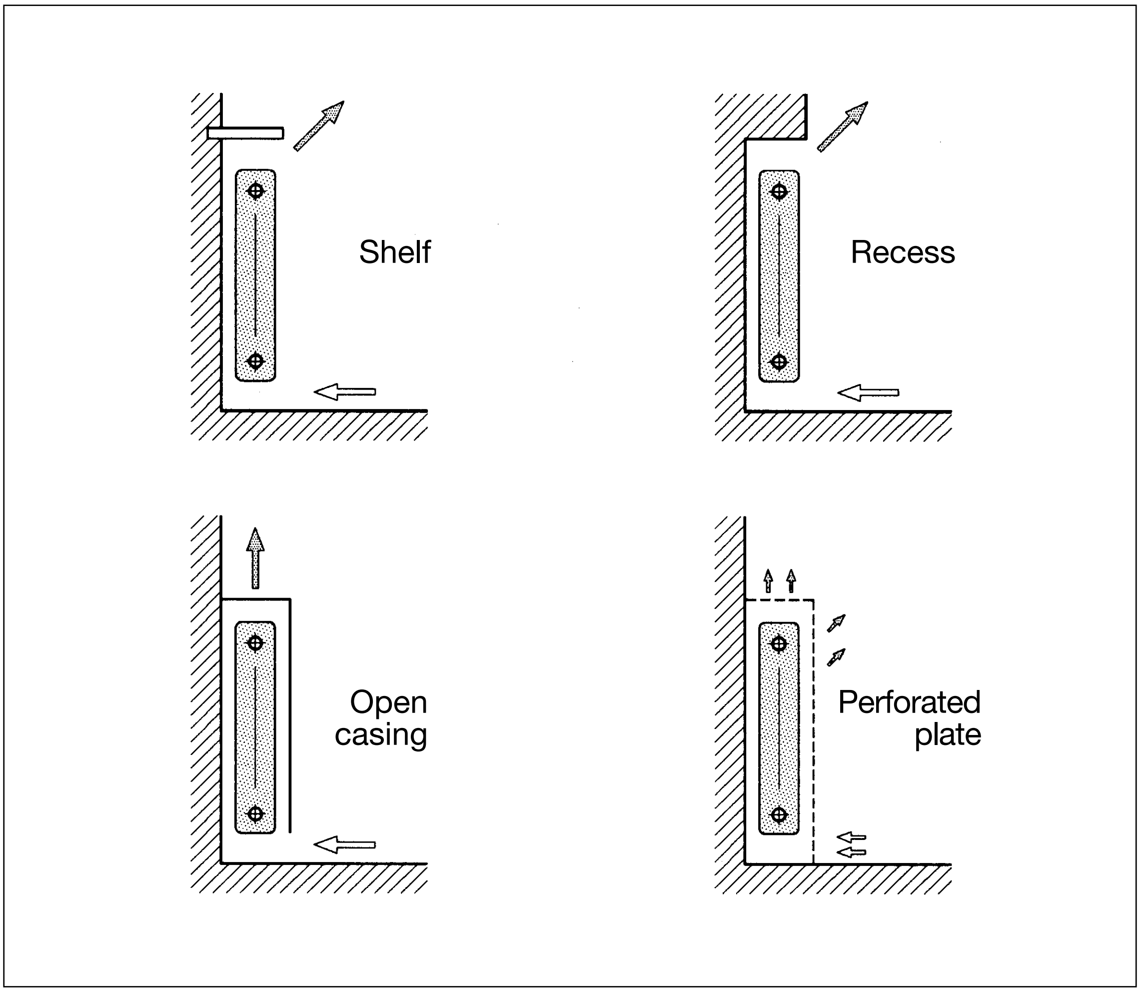
The values of factor F_{al} obtained from the formula (10) **are given in Table 3.**

TAB. 3 - CORRECTION FACTOR F_{al} FOR RADIATORS

altitude	Atmospheric pressure	F_{al}
750 m	92,8 kPa	0,98
1.000 m	90,0 kPa	0,97
1.250 m	87,2 kPa	0,96
1.500 m	84,4 kPa	0,95
1.750 m	81,5 kPa	0,94

CORRECTION FACTOR FOR THE PROTECTION OF THE RADIATOR

This is the factor used to determine the heat output of a radiator installed in recesses under a shelf or in a cabinet. Its value takes into account the fact that enclosure limits and sometimes considerably reduces the heat transfer between the radiator and the surrounding atmosphere.



The following average values can be considered applicable:

F_{pr} = 0,95 ÷ 0,97 for installation under a shelf.

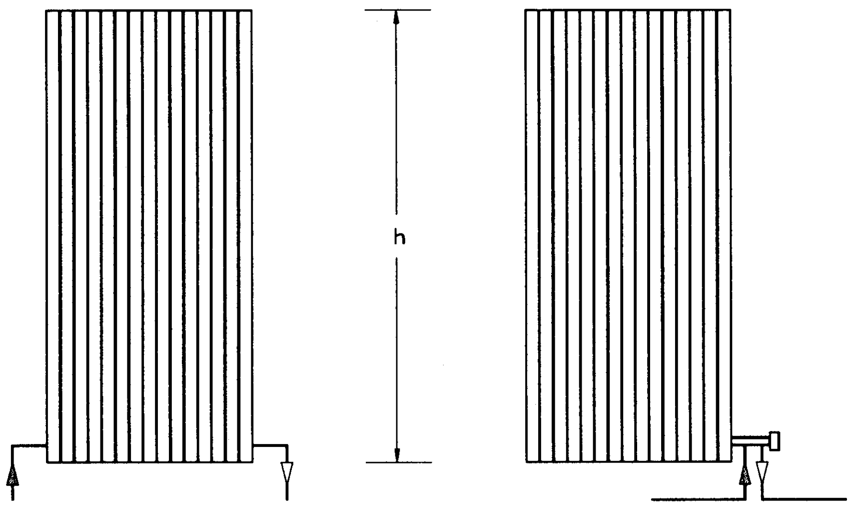
F_{pr} = 0,92 ÷ 0,94 for installation in recess.

F_{pr} = 0,75 ÷ 0,85 for installation with perforated plate.

F_{pr} = 0,95 ÷ 1,00 for installation with open casing.

CORRECTION FACTOR FOR THE RADIATOR CONNECTION ORIENTATION

This is the factor used to determine the output of a radiator which is not installed according to the test conditions: i.e. not installed with the inlet at the top and the outlet at the bottom on the opposite side (top/bottom opposite side = TBOE). Practically speaking it is only worth considering the cases of radiators with both connections at the bottom.



The average value for factor F_{at}, either for bottom connections located on the same side (bottom same side = BSE), or for bottom connections on opposite sides, (bottom opposite ends = BOE) **can be taken as:**

F_{at} = 1,00 for **h** less than 1,20 m.

F_{at} = 0,97 ÷ 0,95 for **h** of between 1,20 and 1,80 m.

F_{at} = 0,95 ÷ 0,90 for **h** greater than 1,80 m.

CORRECTION FACTOR FOR THE EFFECT OF THE PAINTING

This is the factor which used to determine the heat output of a radiator when (after the nominal output test) **it is painted.** Its value takes account of the fact that paint can significantly reduce the thermal energy emitted by radiation.

The following average values can be considered applicable:

F_{vr} = 1,00 for oil paints

F_{vr} = 0,85 ÷ 0,90 for paints with an aluminium or bronze base.

C O N V E C T O R S

- CLASSIFICATION
  - CONVECTORS WITH SINGLE FINS
  - CONVECTORS WITH FINNED CHANNELS
  - CABINET CONVECTORS
  - BASEBOARD CONVECTORS
- NOMINALE HEAT OUTPUT OF A CONVECTOR
- DESIGN TEMPERATURE OF THE HEATING FLUID
- EFFECTIVE HEAT OUTPUT ON A CONVECTOR
  - CORRECTION FACTOR FOR DIFFERENT FLUID TEMPERATURES
  - CORRECTION FACTOR FOR THE EFFECT OF THE ALTITUDE
  - CORRECTION FACTOR FOR THE TYPE OF INSTALLATION
  - CORRECTION FACTOR FOR THE VELOCITY OF THE FLUID

Convectors are heat emitters which give off heat mainly by convection. They are constructed with finned heat exchangers and operate under natural “draught” conditions.

These heat emitters have the following advantages over radiators:

- **They are lighter and less expensive for the same output;**
- **They have a lower thermal inertia;**
- **They provide solutions to the specific installation problems.**

On the other hand they have the following disadvantages:

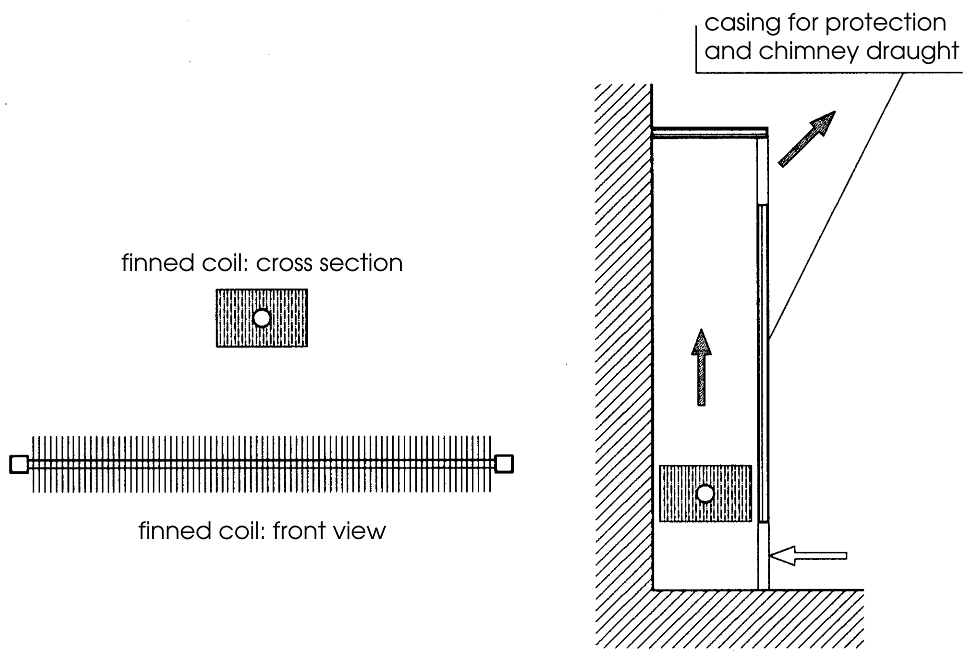
- **They are difficult to clean** so that they should not be used in dusty locations or where satisfactory cleaning is not possible;
- **They are not assembled from elements;**
- **They cannot be used for automatic air temperature control** as their heat output curve has a “bend” (i.e. a significant variation in gradient) for fluid temperatures between 45 and 50°C.

CLASSIFICATION

Convectors can be classified on the basis of their structural characteristics, i.e. those with **single finned tube,** those with **finned channels, cabinet convectors** and **baseboard convectors.**

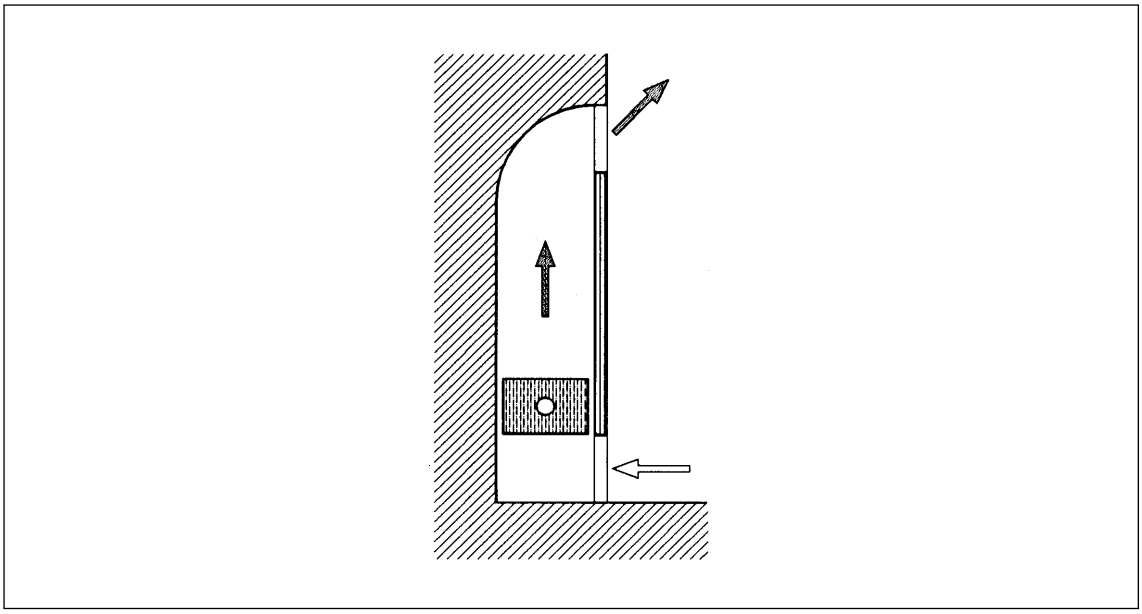
CONVECTORS WITH SINGLE FINS

They are constructed from tubes with flat fins mounted in a casing to form a chimney draught.

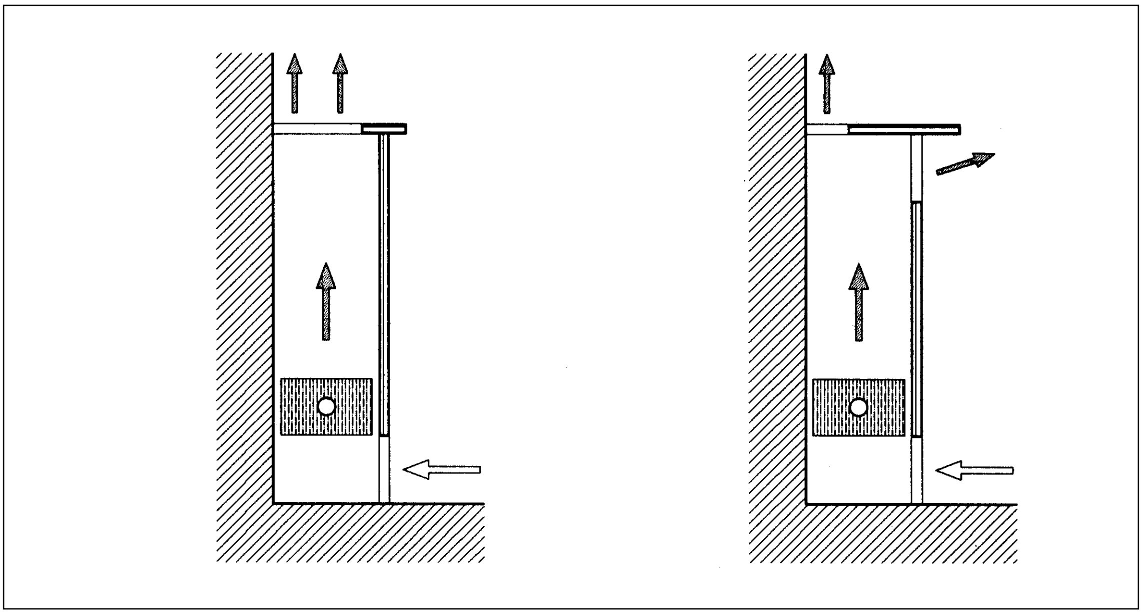


Convector with single fins

The convection path can be formed by using recesses and inlets in walls.

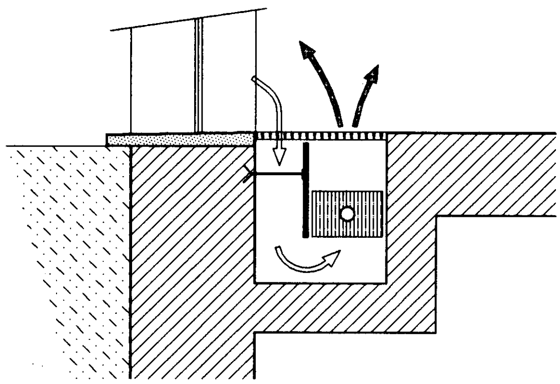


They can also be formed by using suitable linings and shields.

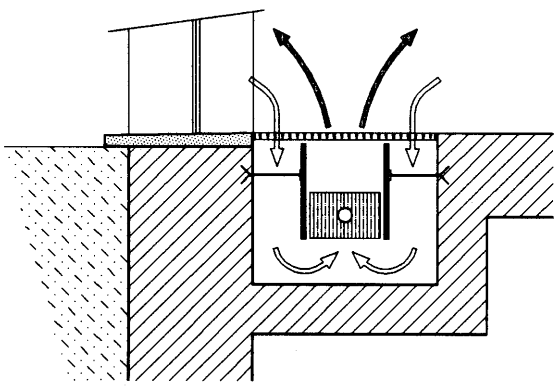


Convectors with single finned tube are also installed below floor level particularly to protect large glazed surfaces from condensation. For this purpose it is advisable to install the heat exchangers as follows:

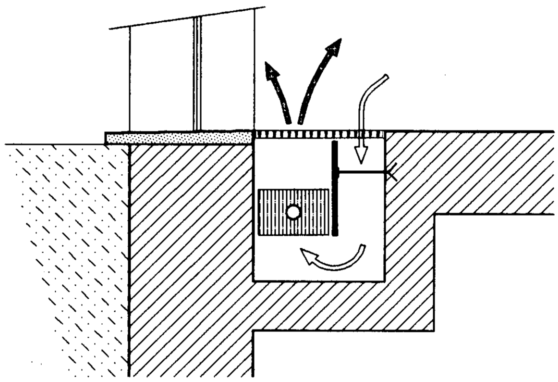
a) **the creation of a chimney on the outside** of the installation if the heat losses of the glazed surface are greater then those of the opposite walls;



b) **the siting of the heat exchangers between two chimneys** if the heat losses of the glazed surface are almost equal to those of the opposite walls;

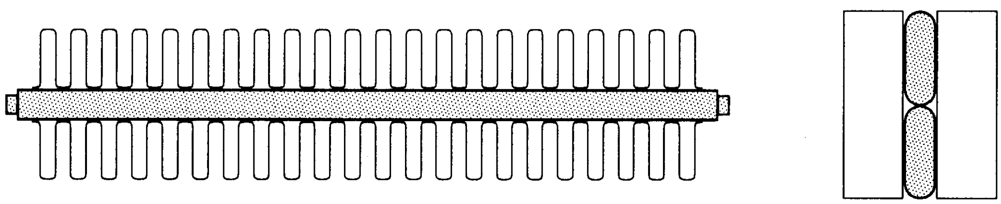


c) **the construction of chimney towards the inside of the installation** if the heat losses of the glazed surface are less than those of the opposite walls.



CONVECTORS WITH FINNED CHANNELS

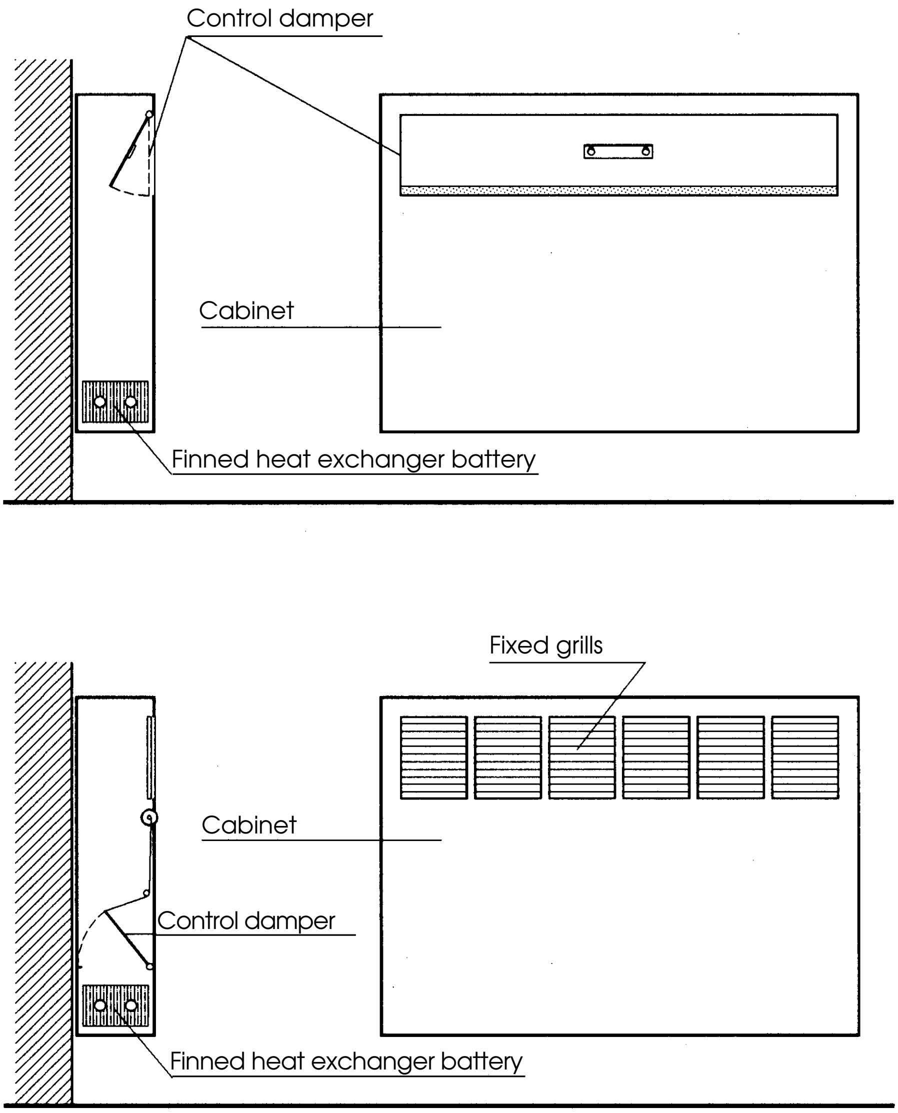
These convectors are constructed with a "fret" of fins arranged in such a way as to form small convective chimneys. These convector, which are of compact and robust construction, can be installed along external walls, in recesses and under the floor.



Plan and sectional view of a convector with finned channels

CABINET CONVECTORS

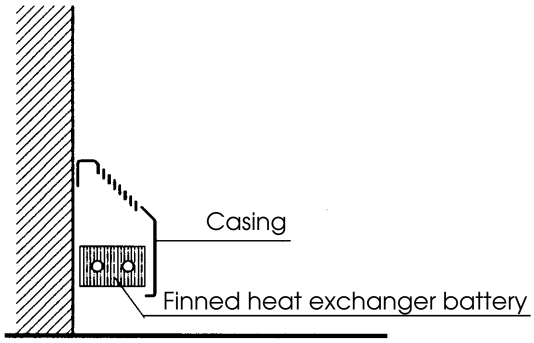
These convectors are constructed from a battery of finned tubes mounted within a cabinet which is used to create the "chimney" effect. The heat emission of these convectors can be controlled by adjusting the dampers which vary the quantity of air passing through the battery.



Cabinet convectors

BASEBOARD CONVECTORS

These are constructed from finned tubes mounted within small casings as shown in the figure below.
The dimensions and shapes of these convectors are such that a simple "baseboard" installation, i.e. along the lower strip of the wall, can be achieved.



Baseboard convector

NOMINAL HEAT OUTPUT OF A CONVECTOR

This is the heat output under test conditions. These conditions taken from various National Standards may be summarised as follows:

- **measuring equipment and instrumentation:** as specified in the Standard;
- **temperatures of the fluids:**
  - t_e = 85°C, inlet temperature of heating fluid,
  - t_u = 75°C, outlet temperature of heating fluid,
  - t_a = 20°C, temperature of the air;
- **test pressure:** 101,3 kPa.

DESIGN TEMPERATURE OF THE HEATING FLUID

It is recommended that this temperature should be between 60 and 70°C. Higer temperatures are not recommended as they could:

- **initiate strong convection currents** and thus contribute to the formation of areas of hot air at ceiling and cold air at the floor;
- **cause a "baking" of atmospheric dust thereby** causing irritation of the respiratory system as well as staining of the walls behind and above the radiator.

On the other hand excessively low design temperatures results in a significant increase of the cost of the system and the space occupied by the convectors.

EFFECTIVE HEAT OUTPUT OF A CONVECTOR

This is the heat output of a convector to atmosphere under actual operating conditions. Its value It may be calculated using the following formula:

$$\boxed{Q_{eff} = Q_{nom} \cdot F} \qquad (1)$$

where: Q_{eff} = effective heat output in W or kcal/h
Q_{nom} = nominal heat output in W or kcal/h
F = nondimensional correction factor

The overall correction factor **F** can be determined with the equation:

$$\boxed{F = F_t \cdot F_{al} \cdot F_{in} \cdot F_v} \qquad (2)$$

where: F_t = correction factor for the **different fluid temperature**
F_{al} = correction factor for **altitude**
F_{in} = correction factor for the type of **installation**
F_v = correction factor for the **velocity of the fluid**

These correction factors are calculated below with respect to the test conditions defined above. In addition the following formula is valid for determining F_t

$$\boxed{Q' = B \cdot (t_m - t_a)^{1{,}4}} \qquad (3)$$

where: Q' = heat output of the convector in W o kcal/h
B = characteristic constant for the convector type in $W/°C^{1{,}4}$ or $kcal/(h \cdot °C^{1{,}4})$
t_m = mean temperature of the heating fluid in °C
t_a = ambient temperature in °C

Note:

Formula (3) **can be considered valid for mean temperatures of the heating fluid between 50 and 100°C.**

CORRECTION FACTOR FOR DIFFERENT FLUID TEMPERATURE

This factor is used to determine the heat output of a convector when the ambient temperature (t_a) and the mean temperature of the heating fluid (t_m) differ from the Standard test temperatures. Its value is given by the following ratio:

$$F_t = \frac{Q'_{eff}}{Q'_{nom}} \qquad (4)$$

Using formula (3) Q'_{eff} e Q'_{nom} can be expressed as follows:

$$Q'_{eff} = B \cdot (t_m - t_a)^{1,4} \qquad (5)$$

$$Q'_{nom} = B \cdot (t_m - t_a)^{1,4} = B \cdot (80 - 20)^{1,4} \qquad (6)$$

Therefore:

$$F_t = \frac{Q'_{eff}}{Q'_{nom}} = \frac{B \cdot (t_m - t_a)^{1,4}}{B \cdot (80 - 20)^{1,4}} \qquad (7)$$

By semplifying the terms of this equation, the following is obtained:

$$\boxed{F_t = \left(\frac{t_m - t_a}{60} \right)^{1,4}} \qquad (8)$$

The values of the factors F_t obtained from formula (8) **are given in Table 1.**

It may also be necessary to calculate **F_t** in cases where only the mean temperature (t_m) of the fluid or only the temperature (t_a) of the ambient air vary. For this purpose by first replacing "t_a" and then "t_m"by the relative test values it is possible to obtain the following formulae:

– **(8.1)** applicable to **variable** t_m and t_a = 20°C (test temperature)
– **(8.2)** applicable to **variable** t_a and t_m = 80°C (test temperature).

$$\boxed{F_t\ (t_a = 20°C) = \left(\frac{t_m - 20}{60}\right)^{1,4}} \tag{8.1}$$

$$\boxed{F_t\ (t_m = 80°C) = \left(\frac{80 - t_a}{60}\right)^{1,4}} \tag{8.2}$$

Note:

A European Standard which specifies a mean temperature of the fluid of 70°C. is at present in a process of being approved for the thermal testing of convectors. Based on this condition **the value of the correction factor Ft** can be determined using:

$$\boxed{F_t = \left(\frac{t_m - t_a}{50}\right)^{1,4}} \tag{9}$$

The values of the factor F_t obtained from the formula (9) **are given in Table 2.**

TAB. 1 - CORRECTION FACTOR F_t FOR CONVECTORS

measurement conditions of nominal heat output
t_m = 80°C and t_a = 20°C

mean temp. of the heating fluid	temperature of the air								
	10°C	12°C	14°C	16°C	18°C	20°C	22°C	24°C	26°C
50°C	0,57	0,53	0,49	0,45	0,41	0,38	0,34	0,31	0,28
52°C	0,61	0,57	0,53	0,49	0,45	0,41	0,38	0,34	0,31
54°C	0,65	0,61	0,57	0,53	0,49	0,45	0,41	0,38	0,34
56°C	0,69	0,65	0,61	0,57	0,53	0,49	0,45	0,41	0,38
58°C	0,73	0,69	0,65	0,61	0,57	0,53	0,49	0,45	0,41
60°C	0,77	0,73	0,69	0,65	0,61	0,57	0,53	0,49	0,45
62°C	0,82	0,77	0,73	0,69	0,65	0,61	0,57	0,53	0,49
64°C	0,86	0,82	0,77	0,73	0,69	0,65	0,61	0,57	0,53
66°C	0,91	0,86	0,82	0,77	0,73	0,69	0,65	0,61	0,57
68°C	0,95	0,91	0,86	0,82	0,77	0,73	0,69	0,65	0,61
70°C	1,00	0,95	0,91	0,86	0,82	0,77	0,73	0,69	0,65
72°C	1,05	1,00	0,95	0,91	0,86	0,82	0,77	0,73	0,69
74°C	1,09	1,05	1,00	0,95	0,91	0,86	0,82	0,77	0,73
76°C	1,14	1,09	1,05	1,00	0,95	0,91	0,86	0,82	0,77
78°C	1,19	1,14	1,09	1,05	1,00	0,95	0,91	0,86	0,82
80°C	1,24	1,19	1,14	1,09	1,05	1,00	0,95	0,91	0,86
82°C	1,29	1,24	1,19	1,14	1,09	1,05	1,00	0,95	0,91
84°C	1,34	1,29	1,24	1,19	1,14	1,09	1,05	1,00	0,95
86°C	1,39	1,34	1,29	1,24	1,19	1,14	1,09	1,05	1,00
88°C	1,44	1,39	1,34	1,29	1,24	1,19	1,14	1,09	1,05
90°C	1,50	1,44	1,39	1,34	1,29	1,24	1,19	1,14	1,09
92°C	1,55	1,50	1,44	1,39	1,34	1,29	1,24	1,19	1,14
94°C	1,60	1,55	1,50	1,44	1,39	1,34	1,29	1,24	1,19
96°C	1,66	1,60	1,55	1,50	1,44	1,39	1,34	1,29	1,24
98°C	1,71	1,66	1,60	1,55	1,50	1,44	1,39	1,34	1,29
100°C	1,76	1,71	1,66	1,60	1,55	1,50	1,44	1,39	1,34

TAB. 2 - CORRECTION FACTOR F_t FOR CONVECTORS

measuring conditions for nominal heat output
t_m = 70°C and t_a = 20°C

temperature of the heating fluid	temperature of the air								
	10°C	12°C	14°C	16°C	18°C	20°C	22°C	24°C	26°C
50°C	0,73	0,68	0,63	0,58	0,54	0,49	0,44	0,40	0,36
52°C	0,78	0,73	0,68	0,63	0,58	0,54	0,49	0,44	0,40
54°C	0,84	0,78	0,73	0,68	0,63	0,58	0,54	0,49	0,44
56°C	0,89	0,84	0,78	0,73	0,68	0,63	0,58	0,54	0,49
58°C	0,94	0,89	0,84	0,78	0,73	0,68	0,63	0,58	0,54
60°C	1,00	0,94	0,89	0,84	0,78	0,73	0,68	0,63	0,58
62°C	1,06	1,00	0,94	0,89	0,84	0,78	0,73	0,68	0,63
64°C	1,11	1,06	1,00	0,94	0,89	0,84	0,78	0,73	0,68
66°C	1,17	1,11	1,06	1,00	0,94	0,89	0,84	0,78	0,73
68°C	1,23	1,17	1,11	1,06	1,00	0,94	0,89	0,84	0,78
70°C	1,29	1,23	1,17	1,11	1,06	1,00	0,94	0,89	0,84
72°C	1,35	1,29	1,23	1,17	1,11	1,06	1,00	0,94	0,89
74°C	1,41	1,35	1,29	1,23	1,17	1,11	1,06	1,00	0,94
76°C	1,48	1,41	1,35	1,29	1,23	1,17	1,11	1,06	1,00
78°C	1,54	1,48	1,41	1,35	1,29	1,23	1,17	1,11	1,06
80°C	1,60	1,54	1,48	1,41	1,35	1,29	1,23	1,17	1,11
82°C	1,67	1,60	1,54	1,48	1,41	1,35	1,29	1,23	1,17
84°C	1,73	1,67	1,60	1,54	1,48	1,41	1,35	1,29	1,23
86°C	1,80	1,73	1,67	1,60	1,54	1,48	1,41	1,35	1,29
88°C	1,86	1,80	1,73	1,67	1,60	1,54	1,48	1,41	1,35
90°C	1,93	1,86	1,80	1,73	1,67	1,60	1,54	1,48	1,41
92°C	2,00	1,93	1,86	1,80	1,73	1,67	1,60	1,54	1,48
94°C	2,07	2,00	1,93	1,86	1,80	1,73	1,67	1,60	1,54
96°C	2,14	2,07	2,00	1,93	1,86	1,80	1,73	1,67	1,60
98°C	2,21	2,14	2,07	2,00	1,93	1,86	1,80	1,73	1,67
100°C	2,28	2,21	2,14	2,07	2,00	1,93	1,86	1,80	1,73

CORRECTION FACTOR FOR ALTITUDE

This factor is used to determine the heat output of a convector not installed at sea level. It takes into account the fact that the density of the air and thus its capacity for conveying heat is reduced as the altitude increases. This factor may be calculated using the following formula:

$$\boxed{F_{al} = \frac{P_o}{1{,}5 \cdot P_o - 0{,}5 \cdot P}} \qquad (10)$$

where: P_o = atmospheric pressure at sea level in kPa
P = atmospheric pressure at site in kPa

The value of P_o is 101,3 kPa whilst the value of P can be calculated using the following equation:

$$\boxed{P = 101{,}3 - 0{,}0113 \cdot H} \qquad (11)$$

where: H = height above sea level in m

The value of the factor F_{al} obtained from formula (10) **are given in Table 3.**

TAB. 3 - CORRECTION FACTOR F_{al} FOR CONVECTORS

altitude	atmospheric pressure	F_{al}
750 m	92,8 kPa	0,96
1.000 m	90,0 kPa	0,95
1.250 m	87,2 kPa	0,93
1.500 m	84,4 kPa	0,92
1.750 m	81,5 kPa	0,91

CORRECTION FACTOR FOR THE TYPE OF INSTALLATION

This factor is used to determine the heat output of a convector not installed in a recess, under the floor or within a cabinet. Its value takes into account the fact that these installations can significantly reduce the heat transfer between the convector and the surrounding atmosphere.

In the case of convectors with single fins and finned channels the average value of factor F_{in} can be considered to be :

F_{in} = 0,95 ÷ 1,03 for installation in a recess.

F_{in} = 0,80 ÷ 0,85 for installation under the floor.

F_{in} = 1,05 ÷ 1,10 for installation within a cabinet.

CORRECTION FACTOR FOR THE VELOCITY OF THE FLUID

This factor is used to determine the heat output of a convector when the velocity of the fluid differs from the test velocity. Its value depends on the structural characteristic of the convector and in practice can only be determined experimentally.

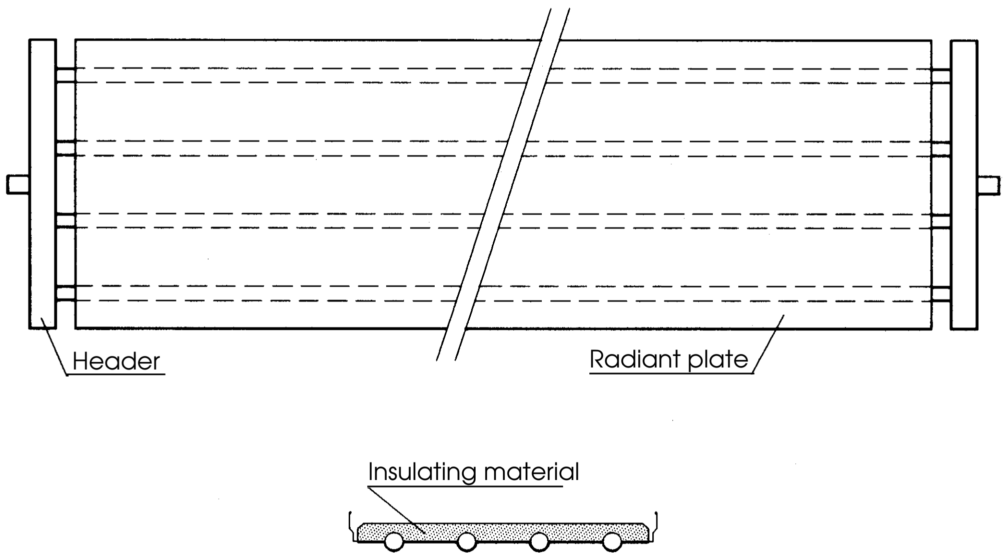
Normally the manufactures give the minimum velocity (or flow rate) **required to allow consider this factor to be considered as zero.**

RADIANT AND NATURAL CONVECTION EMITTERS

- INSTALLATION OF THE EMITTERS
- NOMINAL HEAT OUTPUT OF THE EMITTERS
- DESIGN TEMPERATURE OF THE FLUID
- EFFECTIVE HEAT OUTPUT OF THE EMITTERS
  - CORRECTION FACTOR FOR DIFFERENT FLUID TEMPERATURES
  - CORRECTION FACTOR FOR THE EFFECT OF INSTALLATION HEIGHT
  - CORRECTION FACTOR FOR THE VELOCITY OF THE FLUID

These are emitters which give off heat by natural convection and radiation. They basically consist of grids of tubes onto which are fixed metal plates. These plates normally have lengths of between 4 and 9 metres and are overlaid with a mattress of mineral wool which limits the upwards loss of heat. The plates may also have small dimples which are capable of reducing lateral convective movements.

These emitters are used for heating exhibition halls, gymnasia, swimming pools, stud farms, garages, laboratories, workshops etc.



Plan and sectional view of an emitter

In many cases radiant and natural convection emitters constitute a suitable alternative to other emitters particularly to unit heaters relative to which they have the following advantages:

- **lower running costs** (about 10 - 20%) due to reduced stratification of the air;
- **significantly reduced convective movement** and consequently improved environmental conditions near areas suffering air pollution from industrial processes;
- **operation without a fans,** and consequently no noise problems or problems relating to motor maintenance or fire precautions (this aspect should be carefully considered in locations near inflammable and explosive substances).

On the other hand they have the following disadvantages:

- **higher manufacturing costs;**
- **possible difficulties with location** of the emitters due to space and architectural considerations.

INSTALLATION FOR RADIANT AND NATURAL CONVECTION EMITTERS

The following conditions must be met for the correct installation of these emitters:

1. **Avoid excessive radiation intensities at head height**

   This requirement can be met by installing these emitters at a height which is related to the mean temperature of the fluid and which is **not less than those given in Table 1.**

TAB. 1 - MINIMUM INSTALLATION HEIGHT OF THE EMITTERS

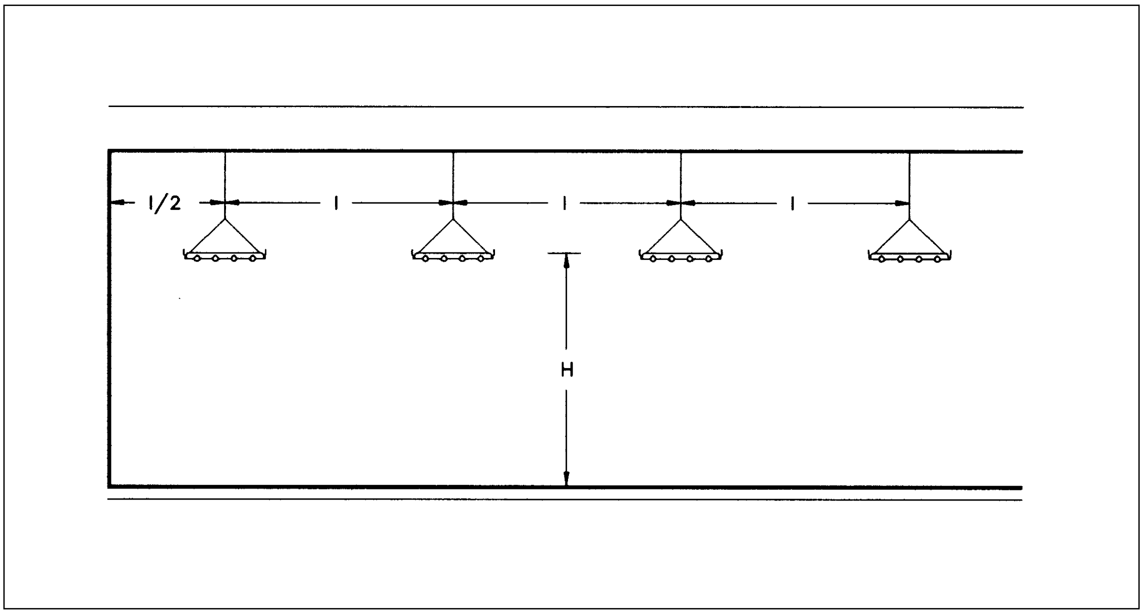
Mean temp. of heating fluid	distance between centres of tubes - 100 mm	distance between centres of tubes 150 mm
60°C	H_{min} = 3,80 mm	H_{min} = 3,60 m
80°C	H_{min} = 4,30 mm	H_{min} = 4,10 m
100°C	H_{min} = 4,70 mm	H_{min} = 4,50 m
120°C	H_{min} = 5,10 mm	H_{min} = 4,90 m
140°C	H_{min} = 5,50 mm	H_{min} = 5,30 m
160°C	H_{min} = 5,90 mm	H_{min} = 5,70 m
180°C	H_{min} = 6,40 mm	H_{min} = 6,20 m

2. **Effective use is made of the radiant effect**

   To achieve this the heating units should be installed as low as possible compatible with the limits defined in Section 1 and with the requirements relating to the use of the area.

3. **A uniform distribution of the heat is obtained**

   For this purpose it is recommended that the distance **I** (see figure below) is less than the installed height **H** of the emitters.



4. **Limiting the areas of shade caused by the emitters**

   This objective can be achivied by installing the emitters with their longitudinal axis parallel to the skylights or the glazed sections of the sawtooth roofs.

NOMINAL HEAT OUTPUT OF RADIANT AND NATURAL CONVECTION EMITTERS

This is the heat output under test conditions. These conditions taken from various National Standards, may be summarised as follows:

- measuring equipment and instrumentation: as specified in the Standard;
- temperatures of the fluids:

- t_e = 85°C, inlet temperature of the heating fluid,
- t_u = 75°C, outlet temperature of the heating fluid,
- t_a = 20°C, temperature of the air.

- test pressure: 101.3 kPa

DESIGN TEMPERATURE OF THE FLUID

On commercial systems it is recommended that this temperature is between 60 and 90°C. Higer values (up to 170, 180°C) could,however, be used on industrial systems

Under all circumstances, to prevent conditions of discomfort, **the design temperature of the fluid should be such that it enables the limits defined Table 1 to be met.**

EFFECTIVE HEAT OUTPUT OF RADIANT AND NATURAL CONVECTION EMITTERS

This is the heat output to atmosphere under installed operating conditions.
Its value may be calculated using the following formula:

$$\boxed{Q_{eff} = Q_{nom} \cdot F} \qquad (1)$$

where: Q_{eff} = effective heat output in W or kcal/h
Q_{nom} = nominal heat output in W or kcal/h
F = nondimensional correction factor

The overall correction factor **F** can be determined using the following equation:

$$\boxed{F = F_t \cdot F_{in} \cdot F_v} \qquad (2)$$

where: F_t = correction factor for the **different fluid temperature**
F_{in} = correction factor for **altitude**
F_v = correction factor for the **velocity of the heating fluid**

These correction factors are calculated below based on the test conditions defined above. In addition the following formula may be used for determining the factor F_t,

$$\boxed{Q' = B \cdot (t_m - t_a)^{1,15}} \qquad (3)$$

where: Q' = heat output of the heating unit in W or kcal/h
B = characteristic constant of the emitter in W/°C ^{1,15} or kcal/(h·°C ^{1,15})
t_m = mean temperature of the heating fluid in °C
t_a = ambient temperature in °C

Note:

Formula (3) **can be considered valid for mean temperatures of the fluid between 60 and 100°C.**

CORRECTION FACTOR FOR DIFERENT FLUID TEMPERATURES

This factor used determine the heat output of a radiant and natural convector emitter when the ambient temperature (t_a) and the mean temperature of the fluid (t_m) differ from the test values. By definition its value is given by the ratio:

$$F_t = \frac{Q'_{eff}}{Q'_{nom}} \tag{4}$$

Using Formula (3) Q'_{eff} e Q'_{nom} may be expressed as follows:

$$Q'_{eff} = B \cdot (t_m - t_a)^{1,15} \tag{5}$$

$$Q'_{nom} = B \cdot (t_m - t_a)^{1,15} = B \cdot (80 - 20)^{1,15} \tag{6}$$

Therefore:

$$F_t = \frac{Q'_{eff}}{Q'_{nom}} = \frac{B \cdot (t_m - t_a)^{1,15}}{B \cdot (80 - 20)^{1,15}} \tag{7}$$

By semplifying the terms of this equation the following is obtained:

$$\boxed{F_t = \left(\frac{t_m - t_a}{60}\right)^{1,15}} \tag{8}$$

The values of factor F_t obtained from formula (8) **are given in Table 2.**

It may also be necessary to calculate F_t in cases where only the mean temperature (t_m) of the heating fluid or only the temperature (t_a) of the ambient air vary. For this purpose by substituting the relative test values in (8), first for t_a and then for t_m, the following formulae can be obtained:

– (8.1) applicable to variable t_m and t_a = 20°C (test temperature)
– (8.2) applicable to variable t_a and t_m = 20°C (test temperature).

$$\boxed{F_t\ (t_a = 20°C) = \left(\frac{t_m - 20}{60}\right)^{1,15}} \qquad (8.1)$$

$$\boxed{F_t\ (t_m = 80°C) = \left(\frac{80 - t_a}{60}\right)^{1,15}} \qquad (8.2)$$

TAB. 2 - CORRECTION FACTOR F_t FOR RADIANT AND NATURAL CONVECTION EMITTERS

measurement conditions of nominal heat output
t_m = 80°C and t_a = 20°C

Mean temp of the heating fluid	temperature of the air								
	10°C	12°C	14°C	16°C	18°C	20°C	22°C	24°C	26°C
60°C	0,81	0,77	0,74	0,70	0,66	0,63	0,59	0,56	0,52
62°C	0,85	0,81	0,77	0,74	0,70	0,66	0,63	0,59	0,56
64°C	0,89	0,85	0,81	0,77	0,74	0,70	0,66	0,63	0,59
66°C	0,92	0,89	0,85	0,81	0,77	0,74	0,70	0,66	0,63
68°C	0,96	0,92	0,89	0,85	0,81	0,77	0,74	0,70	0,66
70°C	1,00	0,96	0,92	0,89	0,85	0,81	0,77	0,74	0,70
72°C	1,04	1,00	0,96	0,92	0,89	0,85	0,81	0,77	0,74
74°C	1,08	1,04	1,00	0,96	0,92	0,89	0,85	0,81	0,77
76°C	1,12	1,08	1,04	1,00	0,96	0,92	0,89	0,85	0,81
78°C	1,15	1,12	1,08	1,04	1,00	0,96	0,92	0,89	0,85
80°C	1,19	1,15	1,12	1,08	1,04	1,00	0,96	0,92	0,89
82°C	1,23	1,19	1,15	1,12	1,08	1,04	1,00	0,96	0,92
84°C	1,27	1,23	1,19	1,15	1,12	1,08	1,04	1,00	0,96
86°C	1,31	1,27	1,23	1,19	1,15	1,12	1,08	1,04	1,00
88°C	1,35	1,31	1,27	1,23	1,19	1,15	1,12	1,08	1,04
90°C	1,39	1,35	1,31	1,27	1,23	1,19	1,15	1,12	1,08
92°C	1,43	1,39	1,35	1,31	1,27	1,23	1,19	1,15	1,12
94°C	1,47	1,43	1,39	1,35	1,31	1,27	1,23	1,19	1,15
96°C	1,51	1,47	1,43	1,39	1,35	1,31	1,27	1,23	1,19
98°C	1,55	1,51	1,47	1,43	1,39	1,35	1,31	1,27	1,23
100°C	1,59	1,55	1,51	1,47	1,43	1,39	1,35	1,31	1,27

CORRECTION FACTOR FOR INSTALLATION HEIGHT

This factor is used to determine the heat output of an emitter of this type in relation to the installation height (H). Its value takes into account the fact that as the height (H) increases the useful radiant effect of the heating unit is reduced.

The average values for the factor F_{in} given in Table 3 may be considered to be correct .

TAB. 3 - CORRECTION FACTOR F_{in} FOR RADIANT AND NATURAL CONVECTION EMITTERS

H	6 m	8 m	10 m	12 m	14 m	16 m	18 m	20 m
F_{in}	1,00	0,95	0,90	0,86	0,82	0,79	0,76	0,75

CORRECTION FACTOR FOR THE VELOCITY OF THE FLUID

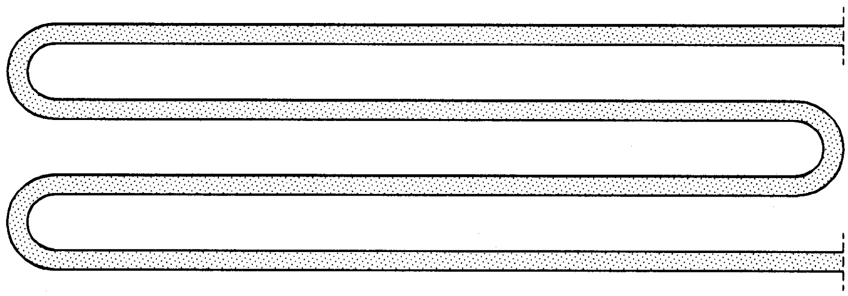
This factor is used to determine the heat output of an emitter of this type when the velocity of the fluid differs from the test value. Its value depends on the structural characteristics of the emitter and, in practice, can be determined experimentally.

Manufacturers normally give the minimum velocity of the fluid (related to the tubes of the emitter) **required to be enable the effect of this factor to effectively be zero.**

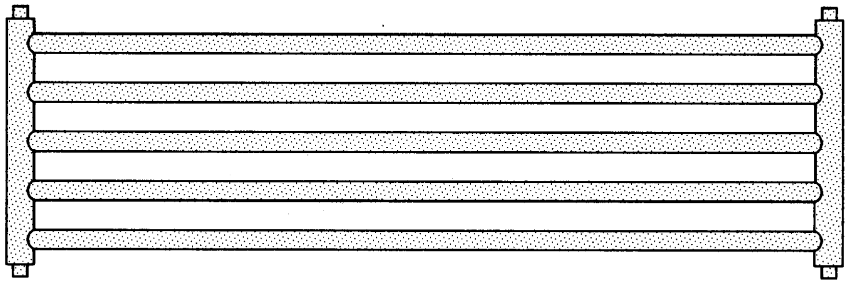
T U B E S

Normal smooth tubes can also be used as heat emitters. They are used particularly for heating industrial areas, depots, warehouses, basements etc

Normally steel tubes are used (with diameters varying from 3/4" to 2") **either singlely, in coils or in a battery arrangement .**



Smooth tube in a coil arrangement



Smooth tubes in a battery arrangement

NOMINAL HEAT OUTPUT PER METRE OF TUBE

This is the heat output per metre of tube under test conditions. These conditions taken from various National Standards, may be summarised as follows:

- measuring equipment and instrumentation: as specified in the Standard;
- temperatures of the fluids:

- t_e = 85°C, inlet temperature of the heating fluid,
- t_u = 75°C, outlet temperature of the heating fluid,
- t_a = 20°C, temperature of the air.

- test pressure: 101.3kPa

TAB. 1 - NOMINAL HEAT OUTPUT OF SMOOTH STEEL TUBES

measurement conditions of the nominal heat output
t_m = 80°C e t_a = 20°C

Arrangements of the tubes	diameter	output kcal / (h · m)	output W/m
Horizontal individual tubes	3/4”	64	74
	1”	76	88
	1 1/4”	93	108
	1 1/2”	105	122
	2”	125	145
Vertical individual tubes	3/4”	55	64
	1”	68	79
	1 1/4”	86	100
	1 1/2”	98	114
	2”	120	139

EFFECTIVE HEAT OUTPUT PER METRE OF TUBE

This is the heat output to atmosphere under installed operating conditions. Its value may be calculated using the following formula:

$$\boxed{Q_{eff} = Q_{nom} \cdot F} \qquad (1)$$

where: Q_{eff} = effective heat output in W or kcal/h
Q_{nom} = nominal heat output W or kcal/h
F = nondimensional correction factor

The overall correction factor **F** can be determined using the following :

$$\boxed{F = F_t \cdot F_{al} \cdot F_s \cdot F_{vr} \cdot F_v} \qquad (2)$$

where F_t = correction factor for the **different fluid temperature**
F_{al} = correction factor for **altitude**
F_s = correction factor for the **orientation of the tubes**
F_{vr} = correction factor for the effect of **painting**
F_v = correction factor depending on the **velocity of the heating fluid**

Factors F_t, F_{al}, F_{vr} may be considered equal to those for radiators (See relative section).

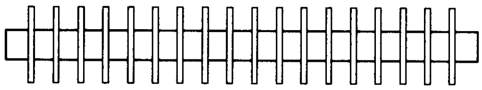
The following average values may be considered **for factor F_s**:

- F_s = 0,90 for tubes in 2 rows,
- F_s = 0,85 for tubes in 3 rows,
- F_s = 0,82 for tubes in 4 rows.

The factor F_v may be considered equal to unity and consequently to have no effect **for fluid velocities of more than 0,3 m/s.**

FINNED TUBES

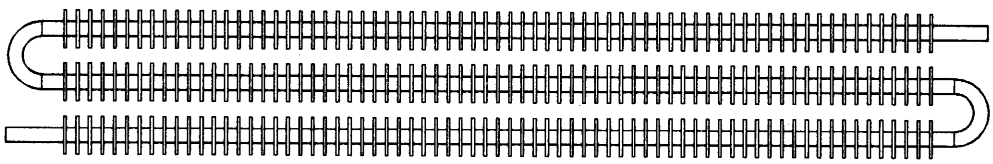
These are smooth tubes to which are attached (by welding or keying) **fins made of plate which can extend as a spiral or orthogonal from the tubes.**



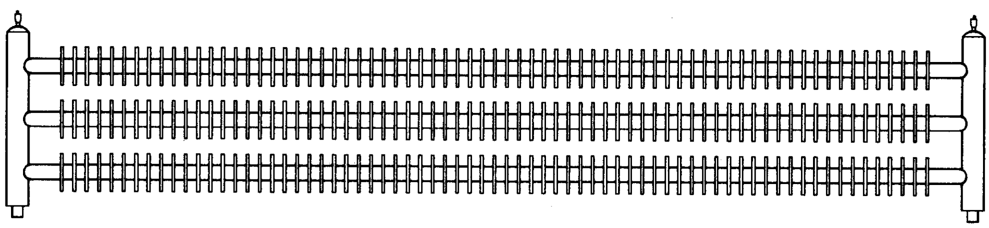
Tube with orthogonal fins

Tube with spiral fins

Finned tubes are used for heating greenhouses, garages, stores, depots, industrial areas, basements etc.



Finned tubes arranged in a coil



Finned tubes arranged as a battery

NOMINAL HEAT OUTPUT PER METRE OF FINNED TUBE

This is the heat output per metre of finned tube under test conditions. These conditions, taken from various National Standards, may be summarised as follows:

- measuring equipment and instrumentation: as specified in the Standard;
- temperatures of the fluids:

- t_e = 85°C, inlet temperature of the heating fluid,
- t_u = 75°C, outlet temperature of the heating fluid,
- t_a = 20°C, temperature of the air.

- test pressure: 101.3 kPa.

TAB. 1 - NOMINAL HEAT OUTPUT OF STEEL FINNED TUBES

measurement conditions of the nominal heat output

t_m = 80°C e t_a = 20°C

tube diameter	height of fins in mm	fins per metre	nominal output kcal / (h · m)	nominal output W/m
1 1/2”	25	80	375	435
		100	450	522
		120	525	609
		150	645	748
	30	60	360	418
		80	450	522
		100	555	644
		120	660	766
2”	25	80	435	505
		100	540	626
		120	615	713
	30	80	540	626
		100	654	759
		120	780	905
	35	80	642	745
		100	780	905
		120	930	1.079

EFFECTIVE HEAT OUTPUT PER METRE OF FINNED TUBE

This is the heat output to atmosphere under installed operating conditions. Its value may be calculated using the following formula:

$$\boxed{Q_{eff} = Q_{nom} \cdot F} \qquad (1)$$

where: Q_{eff} = effective heat output in W or kcal/h
Q_{nom} = nominal heat output in W or kcal/h
F = nondimensional correction factor

The overall correction factor **F** can be determined using the following:

$$\boxed{F = F_t \cdot F_{al} \cdot F_s \cdot F_v} \qquad (2)$$

where: F_t = correction factor for the **different fluid temperature**
F_{al} = correction factor for **altitude**
F_s = correction factor for the **orientation of the tubes**
F_v = correction factor depending on the **velocity of the heating fluid**

Factors F_t, F_{al} may be considered equal to those for convectors (See relative section)

The following average values may be considered **for factor F_s** :

- F_s = 0,90 for 2 rows of tubes,
- F_s = 0,85 for 3 rows of tubes,
- F_s = 0,82 for 4 rows of tubes.

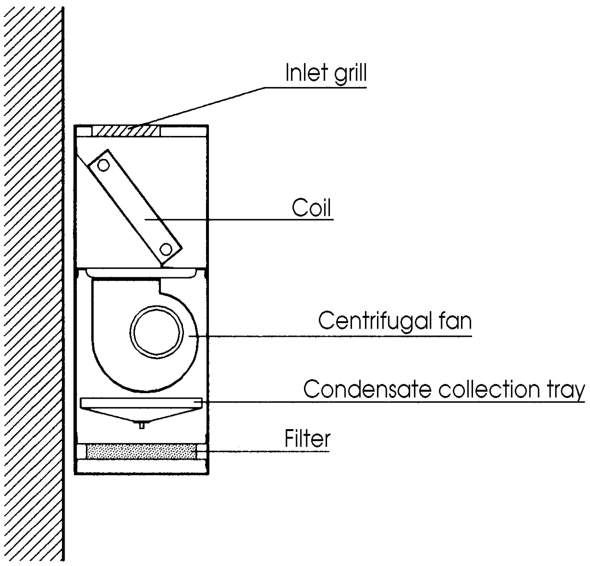
Factors F_v may be considered equal to unity and consequently to have no effect **for fluid velocities of more than 0,3 m/s.**

FAN COIL UNITS

- CLASSIFICATION
- INSTALLATION OF THE FAN COIL UNITS
- CHOICE OF FAN COIL UNITS
  - HEAR OUTPUT AND AIR FLOW RATE OF THE FAN COIL UNITS
  - AIR TEMPERATURE AT THE OUTLET OF THE FAN COIL UNITS IN THE HEATING PHASE
  - PERMISSIBILE SOUND LEVEL
- FAN COIL UNIT MAINTENANCE
- NOMINAL HEAT OUTPUT OF A FAN COIL UNIT IN HEATING PHASE
- DESIGN TEMPERATURE OF THE HEATING FLUID
- EFFECTIVE HEAT OUTPUT OF A FAN COIL UNIT IN HEATING PHASE
- NOMINAL HEAT OUTPUT OF A FAN COIL UNIT IN COOLING PHASE
- DESIGN TEMPERATURE OF THE COOLING FLUID
- EFFECTIVE HEAT OUTPUT OF A FAN COIL UNIT IN COOLING PHASE

Fan coil units give off or withdraw heat from the atmosphere by forced convection. They are constructed with:
– one or two finned coils,
– one or two centrifugal or tangential fans,
– an air filter,
– a condensate collection tray,
– a casing.

They are used for heating and cooling homes, offices, meeting rooms, hotels, hospitals, laboratories etc.



Fan coil unit

CLASSIFICATION

Fan coil units may be classified according to the following criteria:

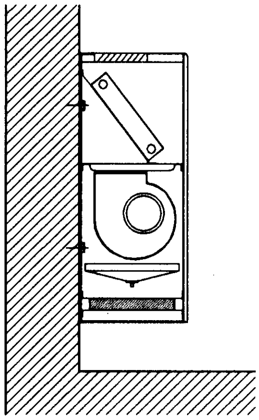
- **on the basis of position:**
  - **the floor,**
  - **the wall,**
  - **the false ceiling,**
  - **the ceiling;**

- **according to the type of enclosure:**
  - **cabinet,**
  - **recessed into structure;**

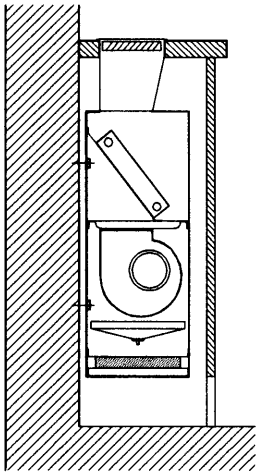
- **on the basis of the position of the fan:**
  - **on the intake** (the fan blows air into the coil),
  - **on suction** (the fan sucks air from the coil);

- **in relation to the characteristics of the air flow:**
  - **with free flow,**
  - **within channels;**

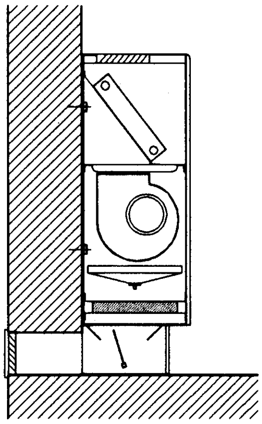
- **on the basis of the number of coils:**
  - **with a single coil** (in 2 tube systems),
  - **with a double coil** (in 4 tube systems, i.e. in systems in which both the hot fluid and the cold fluid circulate at the same time).



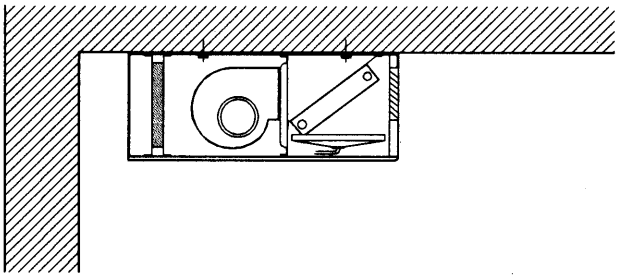
Wall fan coil unit



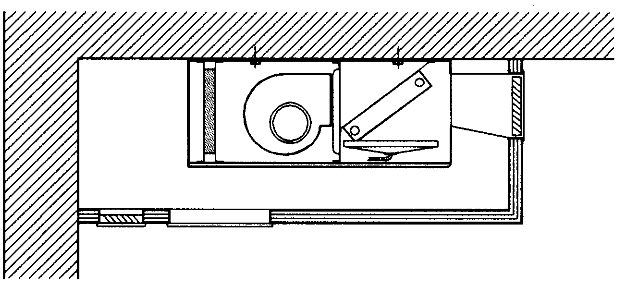
Recessed fan coil unit



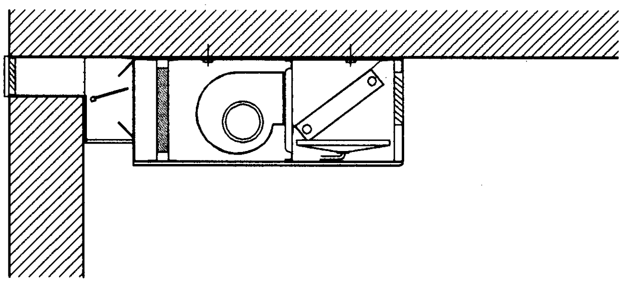
Wall mounted fan coil unit
with external air intake



Ceiling fan coil unit



False ceiling fan coil unit



Ceiling fan coil unit with
external air intake

INSTALLING FAN COIL UNITS

It is advisable to install fan coil units **under a window or along external walls** as in this way:

- **it is possible to counteract the currents of cold air** which form in relation to these surfaces;
- the formation of internal surface condensate around the appliance **is prevented or reduced.**

CHOICE OF FAN COIL UNIT

To make the right choice of fan coil unit the following factors should be considered

- heat output and flow rate of the air from the fan coil units,

- outlet temperature of the air,

- noise level.

HEAT OUTPUT AND AIR FLOW RATE OF FAN COIL UNITS

In medium to large locations it is advisable to subdivide the rquired heat output over several fan coil units. Excessively concentreted heat outputs can cause non-uniform internal temperatures.

To guarantee a satisfactory distribution of the heat it is also advisable that **the air flow rate is not less than 3.5 times the volume of the space to be heated.**

TEMPERATURE OF THE AIR AT THE FAN COIL OUTLET IN THE HEATING PHASE

It is desiderable that during the heating phase the temperature of the air at the outlet of fan coil units is between 35 and 50°C.

With these values it is possible to achieve a good compromise between two different requirements:

• to prevent air currents produced by the fan coil units causing a cold sensation,

• to prevent significant air stratification.

The air temperature at the outlet of fan coil units is normally given in the manufacturers' technical specifications. If not it can be calculated using the following formulae:

$$t_{au} = t_{ae} + \frac{(273 + t_{ae}) \cdot Q}{84{,}6 \cdot G} \qquad (1)$$

$$t_{au} = t_{ae} + \frac{(273 + t_{ae}) \cdot Q}{84{,}6 \cdot G - Q} \qquad (2)$$

where: t_{au} = temperature of the air at the outlet of the fan coil unit in °C
t_{ae} = temperature of the air at the inlet to the fan coil unit in °C
Q = heat output produced under operating conditions in kcal/h
G = flow rate of air at 20°C, m^3/h

Formula (1) **applies to fan coil units with the fan mounted at the inlet,** i.e. fan blows air to the heat exchanger battery.

Formula (2) **applies to fan coil units with fan on suction,** i.e. fan sucks air from the heat exchanger battery.

The temperature of the air at the inlet of the fan coil unit (t_{ae}) **is considered:**

– **equal to the ambient temperature** when there is total recirculation of internal air;

– **equal to the external temperature** when all the air passing through the fan coil unit comes from outside;

– **equal to the temperature of the mixed air** when the air passing through the fan coil unit is partly from the inside and partly from the outside.
The temperature of the mixed air can be determined using formula (3) in the UNIT HEATER section.

PERMISSIBILE NOISE LEVEL

The noise produced by fan coil units, which is normally given on their technical specification, **must not exceed the ambient permissible noise level.** This value depends on the use of the premises and can be fixed on the basis of values recommended in the technical literature.

Table 1 of the UNIT HEATERS section gives details of noise levels which are normally acceptable.

Note:
Fan coil units are generally fitted with three speed fans and it is best to chose them on the basis of the middle speed. In this way **a restricted sound level** under normal operating conditions is obtained and **quick warmup** can be achieved using maximum speed.

FAN COIL UNIT MAINTENANCE

Correct maintenance of fan coil units requires the following operations and checks to be carried out:

- **cleaning of the filters** on average every month with a vacuum cleaner or using a neutral detergent;
- **replacement of the filters** at least once a year;
- **cleaning of the coil** with a soft brush or with compressed air jets. The frequency of these operations depends on the environment and the efficiency of the filters;
- **cleaning of the condensate** collection tray at the beginning of each summer season and removal of any obstructions in the drainage system.

NOMINAL HEAT OUTPUT OF FAN COIL UNITS IN THE HEATING PHASE

This is the heat output of the fan coil under test conditions. These conditions taken from various National Standards can be summarised as follows:

- **measuring equipment and instrumentation:** as specified in the above Standard;
- **temperatures of the fluids:**
  – t_e = 50, 60, 70°C, inlet temperature of the heating fluid,
  – t_u = 40, 50, 60°C, outlet temperature of the heating fluid,
  – t_{ae} = 20°C, temperature of the air;
- **test pressure:** 101.3 kPa;
- **fan speed:** maximum;
- difference in static pressure between the air inlet and the outlet to be zero;

DESIGN TEMPERATURE OF THE HEATING FLUID

It is recommended that this temperature should be between 50 and 75°C.

In order to avoid cold currents and a significant stratification of the air **the design temperature of the heating fluid should be such as to enable the limits laid down in the section** “AIR TEMPERATURE AT THE OUTLET OF THE FAN COIL UNITS IN THE HEATING PHASE” to be met.

EFFECTIVE HEAT OUTPUT OF FAN COIL UNITS IN THE HEATING PHASE

This is the heat output of fan coil unit under actual operating conditions. Its value may be calculated using the following formula:

$$\boxed{Q_{eff} = Q_{nom} \cdot F} \qquad (3)$$

where: Q_{eff} = effective heat output in W or kcal/h
Q_{nom} = nominal heat output in W or kcal/h
F = nondimensional correction factor

The correction factor **F** can be expressed as follows:

$$\boxed{F = F\,[\,t_m,\ t_{ae},\ v_r,\ h,\ v\,]} \qquad (4)$$

where: t_m = mean temperature of the heating fluid in °C
t_{ae} = temperature of the air at the inlet of the fan coil unit °C
v_r = rotational speed of the fan in r.p.m.
h = height above sea level in m
v = velocity of the heating fluid in m/s.

The solution of this equation is analytically very complex. In practice its value can only be determined experimentally.

The fan coil manufacturers normally:

- **give the value of F** (or give directly the effective heat output of the fan coil units) **as a function of the following variables:**
  - t_m, mean temperature of the heating fluid;
  - t_{ae}, air temperature at the inlet to the fan coil unit;
  - v_r, rotational speed of the fan.
- **consider the correction effect due to the variation in altitude as negligible;**
- **indicate the minimum flow rate required in order that the correction effect associated with the velocity of the heating fluid can be ignored.**

NOMINAL HEAT OUTPUT OF FAN COIL UNITS IN THE COOLING PHASE

This is the heat extracted by the fan coil under test conditions. These conditions taken from various National Standards, can be summarised as follows:

- **measuring equipment and instrumentation:** as specified in the above Standard;
- **temperatures of the fluids:**
  - t_e = 7°C, inlet temperature of the cooling fluid,
  - t_u = 12°C, outlet temperature of the cooling fluid,
  - $t_{ae\,(s)}$ = 27°C, dry bulb temperature of the air at the inlet to the fan coil unit,
  - $t_{ae\,(u)}$ = 19°C, wet bulb temperature of the air at the inlet of the fan coil unit;
- **test pressure:** 101.3 kPa;
- **fan speed:** maximum;
- difference in static pressure between the air inlet and the outlet to be zero;

DESIGN TEMPERATURE OF THE COOLING FLUID

The choice of this temperature, which is normally between 7 and 15°C, depends on the quantity of the heat it is intended to take from the air passing through the fan coil units.

EFFECTIVE HEAT OUTPUT OF FAN COIL UNITS IN THE COOLING PHASE

This is the heat (sensible and latent) **extracted by a fan coil unit under normal operating conditions.** Its value can be calculated using the following formula:

$$\boxed{Q_{eff} = Q_{nom} \cdot F} \qquad (5)$$

where: Q_{eff} = effective heat output in W or kcal/h
Q_{nom} = nominal heat output in W or kcal/h
F = nondimensional correction factor

The correction factor **F** may be expressed as follows:

$$\boxed{F = F\,[\ t_m,\ t_{ae\,(u)},\ v_r,\ h,\ v\ ]} \qquad (6)$$

where: t_m = mean temperature of the cooling fluid in °C
$t_{ae\,(u)}$ = wet bulb temperature of the air at the inlet of the fan coil unit in °C
v_r = rotational speed of the fan in r.p.m.
h = height above sea level in m
v = velocity of the cooling fluid in m/s.

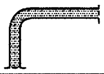
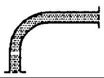
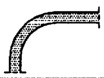
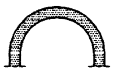
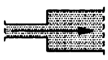
The analytical solution of this equation is very complex. In practice its value can only be found practically.

The manufacturers normally:

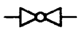
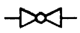
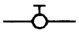
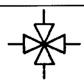
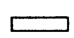
- **give the value of F** (or the effective heat output of the fan coil units) **as a function of the following variables:**
  - t_m, mean temperature of the cooling fluid;
  - $t_{ae\,(u)}$, wet bulb temperature of the air at the inlet to the fan coil unit;
  - v_r, rotational speed of the fan.
- **consider the correction due to the variation in altitude as negligible;**
- **indicate the minimum flow rate required to allow the correction associated with the velocity of the cooling fluid to be ignored;**
- **give either the sensible effective heat or the latent effective heat taken from the room as a function of the following variables:**
  - t_m, mean temperature of the cooling fluid;
  - $t_{ae\,(s)}$, dry bulb temperature of the air at the inlet to the fan coil unit;
  - $t_{ae\,(u)}$, wet bulb temperature of the air at the inlet to the fan coil unit.

REFERENCE TABLE

TAB. 1b - Values of localised loss coefficient ξ (distribution network)

Internal diameter copper tube, PEad, PEX			8÷16 mm	18÷28 mm	30÷54 mm	>54 mm
External diameter steel tube			3/8"÷1/2"	3/4"÷1"	1 1/4"÷2"	>2"
Localised loss type		**Symbol**				
Narrow bend 90°	r/d = 1,5		2,0	1,5	1,0	0,8
Normal bend 90°	r/d = 2,5		1,5	1,0	0,5	0,4
Wide bend 90°	r/d > 3,5		1,0	0,5	0,3	0,3
Narrow bend U	r/d = 1,5		2,5	2,0	1,5	1,0
Normal bend U	r/d = 2,5		2,0	1,5	0,8	0,5
Wide bend U	r/d > 3,5		1,5	0,8	0,4	0,4
Section change			1,0			
Section change			0,5			
T joint			1,0			
T joint			1,0			
T joint			3,0			
T joint			3,0			
Angle joint (45°- 60°)			0,5			
Angle joint (45°- 60°)			0,5			
Bend joint			2,0			
Bend joint			2,0			

TAB. 1b - Values of localised loss coefficient ξ (distribution network)

Internal diameter copper tube, PEad, PEX		8÷16 mm	18÷28 mm	30÷54 mm	>54 mm
External diameter steel tube		3/8"÷1/2"	3/4"÷1"	1 1/4"÷2"	>2"
Localised loss type	**Symbol**				
Shut-off valve		10,0	8,0	7,0	6,0
Shut-off valve		5,0	4,0	3,0	3,0
Reduced passage gate valve		1,2	1,0	0,8	0,6
Total passage gate valve		0,2	0,2	0,1	0,1
Reduced passage ball valve		1,6	1,0	0,8	0,6
Total passage ball valve		0,2	0,2	0,1	0,1
Butterfly valve		3,5	2,0	1,5	1,0
Check valve		3,0	2,0	1,0	1,0
Radiator valve		8,5	7,0	6,0	—
Radiator valve		4,0	4,0	3,0	—
Lockshield		1,5	1,5	1,0	—
Lockshield		1,0	1,0	0,5	—
4-way valve		6,0		4,0	
3-way valve		10,0		8,0	
Passage through radiator		3,0			
Passage through boiler		3,0			

TAB. 2a - Localised pressure losses in mm w.g. for $\sum\xi = 1 \div 15$ (water temp. = 80°C)

velocity m/s	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
0,10	0,5	1,0	1,5	2,0	2,5	3,0	3,5	4,0	4,5	5,0	5,5	5,9	6,4	6,9	7,4
0,11	0,6	1,2	1,8	2,4	3,0	3,6	4,2	4,8	5,4	6,0	6.6	7,2	7,8	8,4	9,0
0,12	0,7	1,4	2,1	2,9	3,6	4,3	5,0	5,7	6,4	7,1	7,8	8,6	9,3	10	11
0,13	0,8	1,7	2,5	3,3	4,2	5,0	5,9	6,7	7,5	8,4	9,2	10	11	12	13
0,14	1,0	1,9	2,9	3,9	4,9	5,8	6,8	7,8	8,7	9,7	11	12	13	14	15
0,15	1,1	2,2	3,3	4,5	5,6	6,7	7,8	8,9	10	11	12	13	14	16	17
0,16	1,3	2,5	3,8	5,1	6,3	7,6	8,9	10	11	13	14	15	16	18	19
0,17	1,4	2,9	4,3	5,7	7,2	8,6	10	11	13	14	16	17	19	20	21
0,18	1,6	3,2	4,8	6,4	8,0	9,6	11	13	14	16	18	19	21	22	24
0,19	1,8	3,6	5,4	7,2	8,9	11	13	14	16	18	20	21	23	25	27
0,20	2,0	4,0	5,9	7,9	9,9	12	14	16	18	20	22	24	26	28	30
0,21	2,2	4,4	6,6	8,7	11	13	15	17	20	22	24	26	28	31	33
0,22	2,4	4,8	7,2	9,6	12	14	17	19	22	24	26	29	31	34	36
0,23	2,6	5,2	7,9	10	13	16	18	21	24	26	29	31	34	37	39
0,24	2,9	5,7	8,6	11	14	17	20	23	26	29	31	34	37	40	43
0,25	3,1	6,2	9,3	12	15	19	22	25	28	31	34	37	40	43	46
0,26	3,3	6,7	10	13	17	20	23	27	30	33	37	40	44	47	50
0,27	3,6	7,2	11	14	18	22	25	29	33	36	40	43	47	51	54
0,28	3,9	7,8	12	16	19	23	27	31	35	39	43	47	50	54	58
0,29	4,2	8,3	13	17	21	25	29	33	38	42	46	50	54	58	63
0,30	4,5	8,9	13	18	22	27	31	36	40	45	49	54	58	62	67
0,31	4,8	9,5	14	19	24	29	33	38	43	48	52	57	62	67	71
0,32	5,1	10	15	20	25	30	36	41	46	51	56	61	66	71	76
0,33	5,4	11	16	22	27	32	38	43	49	54	59	65	70	76	81
0,34	5,7	11	17	23	29	34	40	46	52	57	63	69	74	80	86
0,35	6,1	12	18	24	30	36	42	49	55	61	67	73	79	85	91
0,36	6,4	13	19	26	32	39	45	51	58	64	71	77	83	90	96
0,37	6,8	14	20	27	34	41	47	54	61	68	75	81	88	95	102
0,38	7,2	14	21	29	36	43	50	57	64	72	79	86	93	100	107
0,39	7,5	15	23	30	38	45	53	60	68	75	83	90	98	106	113
0,40	7,9	16	24	32	40	48	55	63	71	79	87	95	103	111	119
0,41	8,3	17	25	33	42	50	58	67	75	83	92	100	108	117	125
0,42	8,7	17	26	35	44	52	61	70	79	87	96	105	114	122	131
0,43	9,2	18	27	37	46	55	64	73	82	92	101	110	119	128	137
0,44	9,6	19	29	38	48	58	67	77	86	96	106	115	125	134	144
0,45	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150
0,46	10	21	31	42	52	63	73	84	94	105	115	126	136	147	157
0,47	11	22	33	44	55	66	77	88	99	109	120	131	142	153	164
0,48	11	23	34	46	57	68	80	91	103	114	126	137	148	160	171
0,49	12	24	36	48	59	71	83	95	107	119	131	143	155	167	178
0,50	12	25	37	50	62	74	87	99	110	124	136	149	161	173	186
0,51	13	26	39	52	64	77	90	103	116	129	142	155	168	180	193
0,52	13	27	40	54	67	80	94	107	121	134	147	161	174	188	201
0,53	14	28	42	56	70	84	97	111	125	139	153	167	181	195	209
0,54	14	29	43	58	72	87	101	116	130	144	159	173	188	202	217
0,55	15	30	45	60	75	90	105	120	135	150	165	180	195	210	225
0,56	16	31	47	62	78	93	109	124	140	155	171	186	202	218	233
0,57	16	32	48	64	80	97	113	129	145	161	177	193	209	225	241
0,58	17	33	50	67	83	100	117	133	150	167	183	200	217	233	250
0,59	17	34	52	69	86	103	121	138	155	172	190	207	224	241	259
velocity m/s	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15

TAB. 2b - Localised pressure losses in mm w.g. for $\sum\xi = 1 \div 15$ (water temp.=80°C)

velocity m/s	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
0,60	18	36	54	71	89	107	125	143	161	178	196	214	232	250	268
0,61	18	37	55	74	92	111	129	147	166	184	203	221	240	258	277
0,62	19	38	57	76	95	114	133	152	171	190	209	229	248	267	286
0,63	20	39	59	79	98	118	138	157	177	197	216	236	256	275	295
0,64	20	41	61	81	101	122	142	162	183	203	223	244	264	284	304
0,65	21	42	63	84	105	126	147	167	188	209	230	251	272	293	314
0,66	22	43	65	86	108	129	151	173	194	216	237	259	281	302	324
0,67	22	44	67	89	111	133	156	178	200	222	245	267	289	311	334
0,68	23	46	69	92	115	137	160	183	206	229	252	275	298	321	344
0,69	24	47	71	94	118	142	165	189	212	236	259	283	307	330	354
0,70	24	49	73	97	121	146	170	194	218	243	267	291	316	340	364
0,71	25	50	75	100	125	150	175	200	225	250	275	300	325	350	375
0,72	26	51	77	103	128	154	180	205	231	257	283	308	334	360	385
0,73	26	53	79	106	132	158	185	211	238	264	290	317	343	370	396
0,74	27	54	81	109	136	163	190	217	244	271	298	326	353	380	407
0,75	28	56	84	111	139	167	195	223	251	279	307	334	362	390	418
0,76	29	57	86	114	143	172	200	229	258	286	315	343	372	401	429
0,77	29	59	88	118	147	176	206	235	264	294	323	353	382	411	441
0,78	30	60	90	121	151	181	211	241	271	301	332	362	392	422	452
0,79	31	62	93	124	155	186	216	247	278	309	340	371	402	433	464
0,80	32	63	95	127	159	190	222	254	285	317	349	381	412	444	476
0,81	33	65	98	130	163	195	228	260	293	325	358	390	423	455	488
0,82	33	67	100	133	167	200	233	267	300	333	366	400	433	466	500
0,83	34	68	102	137	171	205	239	273	307	341	375	410	444	478	512
0,84	35	70	105	140	175	210	245	280	315	350	385	420	454	489	524
0,85	36	72	107	143	179	215	251	286	322	358	394	430	465	501	537
0,86	37	73	110	147	183	220	257	293	330	366	403	440	476	513	550
0,87	38	75	113	150	188	225	263	300	338	375	413	450	488	525	563
0,88	38	77	115	153	192	230	269	307	345	384	422	460	499	537	576
0,89	39	78	118	157	196	235	275	314	353	392	432	471	510	549	589
0,90	40	80	120	161	201	241	281	321	361	401	441	482	522	562	602
0,91	41	82	123	164	205	246	287	328	369	410	451	492	533	574	615
0,92	42	84	126	168	210	252	294	335	377	419	461	503	545	587	629
0,93	43	86	129	171	214	257	300	343	386	429	471	514	557	600	643
0,94	44	88	131	175	219	263	306	350	394	438	482	525	569	613	657
0,95	45	89	134	179	224	268	313	358	402	447	492	537	581	626	671
0,96	46	91	137	183	228	274	320	365	411	457	502	548	594	639	685
0,97	47	93	140	186	233	280	326	373	420	466	513	559	606	653	699
0,98	48	95	143	190	238	286	333	381	428	476	523	571	619	666	714
0,99	49	97	146	194	243	291	340	388	437	486	534	583	631	680	728
1,00	50	99	149	198	248	297	347	396	446	495	545	595	644	694	743
1,10	60	120	180	240	300	360	420	480	540	600	659	719	779	839	899
1,20	71	143	214	285	357	428	499	571	642	713	785	856	927	999	1,070
1,30	84	167	251	335	419	502	586	670	754	837	921	1.005	1.089	1.172	1.256
1,40	97	194	291	388	486	583	680	777	874	971	1,068	1.165	1.262	1.360	1.457
1,50	111	223	334	446	557	669	780	892	1.003	1.115	1.226	1.338	1.449	1.561	1.672
1,60	127	254	381	507	634	761	888	1.015	1.142	1.268	1.395	1.522	1.649	1.776	1.903
1,70	143	286	430	573	716	859	1.002	1.146	1.289	1.432	1.575	1.718	1.861	2.005	2.148
1,80	161	321	482	642	803	963	1.124	1.284	1.445	1.605	1.766	1.926	2.087	2.247	2.408
1,90	179	358	537	715	894	1.073	1.252	1.431	1.610	1.789	1.967	2.146	2.325	2.504	2.683
velocity m/s	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15

TAB. 2c - Localised pressure losses in mm w.g. for $\Sigma\xi = 1 \div 15$ (water temp.=80°C)

velocity m/s	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
2,00	198	396	595	793	991	1.189	1.387	1.585	1.784	1.982	2.180	2.378	2.576	2.775	2.973
2,10	218	437	655	874	1.092	1.311	1.529	1.748	1.966	2.185	2.403	2.622	2.840	3.059	3.277
2,20	240	480	719	959	1.199	1.439	1.679	1.918	2.158	2.398	2.638	2.878	3.117	3.357	3.597
2,30	262	524	786	1.048	1.310	1.573	1.835	2.097	2.359	2.621	2.883	3.145	3.407	3.669	3.931
2,40	285	571	856	1.142	1.427	1.712	1.998	2.283	2.568	2.854	3.139	3.425	3.710	3.995	4.281
2,50	310	619	929	1.239	1.548	1.858	2.168	2.477	2.787	3.097	3.406	3.716	4.026	4.335	4.645
2,60	335	670	1.005	1.340	1.675	2.010	2.345	2.679	3.014	3.349	3.684	4.019	4.354	4.689	5.024
2,70	361	722	1.084	1.445	1.806	2.167	2.528	2.890	3.251	3.612	3.973	4.334	4.695	5.057	5.418
2,80	388	777	1.165	1.554	1.942	2.331	2.719	3.108	3.496	3.884	4.273	4.661	5.050	5.438	5.827
2,90	417	833	1.250	1.667	2.083	2.500	2.917	3.333	3.750	4.167	4.583	5.000	5.417	5.834	6.250
3,00	446	892	1.338	1.784	2.230	2.675	3.121	3.567	4.013	4.459	4.905	5.351	5.797	6.243	6.689
3,10	476	952	1.428	1.905	2.381	2.857	3.333	3.809	4.285	4.761	5.237	5.714	6.190	6.666	7.142
3,20	507	1.015	1.522	2.029	2.537	3.044	3.551	4.059	4.566	5.074	5.581	6.088	6.596	7.103	7.610
3,30	540	1.079	1.619	2.158	2.698	3.237	3.777	4.316	4.856	5.396	5.935	6.475	7.014	7.554	8.093
3,40	573	1.146	1.718	2.291	2.864	3.437	4.009	4.582	5.155	5.728	6.300	6.873	7.446	8.019	8.591
3,50	607	1.214	1.821	2.428	3.035	3.642	4.249	4.885	5.462	6.069	6.676	7.283	7.890	8.497	9.104
3,60	642	1.284	1.926	2.568	3.211	3.853	4.495	5.137	5.779	6.421	7.063	7.705	8.347	8.990	9.632
3,70	678	1.357	2.035	2.713	3.391	4.070	4.748	5.426	6.105	6.783	7.461	8.139	8.818	9.496	10.174
3,80	715	1.431	2.146	2.862	3.577	4.293	5.008	5.724	6.439	7.154	7.870	8.585	9.301	10.016	10.732
3,90	754	1.507	2.261	3.014	3.768	4.522	5.275	6.029	6.782	7.536	8.290	9.043	9.797	10.550	11.304
velocity m/s	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15

CONTINUOS PRESSURE LOSSES

TAB. 5 - STEEL TUBES (inches)
Water temp. = 80°C)

velocity m/s	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
0,60	18	36	54	71	89	107	125	143	161	178	196	214	232	250	268
0,61	18	37	55	74	92	111	129	147	166	184	203	221	240	258	277
0,62	19	38	57	76	95	114	133	152	171	190	209	229	248	267	286
0,63	20	39	59	79	98	118	138	157	177	197	216	236	256	275	295
0,64	20	41	61	81	101	122	142	162	183	203	223	244	264	284	304
0,65	21	42	63	84	105	126	147	167	188	209	230	251	272	293	314
0,66	22	43	65	86	108	129	151	173	194	216	237	259	281	302	324
0,67	22	44	67	89	111	133	156	178	200	222	245	267	289	311	334
0,68	23	46	69	92	115	137	160	183	206	229	252	275	298	321	344
0,69	24	47	71	94	118	142	165	189	212	236	259	283	307	330	354
0,70	24	49	73	97	121	146	170	194	218	243	267	291	316	340	364
0,71	25	50	75	100	125	150	175	200	225	250	275	300	325	350	375
0,72	26	51	77	103	128	154	180	205	231	257	283	308	334	360	385
0,73	26	53	79	106	132	158	185	211	238	264	290	317	343	370	396
0,74	27	54	81	109	136	163	190	217	244	271	298	326	353	380	407
0,75	28	56	84	111	139	167	195	223	251	279	307	334	362	390	418
0,76	29	57	86	114	143	172	200	229	258	286	315	343	372	401	429
0,77	29	59	88	118	147	176	206	235	264	294	323	353	382	411	441
0,78	30	60	90	121	151	181	211	241	271	301	332	362	392	422	452
0,79	31	62	93	124	155	186	216	247	278	309	340	371	402	433	464
0,80	32	63	95	127	159	190	222	254	285	317	349	381	412	444	476
0,81	33	65	98	130	163	195	228	260	293	325	358	390	423	455	488
0,82	33	67	100	133	167	200	233	267	300	333	366	400	433	466	500
0,83	34	68	102	137	171	205	239	273	307	341	375	410	444	478	512
0,84	35	70	105	140	175	210	245	280	315	350	385	420	454	489	524
0,85	36	72	107	143	179	215	251	286	322	358	394	430	465	501	537
0,86	37	73	110	147	183	220	257	293	330	366	403	440	476	513	550
0,87	38	75	113	150	188	225	263	300	338	375	413	450	488	525	563
0,88	38	77	115	153	192	230	269	307	345	384	422	460	499	537	576
0,89	39	78	118	157	196	235	275	314	353	392	432	471	510	549	589
0,90	40	80	120	161	201	241	281	321	361	401	441	482	522	562	602
0,91	41	82	123	164	205	246	287	328	369	410	451	492	533	574	615
0,92	42	84	126	168	210	252	294	335	377	419	461	503	545	587	629
0,93	43	86	129	171	214	257	300	343	386	429	471	514	557	600	643
0,94	44	88	131	175	219	263	306	350	394	438	482	525	569	613	657
0,95	45	89	134	179	224	268	313	358	402	447	492	537	581	626	671
0,96	46	91	137	183	228	274	320	365	411	457	502	548	594	639	685
0,97	47	93	140	186	233	280	326	373	420	466	513	559	606	653	699
0,98	48	95	143	190	238	286	333	381	428	476	523	571	619	666	714
0,99	49	97	146	194	243	291	340	388	437	486	534	583	631	680	728
1,00	50	99	149	198	248	297	347	396	446	495	545	595	644	694	743
1,10	60	120	180	240	300	360	420	480	540	600	659	719	779	839	899
1,20	71	143	214	285	357	428	499	571	642	713	785	856	927	999	1,070
1,30	84	167	251	335	419	502	586	670	754	837	921	1.005	1.089	1.172	1.256
1,40	97	194	291	388	486	583	680	777	874	971	1,068	1.165	1.262	1.360	1.457
1,50	111	223	334	446	557	669	780	892	1.003	1.115	1.226	1.338	1.449	1.561	1.672
1,60	127	254	381	507	634	761	888	1.015	1.142	1.268	1.395	1.522	1.649	1.776	1.903
1,70	143	286	430	573	716	859	1.002	1.146	1.289	1.432	1.575	1.718	1.861	2.005	2.148
1,80	161	321	482	642	803	963	1.124	1.284	1.445	1.605	1.766	1.926	2.087	2.247	2.408
1,90	179	358	537	715	894	1.073	1.252	1.431	1.610	1.789	1.967	2.146	2.325	2.504	2.683
velocity m/s	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15

STEEL TUBES (inches 80°C)

L I T E R A T U R E

1 J. RIETSCHEL - W. RAISS
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SUBJECT INDEX

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