EUROVENT 6/8

RECOMMENDATIONS FOR CALCULATIONS OF ENERGY CONSUMPTION FOR AIR HANDLING UNITS

EUROVENT • CECOMAF

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This document has been prepared by the Eurovent/Cecomaf WG 6C Air Handling Units, which represents the majority of European manufacturers. It is the result of a collective work — chapters prepared by a member of the group have been extensively discussed by all other members.

Particularly important contributions have been provided by Kees van Haperen (Carrier — Holland Heating), Gunnar Berg (Swegon), Kjell Folkesson (Fl kt Woods), Gerne Verhoeven (Verhulst) and Professor Michael Haibel (University of Biberach).

Important note:

Annexes (as listed in the contents) are not available in the downloadable version. These annexes are only available as hard copy which can be ordered from the website.

With the hard copy a CD-ROM is also available with the calculation programs.

This CD-ROM is not available separately.

CONTENTS

PURPOSE

The purpose of these recommendations is to prescribe a model for all energy calculations so that results from different manufactures will be comparable. The energy costs will form the main part in a Life Cycle Cost calculation. A software program for these energy calculations is available in conjunction with the recommendations. The program can also be used for validation of similar calculations in other applications e.g. suppliers software programs.

The guides and software provides a tool for providing consistant interpretations of the calculated energy costs. As engineering judgment is required for entering data and assumptions are made in the analysis this software does not provide 100% accuracy. It is important that all input data and assumptions are detailed when documenting the analysis or presenting the results of an analysis to a third party. When using the Eurovent software this data is presented along with the results.

This document (model) is made as complete as possible from an energy point of view for an Air Handling Unit. The model comprises of; calculations for fans, heating and cooling, energy recovery, humidification and dehumidification, power and energy. It does not however cover the effects due to heat transfer between the ductwork and its surroundings. The calculations have been modelled on a fictitious generic unit as shown below.

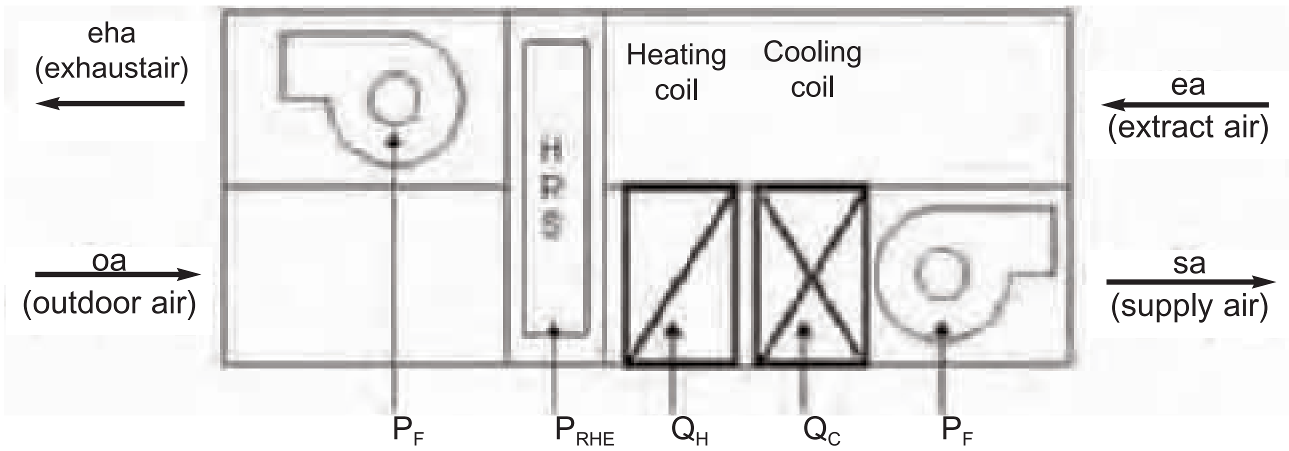


Figure 1 —Air Handling Unit with heat recovery, heating coil and cooling coil

Chapters 1 to 3 handle the fundamental parts like limitations for the document, terms and symbols.

Chapters 4 to 7 deal with the components in an air handling unit e.g. fans. It is written in an educational form and gives background information on energy prices, on secondary energy the components will consume (e.g. pump energy for a heating coil), and under which conditions the energy for a specific component shall be calculated (e.g. the filter conditions when calculating the energy consumption for a fan) and how the functions are influenced by the weather conditions (e.g. freezing in a run around coil).

Chapter 8 describes the Eurovent rules; the general guidelines for the calculation of energy demands in air handling units. Users who only want to know how to calculate may skip the first 7 chapters and go directly to this chapter. Those who are familiar with the subject could also use Annex A and B for a manual calculation or use the computer calculation program.

1 SCOPE

This Eurovent document specifies how to perform calculations of energy demands in air handling units as a whole.
It also deals with specific components of air handling units and prescribes what influence a component will have on secondary energy consumption.
The chapters explain under what conditions the energy consumptions shall be calculated and how weather conditions that influence the performance of a certain component are taken into account

Operation time, operation conditions and variable air volume (VAV) systems will be handled too.
The assessment of prices for thermal heating and cooling energy, based on seasonal efficiency and type of heat— and cold production has also been considered.

The calculations only handle the energy consumptions in air handling units, not for a building as a whole (e.g. how unbalanced air flows will influence the building energy consumption is not calculated).

This document is not applicable to the following:

a) air conditioning units serving a limited area in a building, such as fan coil units;
b) units for domestic ventilation systems;
c) mixing sections;
d) units with heat pumps;
e) units equipped with facilities for thermal energy storage.

2 TERMS AND DEFINITIONS

2.1 GENERAL

Some terms and definitions are taken from the European Standard EN 13053 and EN 12792.

2.1.1
air flow control equipment
device which controls the air flow e.g. frequency inverters, controllable blades, etc.

2.1.2
air handling unit
factory made encased assembly consisting of sections containing a fan or fans and other necessary equipment to perform one or more of the following functions: circulating, filtration, heating, cooling, heat recovery, humidifying, dehumidifying and mixing of air

2.1.3
air heating and cooling coils
heat exchangers by means of which heat is transferred from a heat transfer medium to air (heating coil) or the other way round (cooling coil)

2.1.4
component of air handling unit
smallest discrete functional element of an air-handling unit

2.1.5
external
outside the air handling unit, i.e. in the ductwork upstream and downstream the air handling unit

2.1.6
fan section
section in which one or more fans are installed for air moving

2.1.7
filter section
section including a filter or filters and associated filter frame

2.1.8
heat exchanger
device to transfer thermal energy from one medium to another

2.1.9
heat recovery section
section in which heat (and possibly also moisture) is transferred from one air stream into another, either directly or using an intermediary heat transfer medium

2.1.10
humidifier section
section in which moisture is added to the air

2.1.11
internal
within the boundaries of the air handling unit

2.1.12
purging sector
device of a rotary heat exchanger used to substitute extract air by fresh air in a rotor sector, before it enters the supply air

2.1.13
section of air handling unit
functional element of an air-handling unit, consisting of one or more components in a single casing

2.1.14
transmission
drive system between fan shaft and motor shaft (e.g. V-belt drives)

2.2 FUNCTIONS

2.2.1
air treatment
process by which the state of the air is modified with respect to one or more of its characteristics such as temperature, moisture content, dust content, bacterial count, gas and vapour contents

2.2.2
air type
designation of the air moving through a ventilation, air conditioning or air treatment installation as a function of its location relative to the installation, e.g. outdoor air, supply air, extract air, exhaust air etc.

2.2.3
cooling
removal of latent and/or sensible heat from the air

2.2.4
cooling period
time during the year with demand for additional cooling

2.2.5
dehumidification
reduction of water vapour from the air

2.2.6
frost protection
device to defrost a heat exchanger or to avoid freeze—up of a component

2.2.7
heating
addition of sensible heat to the air

2.2.8
heating period
time during the year with demand for additional heating

2.2.9
humidification
controlled addition of moisture content to an air stream

2.3 CHARACTERISTICS

2.3.1
air flow
movement of air within set boundaries (such as ducts)

2.3.2
air flow rate
mass or volume flow of air passing a given plane divided by time

2.3.3
annual demand of heating energy
amount of additional heating energy during a year

2.3.4
carry over
unwanted transfer of air from external to supply air by one single component

2.3.5
constant air flow
air handling unit operating with a fixed air flow rate

2.3.6
cooling load
the net amount of power to a space, to be conditioned, due to conduction, solar radiation, appliances, people and pets.

2.3.7
design point
performance of a component at the condition it has been designed for

2.3.8
design pressure drop, filter
pressure drop of filter to calculate the design working point of a fan

2.3.9
duration of outdoor condition
time during which a certain outdoor condition prevails

2.3.10
external total pressure difference
difference between the total pressure at the outlet of the air handling unit and the total pressure at the inlet of the unit

2.3.11
heating load
the net amount of power from a space, to be tempered, to outside the space due to conduction and infiltration.

2.3.12
humidification efficiency
ratio between the mass of water evaporated by the humidifier and the theoretical mass needed to achieve saturation at a given temperature

2.3.13
initial pressure drop, filter
pressure drop of filter at the beginning of its life cycle

2.3.14
latent cooling
cooling energy required to reduce the moisture content of the air at fixed temperature

2.3.15
leakage
unwanted air flow

2.3.16
life cycle costs
total costs for the user of an air handling unit from the moment of purchase until the final disposal

2.3.17
momentarily thermal energy consumption
the thermal energy consumption, when operating the air handling unit under a specific outdoor air condition, during a defined length of time, normally one hour

2.3.18
operation conditions
air conditions (air flow rate, pressure, density, temperature and humidity) under which the air handling unit is running

2.3.19
operation time
time during which the air handling unit is running

2.3.20
purging air flow
air flow through the purge section of a heat recovery wheel

2.3.21
recommended final pressure drop, filter
advised pressure drop of filter at the end of its life cycle

2.3.22
room moisture gains
increase of moisture content in the extracted air caused by any internal moisture production in the room and/or moisture transfer from or to the exterior

2.3.23
sensible cooling
cooling energy required to decrease the temperature of the air at fixed moisture content

2.3.24
specific fan power
absorbed electric power consumption of a fan per unit of transported air volume flow

2.3.25
system effect
performance deficit of a fan, caused by an installation type (enclosure, ductwork) deviating from the applied standardised test rig

2.3.26
temperature efficiency
ratio between temperature rises in one air stream divided by the difference in temperature between the inlet temperatures of the two air streams.

2.3.27
variable air flow
air handling unit operating with a controlled variation of the air flow rate

3 SYMBOLS AND ABBREVIATIONS

Some symbols, units and abbreviations are taken from the European Standard EN 13053.

For the purposes of this document, symbols and units given in EN 12792 and in table 3.1 apply, together with those defined with the formulae, in text and in the annexes of this document.

Table 3.1 — Symbols, terms, units and subscripts

Symbol	Term	Unit
α	Angle	¡
c_1, c_2, …	Constant values in formulas	-
c_p	Specific heat capacity	J x (kg x ¡C)$^{-1}$
D, d	Diameter	m
Δ_p	Differential pressure	Pa,kPa
Δ_x	Change in moisture content	g x kg^{-1}
E	Energy price, energy costs	€ x (kWh)$^{-1}$, €
f_p	Porosity factor of rotary heat exchanger structure	-
φ	Relative humidity	%
H_h	Higher heating value	J x m^{-3}, J x kg^{-3}
h	Specific enthalpy	J x kg^{-1}
η	Efficiency	-
η_e	Overall efficiency	-
η_t	Heat recovery efficiency (temperature ratio)	-
η_x	Moisture recovery efficiency (humidity ratio)	-
l	Length	m
n	Rotational speed	r x min^{-1}
P	Power	W
PEL	Electric energy loss on primary side	%
p	Pressure	Pa,kPa
$p_{cooling}$	Costs of thermal cooling energy	€ x (kWh)$^{-1}$
$p_{heating}$	Costs of thermal heating energy	€ x (kWh)$^{-1}$

Table 3.1 *(continued)*

Symbol	Term	Uni
p_{hum}	Costs of thermal energy for humidification	€ x $(kWh)^{-1}$
Q	Thermal energy and annual thermal energy consumption	kJ and kWh x a^{-1}
q_m	Mass flow rate	kg x s^{-1}
q_v	Volume flow rate	m^3 x s^{-1}
ρ	Density	kg x m^{-3}
SFP	Specific fan power	10^{-3} x W x s x m^{-3}
t	Temperature	¡C
t_h	Period of time	h
t_{op}	Annual operation time	h x a^{-1}
t_s	Period of time	s
v_l	Vapour load	g x h^{-1}
W	Annual electric energy consumption	kWh x a^{-1}
x	Moisture content	kg x kg^{-1}
Subscripts		
1	Warmer air (1st subscript) — inlet (2nd subscript)	
2	Colder air (1st subscript) — outlet (2nd subscript)	
11	Warmer air inlet	
12	Warmer air outlet	
21	Colder air inlet	
22	Colder air outlet	
aceq	Air flow control equipment	
ahu	Air handling unit	
air	Air	
an	Annually	
bp	By-pass	
C, c	Cooling	

Table 3.1 *(continued)*

Subscripts	
CAV	Constant air flow
circ	Circulation
coil	Coil
d, day	Daytime (from 06:00 to 18:00)
des	Design
dis	District
distr	Distribution
dry	Dry
E	Eurovent
ea	Extract air
eha	Exhaust air
el	Electric
eq	Equivalent
ext	External air handling unit
f	Fuel
fan	Fan
filter	Filter
fluid	Fluid
H, h	Heating
HE	Heat exchanger
HRS	Heat recovery system
hum	Humidification
i, n	Count number
im	Impact losses (sytem effect generated in a discharge opening of a fan)
in	Inlet
ini	Initial

Table 3.1 *(continued)*

Subscripts	
L, lat	Latent
leak	Leakage
loop	Loop system
m	Motor
man	Manometric head
max	Maximum
min	Minimum
mom	Momentary
n, night	Night time (from 18:00 to 06:00)
oa	Outdoor air
op	Operation
out	Outlet
p	Pump
REC	Recovery
rl	Room load
S, sens	Sensible
s	Seasonal
sa	Supply air
sat	Saturation
shaft	Shaft
t	Temperature
throttle	Throttle
tot	Total
tr	Transmission
x	Moisture
VAV	Variable air volume

Table 3.1 *(continued)*

Subscripts	
w	Water
wet	Wet
Abbreviations	
AHU	Air handling unit
CAV	Constant air volume
HVAC	Heating, ventilation and air conditioning
LCC	Life cycle costs
SFP	Specific fan power
VAV	Variable air volume

4 FANS

4.1 GENERAL

Fan energy is a large part of the energy usage of a ventilating system. A number of new terms, like LCC (Life Cycle Cost) and SFP (Specific Fan Power), have been introduced into the HVAC vocabulary as the economically and ecologically minded have begun making demands on the efficient use of energy and power in buildings.

The fan power is influenced by the overall efficiency of the fan, the resistances in the system and the air flow velocity through the unit and ductwork.

As a fan mounted in a casing is affected by the air velocity field, the performance and sound level will diverge from performance tests obtained from test results of a similar stand-alone fan. As such, the fan performance must be tested with the fan mounted in a fan section in compliance with EN 13053. This European Standard specifies requirements and testing for ratings and performance of air handling units as a whole.

A fan section can comprise of more than one fan; in this paper a single fan in the section will be assumed.

In an installation the electrical power to the motor of the fan is affected by the air flow to the power of around 3 if the fan speed is to be changed proportionally. It is therefore extremely important to ensure the accuracy of the stated air flow in a ventilating system.

Two of the most important parameters for fan control, and hence energy consumption, are the operating time and the use of VAV (variable air volume) instead of CAV (constant air volume). Correct settings of operating time and a control device, which enables prolongation of the normal occupation time for extraordinary use, gives energy savings, so does also the use of VAV which means that only the required air volume is supplied.

4.2 ELECTRICAL POWER

The absorbed power supplied from the mains to each individual fan can be expressed as follows:

$$P_{el} = \frac{q_V \cdot \Delta p_{fan}}{\eta_e \cdot 1000} \tag{4.1}$$

or

$$P_{el} = \frac{P_{shaft}}{\eta_{tr} \cdot \eta_m \cdot \eta_{aceq}} \tag{4.2}$$

where

P_{el}	=	The absorbed electrical power supplied from the mains (kW)
q_V	=	Air volume flow through the fan (m^3/s)
Δp_{fan}	=	Total pressure rise from the fan inlet to the outlet (Pa)
P_{shaft}	=	Mechanical power supplied to the fan shaft (kW)
η_e	=	Overall efficiency of the fan and motor system = $\eta_{shaft} \times \eta_{tr} \times \eta_m \times \eta_{aceq}$
η_{shaft}	=	Fan shaft efficiency (includes bearing losses)
η_{tr}	=	Efficiency of the mechanical transmission
η_m	=	Efficiency of the electric motor excluding any control
η_{aceq}	=	Efficiency of the control equipment including its effect on motor losses

All values are applicable to an air density of ρ_{air} = 1.2 kg/m^3
The fan performance must be tested with the fan mounted in a fan section (factors influenced are; Δp_{fan}, P_{shaft}, η_{shaft} and sound level) because there is a forced air velocity field in a unit which does not exist when the performance is tested as a stand-alone fan (see figure 4.1).

Figure 4.1 —Air velocity field in a fan section and around a stand-alone fan.

In a ducted air handling system two pressure drops, internal and external to the AHU, shall be established. Pressure rise across the fan Δp_{fan} must overcome the sum of those pressure drops. See figures 4.2 and 4.3.

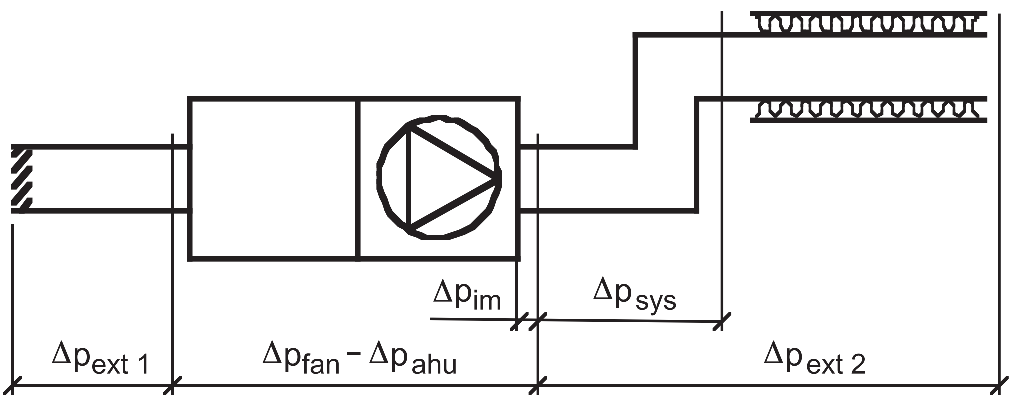


Figure 4.2 — Pressure conditions in the ducting system (external pressure drop)

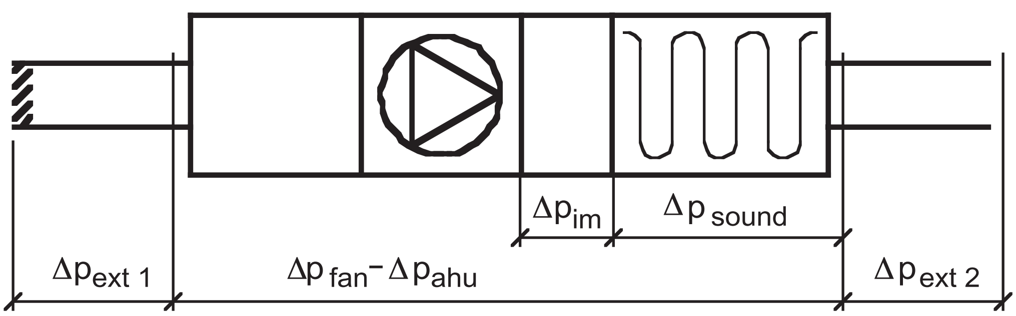


Figure 4.3 — Pressure conditions inside the air handling unit

The consultant or contractor has to determine the external pressure drop of the system and the designer of the air handling unit should calculate the internal pressure drop of the unit. Pressure losses should be assigned to that part of the system where they are actually generated. Particularly at the boundaries of the air handling unit pressure drops arise with indistinct designation. Considering this, the following subdivision in pressure drops can be defined.

Internal pressure drop:

- Pressure drop in all the functional sections of the air handling unit, including extra losses generated by the installation sequence of the components in the unit. The filter pressure drop should be calculated for the average of initial and recommended final pressure drop. The pressure drop across all components involved (e.g. heat exchangers, cooling coils and humidifiers) shall be calculated as the mean value dry and wet at design point.
- Pressure drop at the unit inlet caused by inefficient static regain of velocity pressure in intake opening/duct.
- Fan system effect caused by additional aerodynamic losses at fan inlet and discharge opening. The system effect depends on the fan arrangement in the air handling unit and ductwork connection on the fan discharge.
  The system effect generated in a discharge opening of a fan (Δp_{im}) with downstream components is comprised in the internal pressure losses. If the fan is located at the end of the unit and the fan outlet is the unit outlet; possible downstream system effects are not included in the internal pressure drop.

External pressure drop:

- Pressure drop in any duct system upstream and downstream the AHU including the losses generated in any integrated appurtenance (balancing dampers, fire dampers, grilles attenuators, etc.).
- Fan discharge system effect, generated by inappropriate ductwork (tees, elbows, or abrupt cross section changes) in the vicinity of the fan discharge. These external losses can only occur when the fan outlet is the unit outlet.
  Consult AHU-suppliers recommendations for proper ductwork connection.

The consultant or contractor has to supply the designer with the external pressure drop of the system. Externally generated additional pressure losses at the unit discharge connection are never included in the design internal pressure drop and are the responsibility of the contractor or consultant.

If the air handling unit includes a rotary heat exchanger, make sure that the leakage and purging air flows are from the supply air side to the extract air side. The necessary extra throttling ($\Delta p_{throttle}$) upstream of the rotary heat exchanger to secure the right leakage flow to the extract air side can be calculated as indicated in 4.2.1, and should be included in Δp_{ahu}.

When calculating the P_{el} of the extract air fan, the leakage and purging air volume flows should be included in the air volume flow of the extract fan, q_V.

Transmission efficiency, η_{tr}

The efficiency of the drive system, such as V-belt drives, flat belt drives, reduction gears, hydraulic couplings and eddy current couplings should be determined by applying figures specified by the supplier of the transmission.

If the fan impeller is directly mounted on the rotating motor shaft, specify $\eta_{tr} = 1$.

Motor efficiency, η_m

The efficiency of motors should be obtained from the motor manufacturers.

Efficiency of air flow control equipment η_{aceq}

For systems that permit the incorporation of variable speed devices such as frequency inverters or voltage regulators, the performance data of these devices should preferably be obtained from the supplier. The efficiency should include how the control equipment affects motor losses.

If the fans have controllable blades, pertinent performance data should be obtained from the supplier.

4.2.1 CALCULATION OF EXTRA THROTTLING ON EXTRACT AIR SIDE OF THE ROTARY HEAT EXCHANGER TO ENSURE THE CORRECT AIR LEAKAGE DIRECTION.

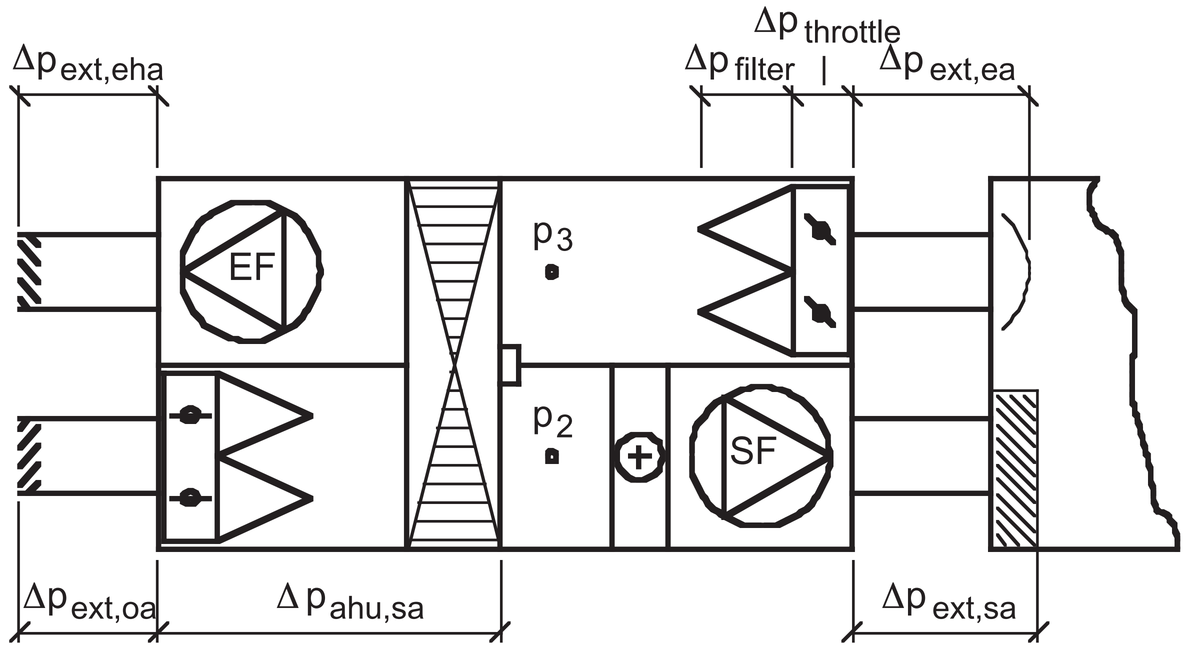


Figure 4.4 —Air handling unit with rotary heat exchanger and extra throttling

To ensure that the leakage across the rotary heat exchanger will be from the supply air to the extract air, the pressure p_3 must be lower than the pressure p_2 as illustrated above.

Hence: $p_3 \leq p_2$

$$p_2 = -(\Delta p_{ext,oa} + \Delta p_{ahu,sa}) \tag{4.3}$$

$$p_3 = -(\Delta p_{ext,ea} + \Delta p_{throttle} + \Delta p_{filter}) \tag{4.4}$$

If $p_2 = p_3$

then:

$$\Delta p_{throttle} = \Delta p_{ext,oa} + \Delta p_{ahu,sa} - \Delta p_{ext,ea} - \Delta p_{filter} \tag{4.5}$$

specify $\Delta p_{ext,oa} + \Delta p_{ext,sa} = \Delta p_{ext,oa+sa}$ and $\Delta p_{ext,ea} + \Delta p_{ext,eha} = \Delta p_{ext,ea+eha}$ thus

$$\Delta p_{thottle} = \Delta p_{ext,oa} + \Delta p_{ahu,sa} - \Delta p_{ext,ea+eha} + \Delta p_{ext,eha} - \Delta p_{filter} \tag{4.6}$$

The supplier of the air handling unit must calculate the Δp_{ahu} and Δp_{filter} used in the equations above.
The external pressure drops of the supply air, $\Delta p_{ext,oa+sa}$ and the external pressure drop of the extract air, $\Delta p_{ext,ea+eha}$ are specified in the quotation documents. On the other hand, $\Delta p_{ext,oa}$ and $\Delta p_{ext,eha}$ are seldom specified. In order to carry out a consistent calculation of $\Delta p_{throttle}$, the following settings can be assumed:

if $\Delta p_{ext,oa+sa}$ > 150 Pa set $\Delta p_{ext,oa}$ = 50 Pa, otherwise set $\Delta p_{ext,oa} = \Delta p_{ext,oa+sa} / 3$
if $\Delta p_{ext,ea+eha}$ > 150 Pa set $\Delta p_{ext,eha}$ = 50 Pa, otherwise set $\Delta p_{ext,eha} = \Delta p_{ext,ea+eha} / 3$

The degree of extra throttling required, $\Delta p_{throttle}$ can then be calculated from the formula above.

The pressure drop, Δp_{filer}, refers to a unit with filters at the design pressure drop.
Negative $\Delta p_{throttle}$ means that no extra throttling is necessary.

4.3 CONSTANT AIR VOLUME (CAV) vs. VARIABLE AIR VOLUME (VAV)

In a **CAV** system, the calculations related to energy consumption must be based on the nominal (maximum) air flow and nominal external pressure drop (pressure drop in the ducting).

In a **VAV** system, the calculations related to energy consumption must be based on an air flow, which is a mean annual value, the partial air flow, and the related external pressure drop, specified by the customer in each air handling unit specification or at another point in the reference documents of the inquiry.
The air handling unit supplier must therefore specify data at nominal (maximum) flow and nominal external pressure drop as well as the partial flow and the related external pressure drop.

If customer has not provided data concerning partial air flow and related external pressure drop, the following figures can be used in the energy calculations for VAV systems:

Design air flow:	65% of the nominal air flow
Design total pressure drop:	65% of the nominal total pressure drop.

Comments:
We have selected 65% of the nominal airflow rate, which we consider realistic as a mean annual value for normal comfort ventilation.
We have set the design total pressure drop at 65% of the pressure drop at nominal airflow rate using conventional calculation methods and assuming the following:
— 62% of the total pressure drop consists of the flow-dependent pressure drop
— 38% of the total pressure drop consists of the flow-independent pressure drop, equivalent to constant pressure control.
The reduction factor for variable air volume system; η_{VAV}, can then be calculated to 0,42.

4.4 ENERGY CALCULATIONS

$$W = \left(P_{el,sa} + P_{el,ea}\right) \cdot t_{op} \tag{4.7}$$

where

W	=	Annual energy consumption (kWh/a)
$P_{el,sa}$	=	The absorbed electric power supplied from the mains to the supply air fan (kW)
$P_{el,ea}$	=	The absorbed electric power supplied from the mains to the extract air fan (kW)
t_{op}	=	Annual operation time (h/a)

NOTE: In a CAV system $P_{el,sa}$ and $P_{el,ea}$ shall be calculated at nominal flow and nominal external pressure drop. In a VAV system $P_{el,sa}$ and $P_{el,ea}$ shall be calculated at 65% of nominal flow and 65% of nominal external pressure drop if nothing else is stated.

Rated values for $P_{el,sa}$ and $P_{el,ea}$ shall be specified in AHU manufacturer s specifications!

4.5 SPECIFIC FAN POWER (SFP)

The SFP value is defined as power divided by air flow. It can easily be proven that the value is also total pressure drop divided by overall efficiency of the fan and motor system and this indicates how to reduce the value.

$$SFP = \frac{P_{el}}{q_V} = \frac{\Delta p_{fan}}{\eta_e \cdot 1000} \tag{4.8}$$

where

SFP	=	The specific fan power of the air handling unit/fan [kW/(m³/s)]
P_{el}	=	The absorbed electric power supplied from the mains to the fan in the air handling unit/fan (kW)
q_V	=	Air flow through the air handling unit/fan (m³/s)
Δp_{fan}	=	Total pressure rise from the fan inlet to the outlet (Pa)
η_e	=	Overall efficiency of the fan

The demanded SFP value is intended for use during the planning stage for determining the absorbed electric power demand and the energy consumption required for transporting air. By stipulating a SFP value, the purchaser can quickly determine whether a given air handling unit will meet the overall demands on power consumption. If the air handling unit consists of only one air stream the demanded SFP value should be less than half the value of a unit with two air steams and heat recovery. To make it easier for a supplier to offer air handling units or fans with the desired power efficiency, it is suitable to specify the highest permissible SFP in connection with other fan performance data in the program specification for example as SFP ≤ 2.5 kW/(m³/s).

EUROVENT has defined the SFP_E (index E for Eurovent), which makes it possible to assess how efficiently individual air handling units utilize electric power.

4.5.1 HEAT RECOVERY AIR HANDLING UNIT WITH SUPPLY AIR AND EXTRACT AIR

The **specific fan power, SFP_E** is the total amount of electric power, in kW, supplied to the fans in the air handling unit, divided by the largest of supply air or extract air flow rate (note, not the outdoor air nor the exhaust air flow rates) expressed in m³/s.

$$SFP_E = \frac{P_{el,sa} + P_{el,ea}}{q_{V\,max}} \tag{4.9}$$

where

- SFP_E = Specific fan power of a heat recovery air handling unit [kW/(m³/s)]
- $P_{el,sa}$ = The absorbed electric power supplied from the mains to the supply air fan (kW)
- $P_{el,ea}$ = The absorbed electric power supplied from the mains to the extract air fan (kW)
- $q_{V\,max}$ = Largest of supply air or extract air flow through the air handling unit (m³/s)

Note that air handling units with liquid-coupled coil heat exchangers and separate supply air and extract air sections also belong to the category of air handling units described in 4.5.1.

$P_{el,sa}$ and $P_{el,ea}$ can be calculated as shown in Section 4.2.

4.5.2 SEPARATE SUPPLY AIR OR EXTRACT AIR HANDLING UNITS AND INDIVIDUAL FANS

The **specific fan power, SFP_E** is the electric power, in kW, supplied to a fan divided by the air flow expressed in m³/s.

$$SFP_E = \frac{P_{el}}{q_V} \tag{4.10}$$

where

- SFP_E = The specific fan power of the air handling unit/fan [kW/(m³/s)]
- P_{el} = The absorbed electric power supplied from the mains to the fan in the air handling unit/fan (kW)
- q_V = Air flow through the air handling unit/fan (m³/s)

P_{el} can be calculated as specified in Section 4.2.

5 HEATING AND COOLING

5.1 GENERAL

The function of a heat exchanger in an air handling unit is to transfer thermal energy from one medium to another. The energy costs for any form of heat transfer depends on many installation and building parameters. Examination and calculation of all these variables for each individual installation would take a lot of time and in some cases would even be impossible.
In these Recommendations realistic values have been established for the most common heating, ventilating and air-conditioning systems in Europe. The data specified in this chapter, shall only be applied if the actual installation is in compliance with the assumed stipulations.

5.2 COSTS OF THERMAL HEATING ENERGY

This chapter should be used as a guidance to calculate the costs of thermal heating energy if no actual prices in / kWh are available!
The price of thermal heating energy shall be established at the location of the consumer (heat exchanger) and depends on:

- <u>*Fuel price*</u>
- <u>*Overall efficiency of the boiler*</u> (heat output measured in water or steam leaving the boiler, divided by input)
  The table below lists overall efficiencies for various kinds of heat generation.

Table 5.1 Overall boiler efficiency at rated performance[1)]

TYPE OF ENERGY SOURCE	*ATMOSPHERIC BURNERS*	*POWER BURNERS[2)]*
Non condensing water boilers	81% [3)]	83% [3)]
Condensing water boilers	86% [3)]	87% [3)]
Steam boilers	80%	82%
Electric boilers (water & steam)	95%	
District heating	100%	

1) Efficiency based on gross calorific value (higher heating value) of boiler fuel
2) Boiler with combustion air fan
3) At 80 …C mean water temperature

- <u>*Seasonal efficiency of thermal heat production*</u>
  The seasonal efficiency is the actual operating efficiency that will be achieved during the heating season at various loadings.
  Apart from the overall efficiency of the energy source, the seasonal efficiency also depends on:
  - Type of boiler control (on-off, high-low-off, modulating, boiler sequence control).
  - Water temperature control strategy (constantly high water temperature during heating season, weather anticipating water temperature control system).
  - Extensiveness of heating distribution pipe network (design and type of heating system).
  Where no actual data for seasonal efficiency is available, table 5.2 recommends values for various types of installation.

Table 5.2 Seasonal efficiency of thermal heat production [1)									
INSTALLATION TYPE		**BURNER TYPE**							
		Atmospheric burners			***Power burners***			***Others***	
	Mean water temperature	***80°C***	***45°C***	***~***	***80°C***	***45°C***	***~***	***80°C***	***45°C***
Installation equipment in one plant room	Non condensing water boiler(s)	77%	80%	—	81%	84%	—	—	—
	Condensing water boiler(s)	83%	89%	—	85%	90%	—	—	—
	Steam boiler(s)	—	—	76%	—	—	80%	—	—
	Electric boiler(s)	—	—	—	—	—	—	93%	96%
	District heating	—	—	—	—	—	—	98%	99%
		Atmospheric burners			***Power burners***			***Others***	
	Mean water temperature	***80°C***	***45°C***	***~***	***80°C***	***45°C***	***~***	***80°C***	***45°C***
Central boiler [2) house with distribution pipe network to AHU s (distance < 250m)	Non condensing water boiler(s)	74%	78%	—	78%	82%	—	—	—
	Condensing water boiler(s)	81%	87%	—	82%	88%	—	—	—
	Steam boiler(s)	—	—	74%	—	—	77%	—	—
	Electric boiler(s)	—	—	—	—	—	—	90%	94%
	District heating	—	—	—	—	—	—	95%	97%
		Atmospheric burners			***Power burners***			***Others***	
	Mean water temperature	***80°C***	***45°C***	***~***	***80°C***	***45°C***	***~***	***80°C***	***45°C***
Central boiler [2) house with distribution pipe network to AHU s (distance > 250m)	Non condensing water boiler(s)	72%	77%	—	76%	80%	—	—	—
	Condensing water boiler(s)	78%	85%	—	79%	86%	—	—	—
	Steam boiler(s)	—	—	71%	—	—	75%	—	—
	Electric boiler(s)	—	—	—	—	—	—	87%	92%
	District heating	—	—	—	—	—	—	92%	95%

[1) Efficiency based on gross calorific value (higher heating value) of boiler fuel
[2) Distribution network indoor or outdoor installed

With the seasonal efficiency and the fuel price or price for electricity the costs of thermal heating energy can be calculated with the following equations:

- ➤ Fuel fired boilers: $p_{heating} = \left(E_f \cdot 3600\right) / \left(H_h \cdot \eta_{sh}\right)$ (5.1)
- ➤ Electric boilers: $p_{heating} = \left(E_{el} / \eta_{sh}\right)$ (5.2)
- ➤ District heating: $p_{heating} = \left(E_{dis} / \eta_{sh}\right)$ (5.3)

where

$p_{heating}$ = costs of thermal heating energy in €/kWh
E_f = fuel price per unit in €/m³ or €/kg or €/l
H_h = higher heating value of boiler fuel in kJ/m³ or kJ/kg or kJ/l
η_{sh} = seasonal efficiency of thermal heat production in %/100
E_{el} = price for electricity in €/kWh
E_{dis} = price for metered district heating in €/kWh
3600 = conversion factor in kJ/kWh

5.3 COSTS OF THERMAL COOLING ENERGY

This chapter should only be used as a guidance to assess the costs of thermal cooling energy if no actual prices in € / kWh are available!

The price of thermal cooling energy shall also be established at the location of the consumer (cooling exchanger) and depends on:

- *Price of primary energy* (electricity, fuel or other energy source)
- *Overall efficiency (C.O.P.-value) of refrigeration plant* (cooling capacity measured in water or refrigerant leaving the refrigerating machine, divided by total primary input)
  A complete refrigeration plant usually consists of several components. Since each component may be combined with various kinds of equipment, a great variety of refrigeration plants is applied in the HVAC-industry. Apart from the composition of the cold production, the performance with matching efficiency also depends on refrigerant type, chilled water or refrigerant temperatures, condenser water temperatures or outdoor temperature.
  The great number of options makes it impossible to provide efficiency figures for all circumstances. Table 5.3 shows total efficiencies of most commonly used refrigeration plants at rated standard performance conditions.

Table 5.3 Overall refrigeration plant efficiency in % at rated performance

TYPE OF REFRIGERATION PLANT	EVAPORATION CONDITIONS		CONDENSER CONDITIONS						
	Evaporation temperature °C	Chilled water temperatures °C	Air temperature [1] °C				Water temperatures °C		
			25	30	35	40	30/35	35/40	40/45
Air-cooled compressor-condensing unit for direct-expansion coils	4° 8° 12°	- - -	340 360 380	300 320 340	270 285 305	240 260 285	- - -	- - -	- - -
Water chillers with air-cooled condensers	- - -	5° / 10° 7° / 12° 10° / 15°	310 320 335	280 290 300	250 260 270	225 235 240	- - -	- - -	- - -
Water chillers with (waste) water-cooled condensers [2]	- - -	5° / 10° 7° / 12° 10° / 15°	- - -	- - -	- - -	- - -	365 385 410	320 340 360	280 295 315
Water chillers with water-cooled (circulation) condensers [3]	- - -	5° / 10° 7°/ 12° 10° / 15°	- - -	- -	- - -	- - -	305 315 330	270 285 295	240 250 265
Hot water absorption refrigerating machine [4] [5]	- - -	5° / 10° 7° / 12° 10° / 15°	- - -	- - -	- - -	- - -	- 75 -	- - -	- - -
Direct fired absorption refrigerating machine [5] [6]	- - -	5° / 10° 7° / 12° 10° / 15°	- - -	- - -	- - -	- - -	Data to be specified by manufacturer or supplier		

1) Ambient air temperature at condenser inlet.
2) Cooling from natural source water or utility water. Water consumption not included in efficiency!
3) Water cooled condensers of water chillers consume additional energy if the cooling water is coming from an air cooled liquid cooler or cooling tower. The listed efficiencies include the energy consumption of a liquid cooler and circulating pump between condenser and liquid cooler! Assumed ambient air temperatures 10 K below water outler temper ature of liquid cooler.
4) Efficiency of thermal heat production not included in refrigeration efficiency (see chapter 7.1.1)
5) Absorbed electric power 0,5% of rated cooling performance
6) Efficiency based on higher heating value of fuel

¥ *Seasonal efficiency of cold production*

The seasonal efficiency can be defined as the actual operating efficiency that can be achieved during the cooling season at various loadings and matching outdoor conditions.

Although this figure is mainly dominated by the overall refrigeration plant efficiency, the seasonal efficiency is also affected by:
- Type of controls on cooling equipment (on-off, high-low-off, modulating, chiller sequence control)
- Relative location of cooling components, with respect to connecting pipe network.
- Extensiveness of chilled water or refrigerant distribution pipe system.

Where no actual data for seasonal efficiency is available, table 5.4 illustrates recommended values for various types of refrigeration plants at several mean operational conditions.

Table 5.4 Seasonal efficiency of cold production
Integrated part load values for different types of cold production in development Relevant data will be established by European working group For the time being figures from table 5.3 should be applied

With the seasonal efficiency of table 5.4 and the price for electricity, price for thermal heating energy (⁄ 5.1) or fuel price; the costs of thermal cooling energy can be calculated with the following equations:

➤ Compression type refrigerating machine: $$p_{cooling} = \left(E_{el} / \eta_{sc}\right) \quad (5.4)$$

➤ Hot water absorption refrigerating machine: $$p_{cooling} = \left(p_{heating} / \eta_{sc}\right) \quad (5.5)$$

➤ Direct fired absorption refrigerating machine: $$p_{cooling} = \left(E_f \cdot 3600\right) / \left(H_h \cdot \eta_{sc}\right) \quad (5.6)$$

➤ District cooling: $$p_{cooling} = \left(E_{dis} / \eta_{sc}\right) \quad (5.7)$$

where

$p_{cooling}$ = costs of thermal cooling energy in €/kWh
E_{el} = price for electricity in €/kWh
η_{sc} = seasonal efficiency of thermal cold production in %/100
$p_{heating}$ = costs of thermal heating energy in €/kWh
E_f = fuel price per unit in €/m^3 or €/kg or €/l
H_h = higher heating value of used fuel in kJ/m^3 or kJ/kg or kJ/l
E_{dis} = price for metered district cooling in €/kWh
3600 = conversion factor in kJ/kWh

5.4 HEATING COILS

Air-heating coils are used to heat the air under forced convection. Related to the applied heating medium the coils can be categorised into the following types:
- water coils
- steam coils
- electric heating coils

Energy can be consumed on the primary side and/or air side of the coil. In all cases, however, thermal heating energy is consumed.

5.4.1 ENERGY CONSUMPTION FOR THE PRIMARY SIDE

The energy consumption for the primary (water-, steam,- electric-) side is considered to be the required energy to convey thermal heating energy from the source to the consumer. The primary energy consumption to be taken into account basically depends on the type of heating coil.

¥ *Water coils*

The water flow through the coil is maintained by one or more circulating pumps. The hydraulic design of the heating distribution pipe network and the water-side pressure drop across the coil determines the energy consumption of the circulating pump(s).

The energy consumption of the pumps is also affected by:
- efficiency of the pump unit (size related)
- type of coil-control (variable water flow, variable water temperature)
- speed control on circulating pump(s)

Due to the number of parameters involved a simplification is made by assuming that the energy consumption is proportional to the coil- and pump flow rate.
The basic equation to determine the annually consumed electric energy is given below:

➤ Consumed annual electric energy for circulation: $W_{circ} = \left(P_{circ} / 1000\right) \cdot t_{eq,op}$ (5.8)

where

W_{circ} = consumed annual electric energy for circulation in kWh/a
P_{circ} = absorbed electric power of heating coil related pump(s) in W
$t_{eq,op}$ = equivalent running time of pump(s) at full load in h/a

The absorbed electric power P_{circ} shall be determined as described below; unless specified data is available.
Where real data is unknown, the equivalent running time $t_{eq,op}$ of circulating pump(s) is assumed to be equal to the operating time of the air handling unit.
For pumps with speed control or in case of optimised heating plants a 50% reduction in running time shall be allowed for.

➤ Hence: $W_{circ} = \left(P_{circ} / 1000\right) \cdot 0{,}5 \cdot t_{op}$ (5.9)

where

W_{circ} = consumed annual electric energy for circulation in kWh/a
P_{circ} = absorbed electric power of heating coil related pump(s) in W
t_{op} = operating time of the air handling unit in h/a

An assessment of the absorbed electric power of pump(s) for water circulation through the coil can be established with the equation below.

➤ Absorbed power for water circulation: $P_{circ} = q_{v\ coil} \cdot \left(2 \cdot \Delta p_{coil} + \Delta p_{distr}\right) / \eta_p$ (5.10)

where

P_{circ} = absorbed electric power of heating coil related pump(s) in W
$q_{v\ coil}$ = water flow through the coil in l/s
Δp_{coil} = water side pressure drop across the coil in kPa
Δp_{distr} = coil related pressure drop in heating system in kPa
η_p = total efficiency of pump(s) and electric motor(s) in %/100

The water flow rate through the coil ($q_{v\ coil}$) and water side pressure drop (Δp_{coil}) shall be taken from the air handling unit specification.Guidelines for the pressure drop in the pipe section to the coil and the pump efficiency are shown in table 5.5.

Table 5.5 Pressure drop and efficiency guidelines for circulation pumps for heating coils

PRESSURE DROP IN HEATING SYSTEM		EFFICIENCY OF PUMP UNIT	
Distance between energy source and heating coil in [m]	Δp_{distr} in [kPa]	$q_{v,sol} \times (2.\Delta p_{coil} + \Delta p_{distr})$ [Watt]	η_p [%]
d ≤ 10	8	Pcirc x η_p ≤ 10	15
10 < d ≤ 16	9	10 < Pcirc x η_p ≤ 16	25
16 < d ≤ 25	11	16 < Pcirc x η_p ≤ 25	30
25 < d ≤ 40	15	25 < Pcirc x η_p ≤ 40	34
40 < d ≤ 63	20	40 < Pcirc x η_p ≤ 63	37
63 < d ≤ 100	30	63 < Pcirc x η_p ≤ 100	39
100 < d ≤ 125	36	100 < Pcirc x η_p ≤ 160	41
125 < d ≤ 160	45	160 < Pcirc x η_p ≤ 250	43
160 < d ≤ 200	55	250 < Pcirc x η_p ≤ 400	45
200 < d ≤ 250	65	400 < Pcirc x η_p ≤ 630	48
250 < d ≤ 315	85	630 < Pcirc x η_p ≤ 1000	51
315 < d ≤ 400	105	1000 < Pcirc x η_p ≤ 1600	53
400 < d ≤ 500	130	1600 < Pcirc x η_p ≤ 2500	54
500 < d ≤ 630	160	2500 < Pcirc x η_p ≤ 4000	55
630 < d ≤ 800	200	4000 < Pcirc x η_p ≤ 6300	57
800 < d ≤ 1000	250	6300 < Pcirc x η_p ≤ 10000	61

¥*Electric heating coils*

Primary side energy losses for electric heating coils are caused by heat generation in the power supply cables. The current to the heating element and the electric resistance of the cables determines the loss. For safety reasons, the voltage drop with subsequent heat production in the electric cables is limited by National and International Standards and Regulations.

Table 5.6 gives appropriate values to assess the energy consumption on the primary side of electric heating coils; based on moderate cable diameters and voltage drops.

Table 5.6 Primary energy losses in power supply cables to electric heaters in % of heater capacity

HEATER CAPACITY	DISTANCE BETWEEN POWER PANEL AND ELECTRIC HEATER								
kW	10m	16m	25m	40m	63m	100m	160m	250m	400m
10	0,7	1,2	1,8	2,9	4,6	5,0	5,0	5,0	5,0
12,5	0,7	1,2	1,8	2,8	4,4	5,0	5,0	5,0	5,0
16	0,7	1,1	1,8	2,8	4,4	5,0	5,0	5,0	5,0
20	0,6	0,9	1,4	2,2	3,5	5,0	5,0	5,0	5,0
25	0,5	0,7	1,1	1,8	2,9	4,6	5,0	5,0	5,0
31,5	0,4	0,6	0,9	1,4	2,2	3,5	5,0	5,0	5,0
40	0,3	0,5	0,7	1,1	1,7	2,7	4,4	5,0	5,0
50	0,3	0,4	0,7	1,1	1,7	2,7	4,4	5,0	5,0
63	0,3	0,4	0,7	1,1	1,7	2,7	4,4	5,0	5,0
80	0,2	0,3	0,4	0,7	1,1	1,8	2,8	4,4	5,0
100	0,2	0,3	0,4	0,6	1,0	1,6	2,5	3,9	5,0
125	0,2	0,2	0,4	0,6	0,9	1,5	2,3	3,6	5,0
160	0,2	0,2	0,4	0,6	0,9	1,5	2,3	3,6	5,0
200	0,1	0,2	0,3	0,5	0,7	1,2	1,9	3,0	4,7
250	0,1	0,2	0,2	0,4	0,6	0,9	1,5	2,3	3,7
315	0,1	0,1	0,2	0,4	0,5	0,9	1,4	2,2	3,5
400	0,1	0,1	0,2	0,4	0,5	0,9	1,4	2,2	3,5

The consumed additional electric energy on the primary side of an electric heating coil is calculated with the equation:

$$W_{el} = (PEL/100) \cdot Q_{H,an} \tag{5.11}$$

where

W_{el} = annual electric energy consumption on the primary side of an electric heating coil (energy losses in power supply cables) in kWh/a
PEL = primary energy loss according to table 5.6 in %
$Q_{H,an}$ = annual thermal energy consumption of the electric heating coil in kWh/a (see ⁄ 5.4.3)

• *Steam coils*
Primary energy consumption for steam coils is considered to be included in the costs of thermal heating energy. The steam pressure in the boiler will maintain the steam- and condensate flow.

5.4.2 ENERGY CONSUMPTION FOR THE AIR SIDE

To create the required air flow over a coil a certain differential pressure across the coil shall be maintained by the fan in the air handling unit. The energy consumption involved is included in the calculated consumption of electric energy of the fan.

5.4.3 CONSUMPTION OF THERMAL HEATING ENERGY

To establish the thermal energy consumption 3 basic extract and supply air temperature scenarios are presented for 3 different types of installation. The indicated supply temperatures could serve as typical values for the particular installation type but are not mandatory. Extract air temperatures however are supposed to be in accordance with the indicated values!

• *Installation type I*
- Central ventilation system
- Additional local heating and/or cooling in case of heating and/or cooling load

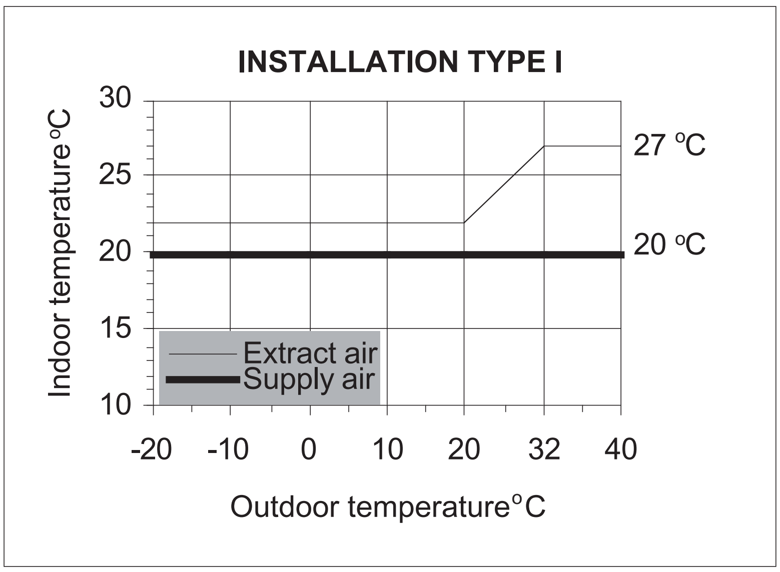


Figure 5.1 —Temperature scenario, type I

The function of the air handling unit(s) in this installation type is merely to supply the quantity of fresh air into the building at the desired supply temperature. Air handling units for this installation type are not usually equipped with a cooling coil. Heating and/or cooling loads are not covered by the air handling unit.
If the unit is equipped with a cooling coil; some cooling capacity is obtained from a differential temperature between room and supply air. Typical values for temperatures of supply and extract air are often in compliance with the graph on the left.

• *Installation type II*
- Central ventilation and cooling system
- Additional local heating in case of heating load
- Additional local cooling in case of high cooling load

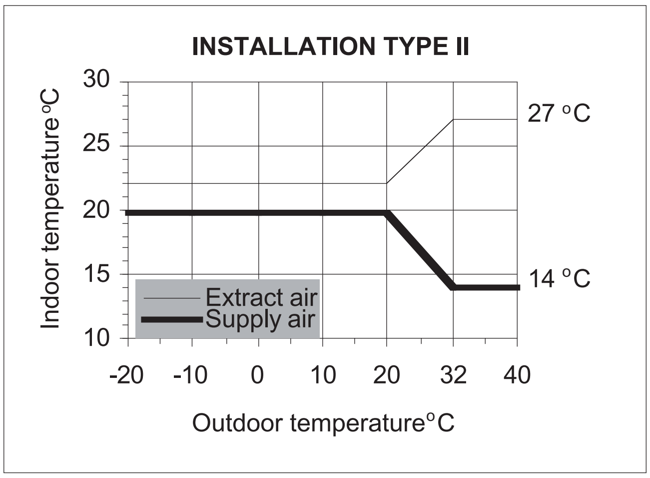


Figure 5.2 —Temperature scenario, type II

In this type of installation the air handling unit supplies the air in such a condition that moderate cooling loads are compensated for by the temperature difference between room and supply air. Heat losses are not covered by the air handling unit.
The adjacent graph illustrates a typical relationship between outdoor temperature and supply- and extract air temperature. Supply temperatures may be changed, if appropriate.
The indicated extract temperatures on the other hand are supposed to be fixed values.

• *Installation type III*
- Central air conditioning system for ventilation, heating and cooling
- Additional local heating in case of high heating load
- Additional local cooling in case of high cooling load

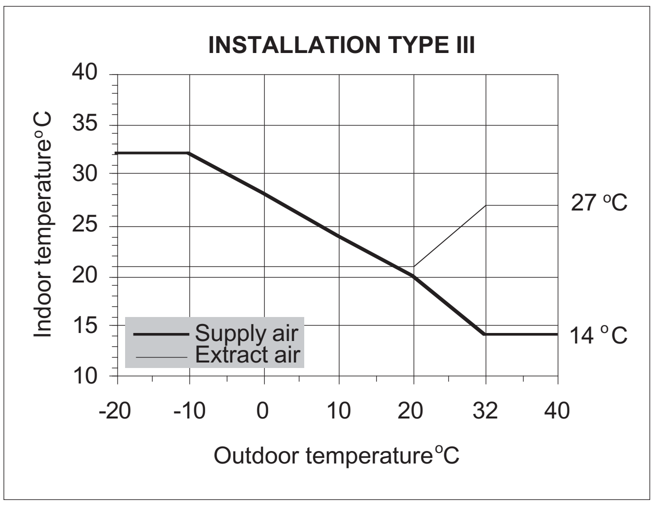


Figure 5.3 —Temperature scenario, type III

The air handling unit in this type of installation ventilates and conditions the building completely. The air is supplied in such a condition that moderate heating and cooling loads are covered by the temperature difference between room and supply air. For extreme loads additional local heating or cooling may be applied.
A typical temperature scenario for this installation type is shown in the graph on the left. The indicated extract temperatures are fixed. The supply temperature scenario however may be adapted by altering the supply temperatures at —10¡C, +20¡C and the maximum outdoor temperature.

The momentary thermal energy consumption of a heating coil is calculated with the equation:

$$Q_{mom} = q_v \cdot \rho \cdot (h_{out} - h_{in}) \cdot t_s \qquad (5.12)$$

where

Q_{mom} = energy consumption for a time period t_s in kJ
q_v = air flow rate over the coil in m^3/s
ρ = density of the considered air flow rate in kg/m^3
h_{out} = enthalpy of the air at the outlet of the coil in kJ/kg
h_{in} = enthalpy of the air at the inlet of the coil in kJ/kg
t_s = period of time with steady in- and outlet conditions in s

Since the moisture content of the air does not change, (h_{out} —h_{in}) may be substituted by c_p x (t_{out} —t_{in}):

$$Q_{mom} = q_v \cdot \rho \cdot c_p \cdot (t_{out} - t_{in}) \cdot t_s \qquad (5.13)$$

where

Q_{mom} = energy consumption for a time period t_s in kJ

q_v = air flow rate over the coil in m^3/s

ρ = density of the considered air flow rate in kg/m^3

c_p = specific heat of the air in kJ/kg·¡C

t_{out} = temperature of the air leaving the coil in ^oC

t_{in} = temperature of the air entering the coil in ^oC

t_s = period of time with steady in- and outlet conditions in s

Strictly considered the sensible heat increase of the water vapour is disregarded in the last formula. This is an acceptable assumption for normal applications in comfort installations where the moisture content of the air is relatively small.

The annual thermal energy consumption of a heating coil shall be established taking into account the following factors:

- geographic location of the air handling unit with matching outdoor conditions
- the effectiveness of any heat recovering device characterised by its dry temperature efficiency
- the selected appropriate temperature scenario
- the operating time of the unit

A correct valuation of the aforementioned factors enables an accurate calculation of the momentarily thermal energy consumption for the duration of any outdoor condition during the operating time of the unit. The annual energy consumption of the heating coil then is the sum of the momentary energy consumptions according to:

$$Q_{H,an} = \sum_{i=Q_{mom,1}}^{i=Q_{mom,n}} \left(Q_{mom,i} / 3600\right) \qquad (5.14)$$

where

$Q_{H,an}$ = annual thermal energy consumption of the heating coil in kWh/a

$Q_{mom,i}$ = energy consumption during time $t_{s\ i}$ in kJ/a

$t_{s,i}$ = period of time with fixed in- and outlet conditions during annual operation time of unit in s

3600 = conversion factor in kJ/kWh

For further information on this topic see also chapter 8.

5.5 COOLING COILS

Cooling coils are used for air cooling under forced convection, with or without accompanying dehumidification. The dehumidification rate depends on the coil construction, the dew point of the entering air, the water flow and the water temperatures. The sensible heat ratio of the coil, defined as the ratio of sensible heat to total heat removal, can be used to evaluate the dehumidification rate. Based on the applied cooling fluid the coils can be divided into the following types:

- water coils
- direct expansion coils

To avoid freezing in cold climates during winter time, a coolant based on a mixture of water and antifreeze, may be applied in water coils.

Similar to heating coils, energy may be consumed on the primary side and/or air side of the coil. In all cases however thermal cooling energy is consumed.

5.5.1 ENERGY CONSUMPTION ON PRIMARY SIDE

The energy consumption on the primary (water-, refrigerant-) side is considered to be the required energy to convey thermal cooling energy from the source to the consumer. The primary energy consumption to be taken into account, principally depends on the type of coil.

• *Water (coolant) coils*

The water circulation through the coil is achieved by one or more circulating pumps. The hydraulic design of the chilled water distribution system and the water-side pressure drop across the coil (and refrigerating machine) determine the energy consumption of the circulating pump(s).
The energy consumption of the pumps is also affected by:
- efficiency of the pump unit (size related)
- type of coil-control (variable water flow, variable water temperature)
- speed control on circulating pump(s)

Due to the number of parameters involved a simplification is made by assuming that the energy consumption is proportional to the coil- and pump flow rate.
The basic equation to determine the annually consumed electric energy is given below:

➤ Consumed annual electric energy for circulation: $W_{circ} = \left(P_{circ} / 1000\right) \cdot t_{eq,op}$ (5.15)

where

W_{circ} = consumed annual electric energy for circulation in kWh/a
P_{circ} = absorbed electric power of cooling coil related pump(s) in W
$t_{eq,op}$ = equivalent running time of pump(s) at full load in h/a

The absorbed electric power P_{circ} shall be determined as described below; unless specified data is available.
Where real data is unknown, the equivalent running time $t_{eq,op}$ of circulating pump(s) is assumed to be equal to the operating time of the air handling unit.
For pumps with speed control or in case of optimised cooling plants a 50% reduction on running time shall be allowed for.

➤ Hence: $W_{circ} = \left(P_{circ} / 1000\right) \times 0{,}5 * t_{op}$ (5.16)

where

W_{circ} = consumed annual electric energy for circulation in kWh/a
P_{circ} = absorbed electric power of cooling coil related pump(s) in W
t_{op} = operating time of the air handling unit in h/a

An assessment of the absorbed electric power of pump(s) for water circulation through the coil can be established with the equation below.

➤ Absorbed power for water circulation: $P_{circ} = q_{v\ coil} \times \left(2 * \Delta p_{coil} + \Delta p_{distr}\right) / \eta_p$ (5.17)

where

P_{circ} = absorbed electric power of cooling coil related pump(s) in W
$q_{v\ coil}$ = water flow through the coil in l/s
Δp_{coil} = water side pressure drop across the coil in kPa
Δp_{distr} = coil related pressure drop in chilled water system in kPa
η_p = total efficiency of pump(s) and electric motor(s) in %/100

The water flow through the coil ($q_{v\ coil}$) and water side pressure drop (Δp_{coil}) shall be taken from the air handling unit specification.
Guidelines for the pressure drop in the chilled water pipe section to the coil and the pump efficiency are shown in table 5.7

Table 5.7 Pressure drop and efficiency guidelines for circulation pumps for cooling coils			
PRESSURE DROP IN CHILLED WATER SYSTEM		EFFICIENCY OF PUMP UNIT	
Distance between refrigerating machine and cooling coil in [m]	Δp_{dstr} in [kPa]	$q_{v,coil} \times (2.\Delta p_{coil} + \Delta p_{dstr})$ [Watt]	η_p [%]
d ≤ 10	22	25 < Pcirc x η_p ≤ 40	34
10 < d ≤ 16	27	40 < Pcirc x η_p ≤ 63	37
16 < d ≤ 25	32	63 < Pcirc x η_p ≤ 100	39
25 < d ≤ 40	39	100 < Pcirc x η_p ≤ 160	41
40 < d ≤ 63	48	160 < Pcirc x η_p ≤ 250	43
63 < d ≤ 100	60	250 < Pcirc x η_p ≤ 400	45
100 < d ≤ 125	70	400 < Pcirc x η_p ≤ 630	48
125 < d ≤ 160	80	630 < Pcirc x η_p ≤ 1000	51
160 < d ≤ 200	95	1000 < Pcirc x η_p ≤ 1600	53
200 < d ≤ 250	110	1600 < Pcirc x η_p ≤ 2500	54
250 < d ≤ 315	130	2500 < Pcirc x η_p ≤ 4000	55
315 < d ≤ 400	155	4000 < Pcirc x η_p ≤ 6300	57
400 < d ≤ 500	180	6300 < Pcirc x η_p ≤ 10000	61
500 < d ≤ 630	215	10000 < Pcirc x η_p ≤ 16000	64
630 < d ≤ 800	260	16000 < Pcirc x η_p ≤ 25000	67
800 < d ≤ 1000	315	25000 < Pcirc x η_p ≤ 40000	69

¥*Direct expansion coils*

The primary energy consumption required to maintain the refrigerant flow through the coil shall be included in the costs of thermal cooling energy. The compressor in the refrigerating machine creates the differential pressure across the coil and subsequent refrigerant pipe network.

5.5.2 ENERGY CONSUMPTION ON AIR SIDE

To create the required air flow over a coil a certain differential pressure across the coil shall be maintained by the fan in the air handling unit. The energy consumption involved is included in the calculated consumption of electric energy of the fan.

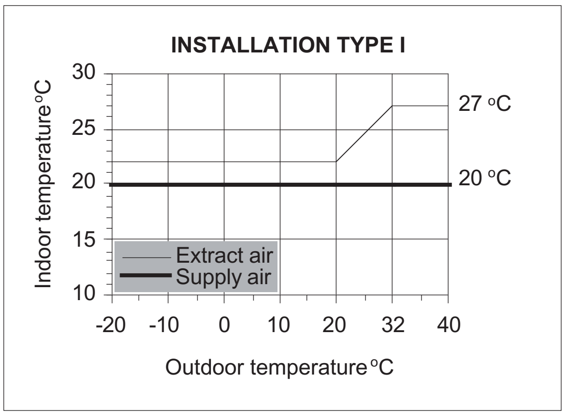
5.5.3 CONSUMPTION OF THERMAL COOLING ENERGY

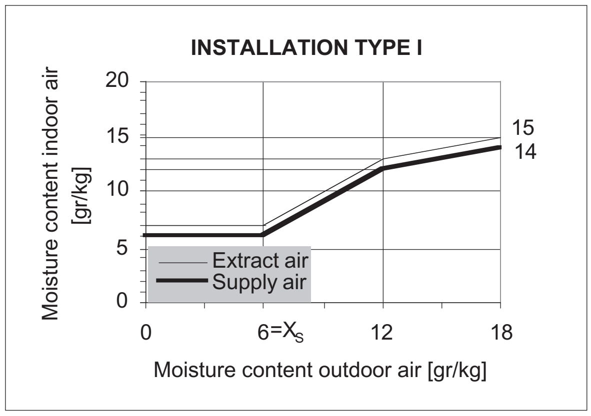
The thermal energy consumption of a cooling coil is not only affected by entering and leaving air temperatures but also by the moisture content of in- and outlet air, in those cases where the water inlet temperature (or evaporation temperature) is below the dew point temperature of the entering air (dehumidification).

To calculate the sensible thermal cooling energy consumption 3 basic extract and supply air temperature scenarios are presented for 3 different types of installation. The indicated supply temperatures could serve as typical values for the particular installation but are not mandatory. Extract air temperatures, however, should be in accordance with the indicated values.

For the computation of the latent cooling energy consumption the corresponding basic moisture scenarios are provided. For comfort installations, which are controlled on temperature, the moisture content of the supply air during the cooling period is physically related to the supply temperature and the maximum moisture content of the outdoor air for the location of interest!
This relationship will be explained later.

• <u>*Installation type I*</u>
- Central ventilation system
- Additional local heating and/or cooling in case of heating and/or cooling load

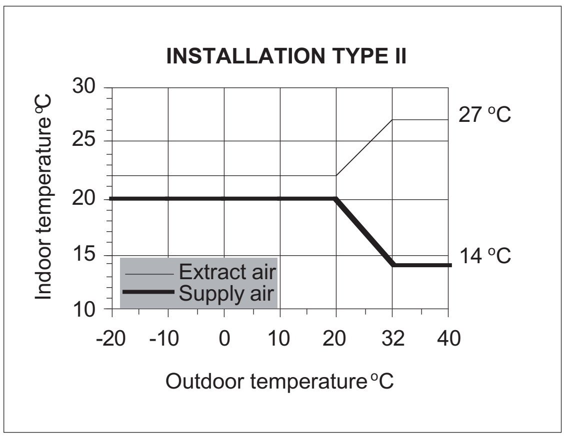


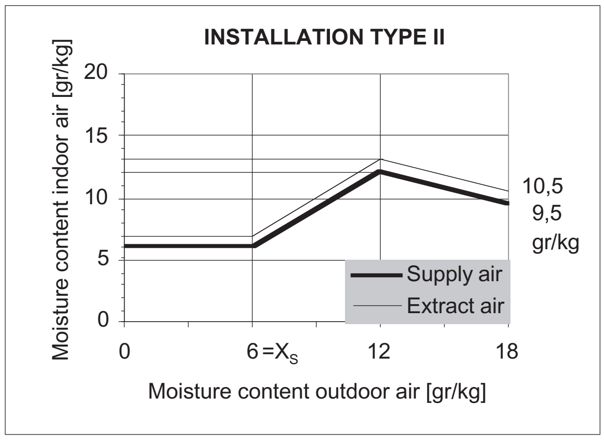


Temperature scenario ***Moisture scenario***

Figure 5.4 —Temperature and moisture scenario, installation type I

• <u>*Installation type II*</u>
- Central ventilation and cooling system
- Additional local heating in case of heating load
- Additional local cooling in case of high cooling load

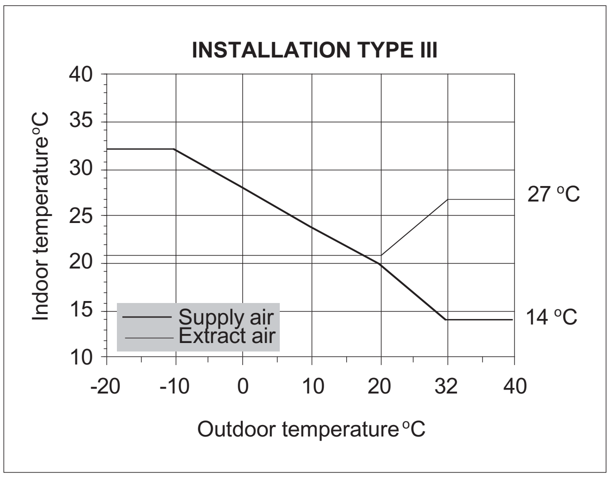




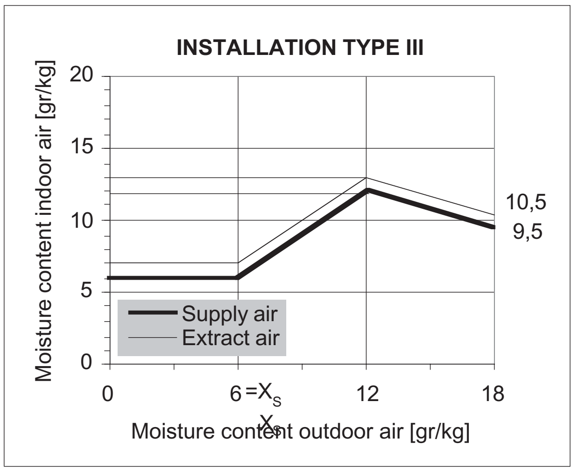
Temperature scenario ***Moisture scenario***

Figure 5.5 —Temperature and moisture scenario, installation type II

• *Installation type III*
- Central air conditioning system for ventilation, heating and cooling
- Additional local heating in case of high heating load
- Additional local cooling in case of high cooling load



Temperature scenario



Moisture scenario

Figure 5.6 —Temperature and moisture scenario, installation type III

These installation types have been explained in chapter 5.4.3.
The humidity scenarios are valid for both humidification and dehumidification.
Installation types II & III have identical humidity scenarios if the temperature scenarios for summer operation are the same.
The difference in absolute humidity between extract and supply air has been fixed on 1 g/kg of dry air, caused by internal moisture gains.

As previously stated; the moisture scenario for latent cooling is related to the selected temperature scenario for summer operation. The following considerations explain how the appropriate moisture scenario for the cooling period shall be defined in accordance with the selected temperature scenario for the cooling period.

Since high outdoor moisture content generally corresponds to high outdoor temperatures; an inversely proportional relationship between moisture content of outdoor air and supply air has been assumed from the point where dehumidification starts.

➤ The first important point to take into consideration is at what outdoor temperature dehumidification is likely to commence.
It is obvious that the selected supply air temperature at 20¡C outdoor temperature is determinative for this value!
If one selects e.g. $(t_{sa}\text{-}1)_{max,C}$ =18¡C; the maximum moisture content is 13 g/kg. The scenario dictates that outdoor air of 20¡C with any moisture content, has to be cooled down to 18¡C in the cooling coil. Although there is no physical need for dehumidifying the outdoor air as long as the moisture content is below 13 g/kg; some dehumidification will already occur at lower moisture content in the outdoor air, because a part of the external coil surface will have a temperature below the dew point of the entering air! Therefore it is realistic to assume that for this case dehumidification is likely to commence at an outdoor moisture content of 11 g/kg.
Similar reasoning for lower supply temperatures at 20 ¡C outdoor temperature lead to lower values for the outdoor moisture content where dehumidification starts.
It is obvious that for this particular point the moisture content of the outdoor air is equal to the moisture content of the supply air since the dehumidifying process just starts.

➤ The second important point in the moisture scenario for the cooling period is the moisture content of the supply air at maximum outdoor moisture content for the location of interest.

The moisture content of the supply air at maximum outdoor humidity is considered to be more or less equal to the moisture content of saturated supply air at rated minimum temperature.

An extra table in chapter 8 or Annex A recommends values for the moisture content of the supply air at 20¡C outdoor temperature and maximum outdoor temperature, depending on the corresponding values of the desired supply air temperature.
Stipulated figures are in accordance with practical values, established for commonly operating cooling coils.
The conditions of the supply air (temperature and moisture content) at 20¡C outdoor temperature, plotted in the Mollier diagram, are in between 80% and 90% relative humidity (the lower the supply temperature, the higher the relative humidity). The plotted values represent the assumed maximum relative humidity of the supply air at initial condensation!
The same exercise for the supply condition at maximum outdoor temperature shows air conditions around 95% relative humidity. These conditions represent the maximum relative humidity of the supply air at minimum temperature!

For additional information on temperature and humidity scenarios see also chapter 8 and Annex A of this document.
The method to determine the annual thermal cooling energy has been split up in two parts

5.5.3.1 ENERGY CONSUMPTION FOR SENSIBLE COOLING

The momentary thermal energy consumption for sensible cooling of air in a cooling coil is calculated with the equation:

$$Q_{mom} = q_v \cdot \rho \cdot (h_{in} - h_{out}) \cdot t_s \tag{5.18}$$

where

Q_{mom} = energy consumption for a time period t_s in kJ
q_v = air flow rate over the coil in m^3/s
ρ = density of the considered air flow rate in kg/m^3
h_{in} = enthalpy of the air at the inlet of the coil in kJ/kg
h_{out} = enthalpy of the air at the outlet of the coil in kJ/kg
t_s = period of time with steady in- and outlet conditions in s

Since the moisture content of the air in the case of <u>sensible</u> cooling does not change, (h_{in} — h_{out}) may be substituted by c_p x (t_{in} — t_{out}):

$$Q_{mom} = q_v \cdot \rho \cdot c_p \cdot (t_{in} - t_{out}) \cdot t_s \tag{5.19}$$

where

Q_{mom} = energy consumption for a time period t_s in kJ
q_v = air flow rate over the coil in m^3/s
ρ = density of the considered air flow rate in kg/m^3
c_p = specific heat of the air in kJ/kg•¡C
t_{in} = temperature of the air entering the coil in °C
t_{out} = temperature of the air leaving the coil in °C
t_s = period of time with steady in- and outlet conditions in s

Strictly considered the sensible heat decrease of the water vapour is disregarded in the last formula; which is allowed for normal applications in comfort installations where the moisture content of the air is relatively small.
The annual thermal energy consumption of a cooling coil for sensible cooling shall be established taking into account the following factors:
- geographic location of the air handling unit with matching outdoor conditions
- the effectiveness of any energy recovering device mainly characterised by its dry temperature efficiency

- the selected appropriate temperature scenario
- the operating time of the unit

A correct valuation of the aforementioned factors enables an accurate calculation of the momentarily thermal energy consumption for the duration of any outdoor condition during the operating time of the unit. The annual energy consumption of the cooling coil then is the sum of the momentary energy consumptions for sensible cooling according to:

$$Q_{C,S,an} = \sum_{i=Q_{mom,1}}^{i=Q_{mom,n}} \left(Q_{mom,i} / 3600 \right) \tag{5.20}$$

where

$Q_{C,S,an}$	=	annual thermal energy consumption of the cooling coil for sensible cool ing in kWh/a
$Q_{mom,i}$	=	energy consumption during time $t_{s,i}$ in kJ/a
$t_{s,i}$	=	period of time with fixed in- and outlet conditions during annual operation time of unit in s
3600	=	conversion factor in kJ/kWh

For further information on this topic see also chapter 8.

5.5.3.2 ENERGY CONSUMPTION FOR LATENT COOLING (DEHUMIDIFICATION)

The momentary energy consumption for latent cooling (dehumidification of air in a cooling coil) is established with the formula:

$$Q_{mom} = q_v \cdot \rho \cdot \left(x_{in} - x_{out} \right) \cdot 2500 \cdot t_s \tag{5.21}$$

where

Q_{mom}	=	energy consumption for a time period t_s in kJ
q_v	=	air flow rate over the coil in m^3/s
ρ	=	density of the considered air flow rate in kg/m^3
x_{in}	=	moisture content of the air at the inlet of the coil in kg/kg
x_{out}	=	moisture content of the air at the outlet of the coil in kg/kg
2500	=	condensation (evaporation) heat of water vapour at moderate coil outlet temperatures in kJ/kg
t_s	=	period of time with steady in- and outlet conditions in s

The annual thermal energy consumption of a cooling coil for latent cooling shall be established taking into account the following factors:

- geographic location of the air handling unit with matching outdoor conditions
- the effectiveness of any energy recovering device characterised by its moisture recovery efficiency
- the selected appropriate humidity scenario
- the operating time of the unit

A correct valuation of the aforementioned factors enables an accurate calculation of the momentarily thermal energy consumption for the duration of any outdoor condition during the operating time of the unit. The annual energy consumption of the cooling coil then is the sum of the momentary energy consumptions for latent cooling according to:

$$Q_{C,L,an} = \sum_{i=Q_{mom,1}}^{i=Q_{mom,n}} \left(Q_{mom,i} / 3600 \right) \tag{5.22}$$

where

$Q_{C,L,an}$ = annual thermal energy consumption of the cooling coil for latent cooling in kWh/a

$Q_{mom,\,i}$ = energy consumption during time $t_{s,\,i}$ in kJ/a

$t_{s,\,i}$ = period of time with fixed in- and outlet conditions during annual operation time of unit in s

3600 = conversion factor in kJ/kWh

For further information on this topic see also chapter 8.

5.5.3.3 TOTAL ENERGY CONSUMPTION FOR SENSIBLE AND LATENT COOLING

From the previous information it will be evident that the total annual thermal energy consumption of a cooling coil is the sum of the energy consumption for sensible and latent cooling!
Hence:

$$Q_{C,tot,an} = Q_{C,S,an} + Q_{C,L,an} \tag{5.23}$$

where

$Q_{C,tot,an}$ = total annual thermal energy consumption of the cooling coil in kWh/a

$Q_{C,S,an}$ = annual thermal energy consumption of the cooling coil for sensible cooling in kWh/a

$Q_{C,L,an}$ = annual thermal energy consumption of the cooling coil for latent cooling in kWh/a

6 ENERGY RECOVERY

6.1 GENERAL

The application of energy recovery devices in air handling systems is a widely used method to decrease the thermal energy consumption of air handling units. A reduction of energy consumption is attained by transferring thermal energy (latent and/or sensible) from one air stream to another.
This Recommendation describes the following most frequently applied energy recovery systems:

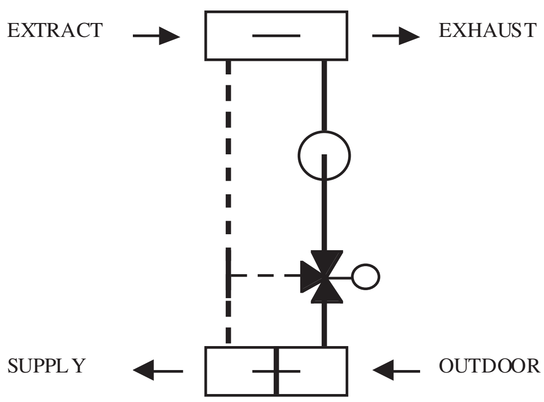
¥ Run around coils
¥ Plate heat exchangers
¥ Rotary heat exchangers

For other energy recovery systems (e.g. heat pipes); the calculation rules for one of the described systems may be applied if both systems are technically comparable.
The presented data, as specified in this chapter, shall only be applied if the actual installation is in compliance with the stipulations. In these cases the described methods enable a proper assessment of the net energy savings that will be accomplished with the computed energy recovery system.

6.2 RUN AROUND COILS

6.2.1 GENERAL

Run around coils are used to recover heat from the extracted air (winter operation) or to transfer sensible energy from the outdoor air to the exhaust air (summer operation).
A run around coil system basically consists of a coil in the extract air and a coil in the supply air; with a pipe system in between to transfer the recovered energy from one air stream to another.
The sketch below shows a run around coil system with circulating pump and three way-control valve in winter operation.



This hydraulic design facilitates defrosting of the coil in the extract air, by closing the control valve and increasing the water inlet temperature to this (cooling) coil.
The efficiency decrease during defrosting periods, due to the reduced flow over the heating coil, shall be taken into account in the seasonal heat recovery efficiency factor of the system. Actual values for seasonal efficiency will be stipulated in the next clause.

Figure 6.1 — Run around coils system

6.2.2 HEAT RECOVERY EFFICIENCY

The heat recovery efficiency is defined as:

➤ Winter operation (cold side): $$\eta_{t2} = \frac{t_{22} - t_{21}}{t_{11} - t_{21}} \qquad (6.1)$$

where

η_{t2} = heat recovery efficiency factor (temperature ratio) for air stream being heated
t_{22} = supply air temperature (outlet colder air stream) in ¡C
t_{21} = outdoor air temperature (inlet colder air stream) in ¡C
t_{11} = extract air temperature (inlet warmer air stream) in ¡C

➤ Summer operation (warm side): $\eta_{t1} = \frac{t_{11} - t_{12}}{t_{11} - t_{21}}$ (6.2)

where

η_{t1}	=	heat recovery efficiency factor (temperature ratio) for air stream being cooled
t_{11}	=	outdoor air temperature (inlet warmer air stream) in °C
t_{12}	=	supply air temperature (outlet warmer air stream) in °C
t_{21}	=	extract air temperature (inlet colder air stream) in °C

At lower outdoor temperatures combined with a relatively high moisture content in the extracted air condensation may occur on the surface of the cooling coil giving a corresponding rise of the recovery efficiency. Under extreme conditions the return water temperature will drop below zero and a potential risk of the condensate freezing occurs.
The graph below demonstrates the basic relationship between the efficiency for winter operation and the lower temperature of the heat transfer fluid (water or freeze preventive solution) at decreasing fluid temperature.

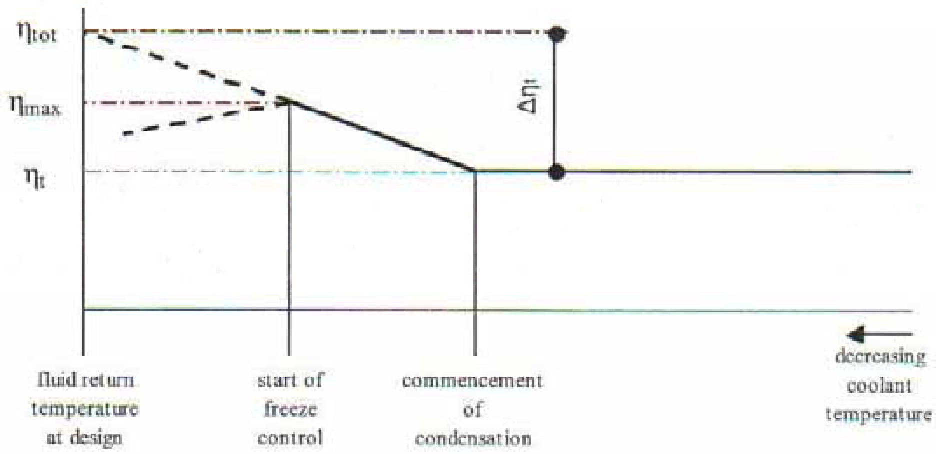


Figure 6.2 – Temperature efficiency versus outdoor temperature

A freeze protection will have to control the valve in such a way that the cooling coil does not freeze up. This device will limit the increase of total heat recovery efficiency to its maximum practical value. In case of a continuous modulating freeze control on the basis of a fixed minimum fluid return set point temperature; the maximum efficiency is achieved when the freeze control is initiated. At lower fluid return temperatures the efficiency diminishes!
For an on/off defrosting device (e.g. activated when a maximum coil pressure drop is exceeded), the efficiency during the defrosting cycle will decrease substantially, depending on the flow through the heating coil. At normal operation however the total efficiency will vary with the entering air conditions of the run around coils and may reach its maximum value.

Technical data for run around coils usually specify the temperatures and efficiencies at design outdoor- and extract temperatures for winter and summer operation. As the graph shows the efficiency at winter design condition is the maximum theoretical value when no freeze control is required. This maximum value is defined as the total efficiency of the system η_{tot} and includes the efficiency increase (Δ_{η_t}) due to condensation on the cooling coil in the extract air.
When the freeze protection is activated the efficiency diminishes.
During dry operation (no dehumidification of extract air), the effectiveness is lower and defined as the dry heat recovery efficiency η_t. The dry efficiency hardly changes with varying water temperatures but be aware of change when there are extremely low outdoor temperatures.

Between commencement of condensation and initiation of freeze control the efficiency shifts between η_t and η_{max}, or between (η_t and η_{tot}); depending on the type of freeze control.
The seasonal efficiency used for energy calculation purposes has to represent the mean value that will be achieved during the season whilst the unit is running; taking into account:
- frequency distribution of outdoor temperature
- humidity of the extracted air
- minimum outdoor temperature.

For summer operation the seasonal effectiveness will not vary with the air inlet conditions, since the cooling coil is likely to operate under dry conditions. The seasonal efficiency will be equal to the design temperature ratio.
During winter operation however, the efficiency may vary; depending on outdoor temperature, extract air conditions and the influence of a freeze protection control if applicable.

Table 6.1 Seasonal effectiveness of run around coil systems

Kind of operation and installation type	Seasonal efficiency $\eta_{t,s}$
Winter operation	
♦ Without humidification	η_t
♦ With humidification	
• No freeze control required	$(2 \cdot \eta_t + \eta_{tot}) / 3$
• Freeze protection required	$(4 \cdot \eta_t + \eta_{tot}) / 5$
Summer operation	η_t

The upcoming table specifies seasonal heat recovery efficiencies to be used for coil energy recovery calculations in common installations.
The outdoor temperature below which a potential need for freeze control is required can be derived from the dry efficiency and the rated fluid flows, by substituting the formula below.

$$t_{21F} = \frac{q_{v(air)} \cdot c_{p(air)} \cdot \rho_{(air)} \cdot \eta_t - \left(c_{p(liquid)} \cdot q_{m(liquid)}\right)}{q_{v(air)} \cdot c_{p(air)} \cdot \rho_{(air)} \cdot \eta_t + \left(c_{p(liquid)} \cdot q_{m(liquid)}\right)} \cdot t_{11} \qquad (6.3)$$

where

t_{21F} = outdoor air temperature at which freeze control starts in ¡C
t_{11} = extract air temperature in ¡C
$q_{v(air)}$ = air flow rate in m³/s
$c_{p(air)}$ = specific heat of the air in kJ/kg·K
$\rho_{(air)}$ = air density in kg/m³
η_t = sensible heat recovery effectiveness in % / 100
$c_{p(liquid)}$ = specific heat of heat transfer fluid in kJ/kg·K
$q_{m(liquid)}$ = rated mass flow rate of heat transfer fluid in kg/s

Note that the numerator is always negative and the denominator is always positive; hence higher extract temperatures lead to lower outdoor temperatures with the potential risk of the condensate freezing!

The equation is only valid if the following conditions are met:
- two coil system with identical coils
- cross — counterflow heat exchange in coils
- initiation of freeze control at 0¡C fluid temperature
- equal air mass flow rate through heating and cooling coil

Further more it has been assumed that there is no condensation until the freeze control starts!
This assumption keeps the calculation on the safe side because condensation above the calculated - freeze protection start temperature, results in higher fluid temperatures.

6.2.3 ENERGY RECOVERY

For any heat recovery system not only the thermal energy savings, but also the additional energy consumption shall be taken into account.
The difference between annual thermal energy savings and additional electric energy consumption has to be sufficiently high to compensate the extra investment and installation costs of the twin coil system.
The profits of the thermal energy savings can be derived from the equations and tables in chapter 8 and the information on energy costs in chapter 5.
Basically the following procedure should be applied:

1) Calculate the total annual demand for heating without heat recovery by substituting $\eta_{t,s}$ = 0 in the relevant equations of chapter 8.4.4.
2) Carry out the computation again with heat recovery, using the seasonal effectiveness $\eta_{t,s}$ in compliance with chapter 6.2.2.
3) Subtract both heating demands 1) & 2) and multiply the result with the costs of thermal heating energy established in accordance with chapter 5.2.
4) Repeat, if applicable, the procedure for the annual demand of sensible cooling energy by applying ncapable to carry over any moisture and the supply air in the recovery coil will not be dehumidified.
5) Determine the sum of the profits obtained from the annual thermal energy savings with the coil energy recovery system.
6) Deduct the costs for additional energy, consumed to operate the system (see next clause).

6.2.4 ENERGY COSTS FOR RUN AROUND COILS

• *Primary side*

The energy consumption on the primary (fluid) side is the required energy to pump the heat transfer fluid through the heat recovery system.
The annual energy consumption of the circulating pump depends on the hydraulic design of the connecting pipe network and the water side pressure drops across the coils.
Other parameters affecting the energy consumption are:
- efficiency of the pump unit (size related)
- type of coil control (variable water flow / variable water temperature)
- speed control on circulating pump
- optimised switch off control in case of inefficient energy recovery

Basically the annually consumed electric energy of the pump is calculated with the equation:

$$W_{circ} = \left(P_{circ} / 1000\right) \cdot t_{eq,op} \qquad (6.4)$$

where

W_{circ} = consumed annual electric energy for circulation in kWh/a
P_{circ} = absorbed electric power of run around coil related pump(s) in W
$t_{eq,op}$ = equivalent running time of pump(s) at full load in h/a

If the loop system is delivered by the AHU manufacturer, the rated absorbed electric power of the pump (P_{circ}) should be specified in the technical data of the heat recovery system. Where no actual data on power consumption is available the following equations and table should be used to determine the absorbed power consumption of the circulating pump(s).

➤ Absorbed power for fluid circulation: $$P_{circ} = q_{v-fluid} \cdot \left(3 \cdot \Delta p_{coil} + \Delta p_{loop}\right) / \eta_p \qquad (6.5)$$

where

P_{circ} = absorbed electric power of run around coil related pump(s) in W
$q_{v\text{-}fluid}$ = rated fluid flow through the system in l/s
Δp_{coil} = fluid side pressure drop across the recovery coil in kPa

Δp_{loop} = pressure drop in the loop system, excluding control valve in kPa

η_p = total efficiency of the pump(s) and electric motor(s) in %/100

For this equation a two coil system has been assumed with identical coils. For multiple coil systems with parallel coils in the extracted air, $3 \cdot \Delta p_{coil}$ shall be substituted by the highest fluid side pressure drop of any coil in the extract air plus twice the fluid side pressure drop of the coil in the supply air.

➤ Hence: $$3 \cdot \Delta p_{coil} = \Delta p_{\max,ea} + 2 \cdot \Delta p_{coil,sa} \qquad (6.6)$$

The liquid flow through the system (=flow through coil in supply air) and pressure drop(s) shall be taken from the air handling unit specification.
Guidelines for the pressure drop in the coil connecting pipe system and pump efficiency are provided in table 6.2

Table 6.2
Network pressure drop & efficiency guidelines for pumps in coil energy recovery loops

PRESSURE DROP IN PIPE LOOP SYSTEM		EFFICIENCY OF PUMP UNIT	
Distance between coil in supply air and most remote coil in extract air [m]	Δp_{loop} in [kPa]	$q_{v,fluid} \times (3 \cdot \Delta p_{coil} + \Delta p_{loop})$ [Watt]	η_p [%]
$d \le 10$	4	$100 < Pcirc \times \eta_p \le 160$	41
$10 < d \le 16$	6	$160 < Pcirc \times \eta_p \le 250$	43
$16 < d \le 25$	10	$250 < Pcirc \times \eta_p \le 400$	45
$25 < d \le 40$	16	$400 < Pcirc \times \eta_p \le 630$	48
$40 < d \le 63$	25	$630 < Pcirc \times \eta_p \le 1000$	51
$63 < d \le 100$	40	$1000 < Pcirc \times \eta_p \le 1600$	53
$100 < d \le 125$	50	$1600 < Pcirc \times \eta_p \le 2500$	54
$125 < d \le 160$	64	$2500 < Pcirc \times \eta_p \le 4000$	55
$160 < d \le 200$	80	$4000 < Pcirc \times \eta_p \le 6300$	57
$200 < d \le 250$	100	$6300 < Pcirc \times \eta_p \le 10000$	61
$250 < d \le 315$	126	$10000 < Pcirc \times \eta_p \le 16000$	64
$315 < d \le 400$	160	$16000 < Pcirc \times \eta_p \le 25000$	67
$400 < d \le 500$	200	$25000 < Pcirc \times \eta p \le 40000$	69

The equivalent full load running time of the pump is related to the operating time of the air handling unit.
Depending on single (winter <u>or</u> summer) or dual (winter <u>plus</u> summer) application and assuming an effective control regime; the equivalent full load running time is considered to be:

_ $t_{eq,op} = 0{,}4 \cdot t_{op}$ →for single operation application (6.7)

_ $t_{eq,op} = 0{,}8 \cdot t_{op}$ →for dual application (6.8)

where

$t_{eq,op}$ = equivalent running time of pump(s) at full load in h/a

t_{op} = operating time of the air handling unit in h/a

¥ *Air side*

The differential pressure to overcome the resistance of the coils is generated by the fans in the corresponding air handling units. As such, the energy consumption involved is included in the calculated electric energy consumption of the fans.

If a separate additional air side energy consumption for the run around coil system has to be established; it shall be derived from the annual electric energy consumption of the fans in the units that comprise an energy recovery coil.
The calculation procedure for fans has been described in chapter 4.
The supplementary annual air side energy consumption of the run around coil system is calcu lated with the equation:

$$W_{CER} = \sum_{fan\ 1}^{fan\ i} \frac{\Delta p_{coil\ 1}}{\Delta p_{fan\ 1}} \cdot W_{fan\ 1} + \ldots\ldots\ldots + \frac{\Delta p_{coil\ i}}{\Delta p_{fan\ i}} \cdot W_{fan\ i} \qquad (6.9)$$

where

- W_{CER} = supplemantary annual air side energy consumption of coil energy recovery system in kWh/a
- $\Delta p_{coil\ 1}$ = air side pressure drop of considered coil (where applicable mean
- Δp_{fan1} = total fan curve pressure of matching fan in Pa
- W_{fan1} = annual energy consumption of the considered fan in kWh/a (calculated in compliance with chapter 4).
- Σ = sum of individual supplementary air side energy consumptions of energy recovery related coils (usually only 2 coils).

6.3 PLATE HEAT EXCHANGER

6.3.1 GENERAL

For heat recovery in air handling systems plate heat exchangers, can be used. In a plate heat exchanger, sensible heat is transferred through corrugated plates from a warm airflow to a cold airflow. During wintertime the heat flow is from the extract air to the fresh air. In summer time it can be the other way round. The heat transfer is realised by conduction through these corrugated plates. In this chapter the heat exchangers are cross flow heat exchangers.

6.3.2 DESIGN

Aluminium is most common material for the corrugated plates although other materials such as plastic are sometimes used. The plates are stacked in parallel and maintained secured by a frame on each corner (see figure 6.3).

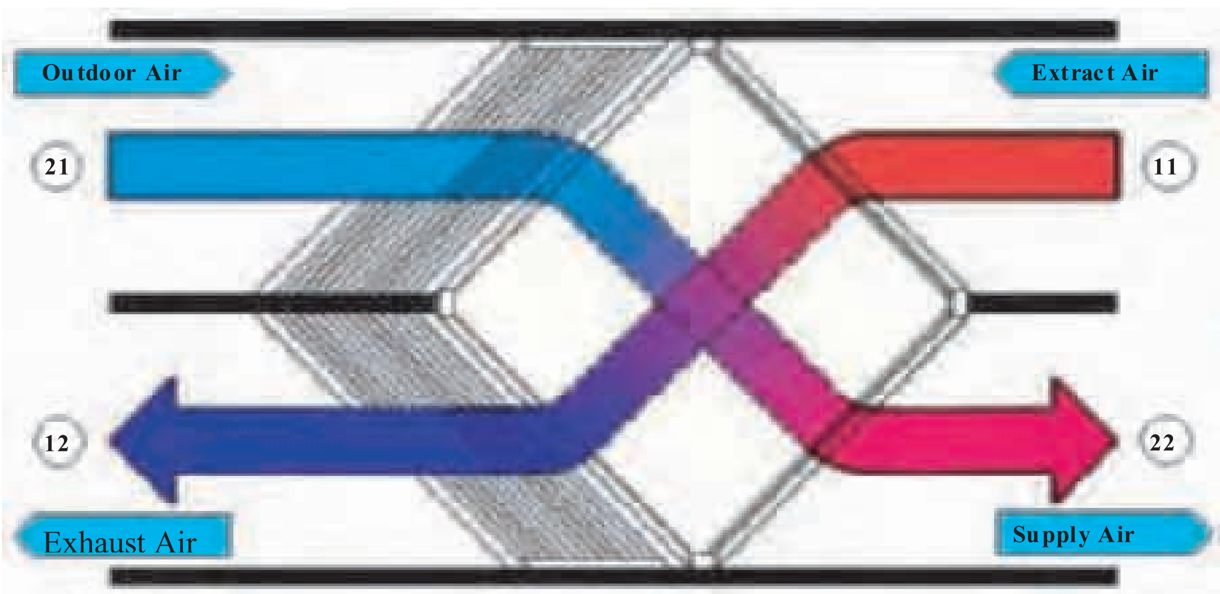


Figure 6.3 — Airflows through plate heat exchanger (winter operation)

6.3.3 HEAT RECOVERY

The most important relation for calculating the energy performance of a plate heat exchanger is the temperature efficiency ratio of the exchanger. This ratio depends on several parameters like: air velocity, mass balance, type of corrugations, plate thickness, condensation, etc.
The heat recovery efficiency is defined as:

➤ Winter operation (cold side): $$\eta_{t2} = \frac{t_{22} - t_{21}}{t_{11} - t_{21}} \qquad (6.10)$$

where

η_{t2}	=	heat recovery efficiency factor (temperature ratio) for air stream being heated
t_{22}	=	supply air temperature (outlet colder air stream) in ¡C
t_{21}	=	outdoor air temperature (inlet colder air stream) in ¡C
t_{11}	=	extract air temperature (inlet warmer air stream) in ¡C

➤ Summer operation (warm side): $$\eta_{t1} = \frac{t_{11} - t_{12}}{t_{11} - t_{21}} \qquad (6.11)$$

where

η_{t1}	=	heat recovery efficiency factor (temperature ratio) for air stream being cooled
t_{11}	=	outdoor air temperature (inlet warmer air stream) in ¡C
t_{12}	=	supply air temperature (outlet warmer air stream) in ¡C
t_{21}	=	extract air temperature (inlet colder air stream) in ¡C

At lower outdoor temperatures in combination with relatively high moisture content in the extracted air condensation may occur on the surface of the plates of the heat exchanger.

It is important for some cases to install a freeze protection system. The necessity for such a system depends on the lowest expected fresh air temperature in relation to the air conditions of the extract air, the type of corrugations and the mass balance. If condensation occurs a simultaneous rise of the heat recovery efficiency will result due to the condensation heat released from the water vapour in the extract air. At very cold conditions the plate temperature can drop below zero and a potential risk of freezing of condensate exists! Icing in the channels affects the performance of the heat exchanger, through increased pressure drop and reduced heat transfer rates, and can lead to damage of the plates.

Figure 6.4 gives the highest temperatures when freezing starts. For systems where the temperature gets below these values it is recommended to install a proper freeze protection system.

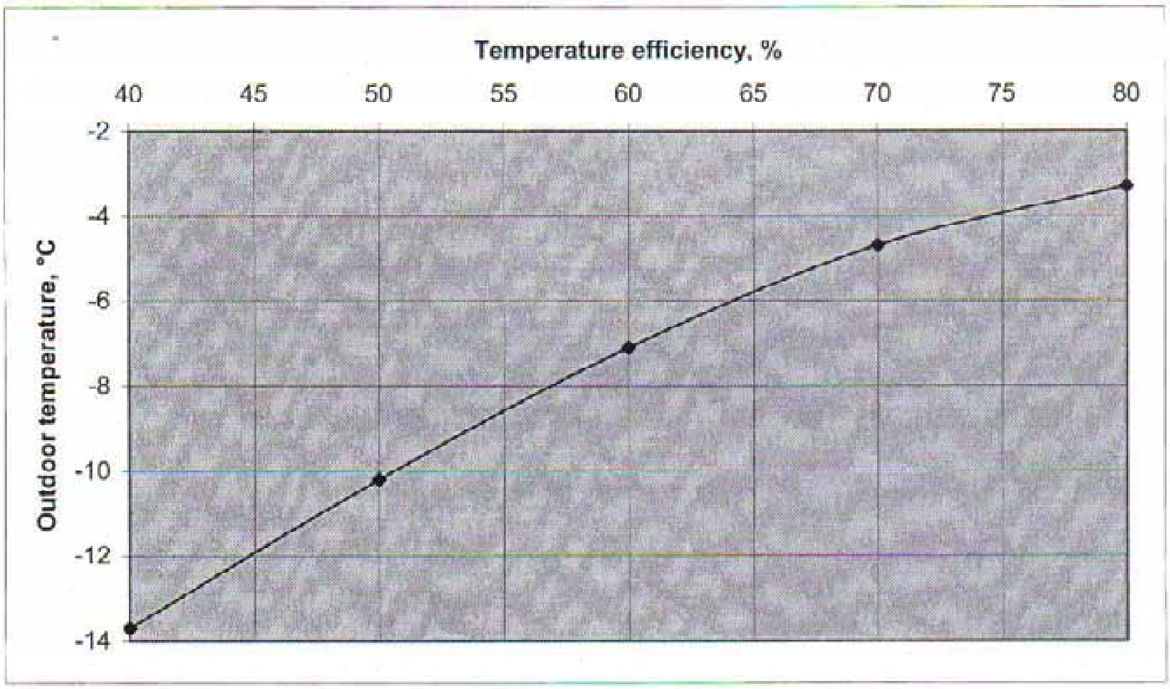


Figure 6.4 – Highest outdoor temperature with risk of freezing versus temperature efficiency

Figure 6.4 is valid for a conventional aluminium plate heat exchanger with equal mass balance between extract and supply air and extract air conditions: 22°C and 25% RH. This is the worst case for a plate heat exchanger. At a higher or lower relative humidity or at higher temperatures of the extract air, the highest temperature when freezing starts is lower. It is also assumed that the plate heat exchanger is positioned in a vertically so that condensate will flow out in a drain pan below it.
This curve may **not** be used to control defrosting systems in general. This has to be checked with the actual supplier of the plate heat exchanger first.

The three most common solutions to prevent damage by freezing are: 1/ to provide a bypass damper in the supply air, 2/ sectional defrosting and 3/ to preheat the fresh air. The preferred solution depends on the outdoor climate where the system is installed. Looking only at the energy consumption, preheating the air to the highest outdoor temperature, according figure 6.4, is the best way (see figure 6.5 and 6.6).

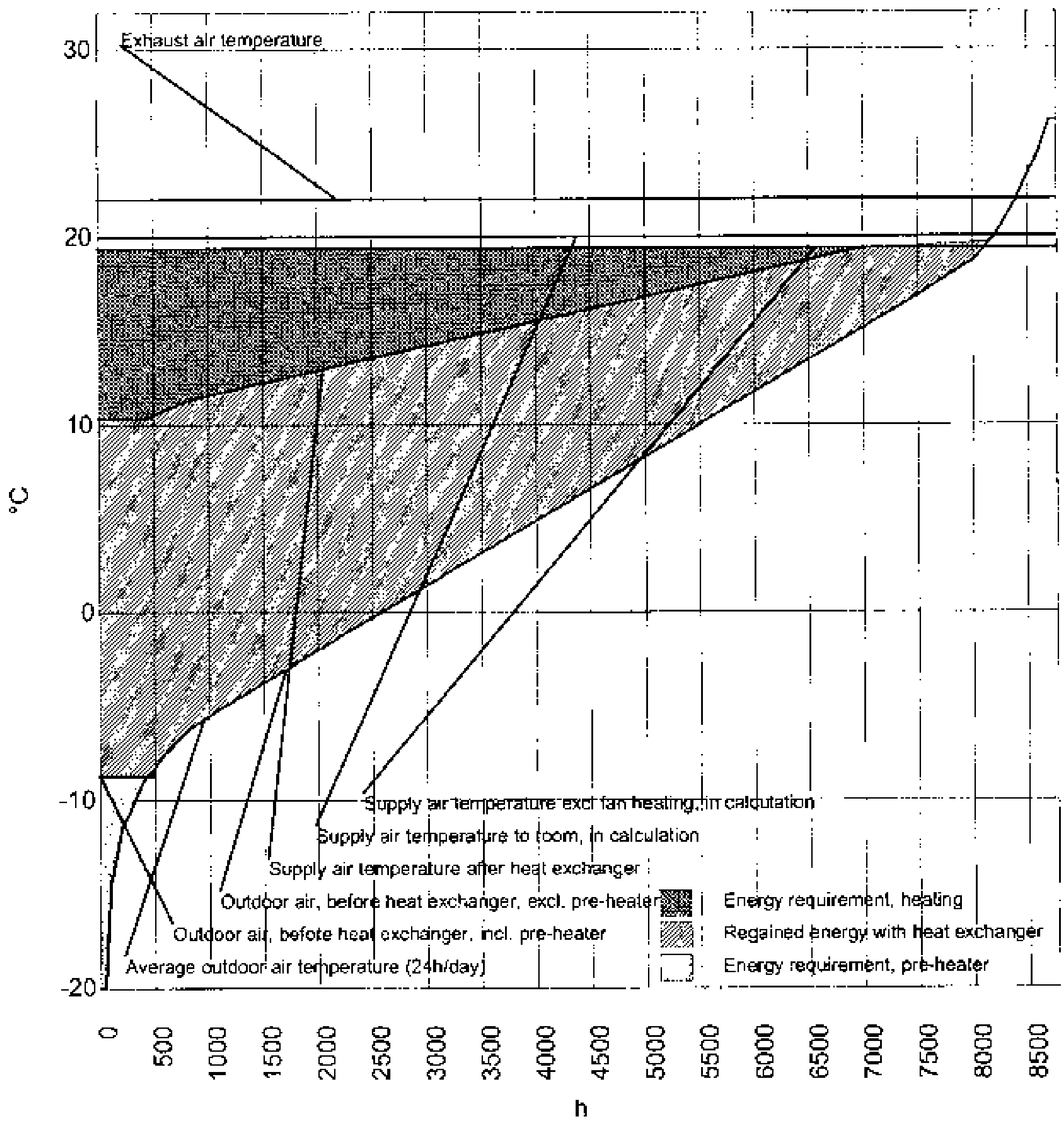


Figure 6.5 – Duration diagram for Sweden for a freeze protection system with pre-heater

The function of the bypass on the cold air flow is to change the mass balance between extract air and supply air in a way that the surplus energy in the extract air is used for defrosting the plates.

The same physical process is used for sectionalised defrosting. This system uses a moving plate or a series of small dampers to cover and close off approximately 10 to 25 % of the heat exchanger surface area on the cold air entering side. As the plate moves or each damper closes, the covered area changes. The covered area will get a momentary relief from the cold air. The extract air passages next to the covered cold air passages can defrost themselves.

For these systems the consequence is that the overall temperature efficiency ratio goes down. This shall be taken into account in the seasonal heat recovery efficiency factor of the system.

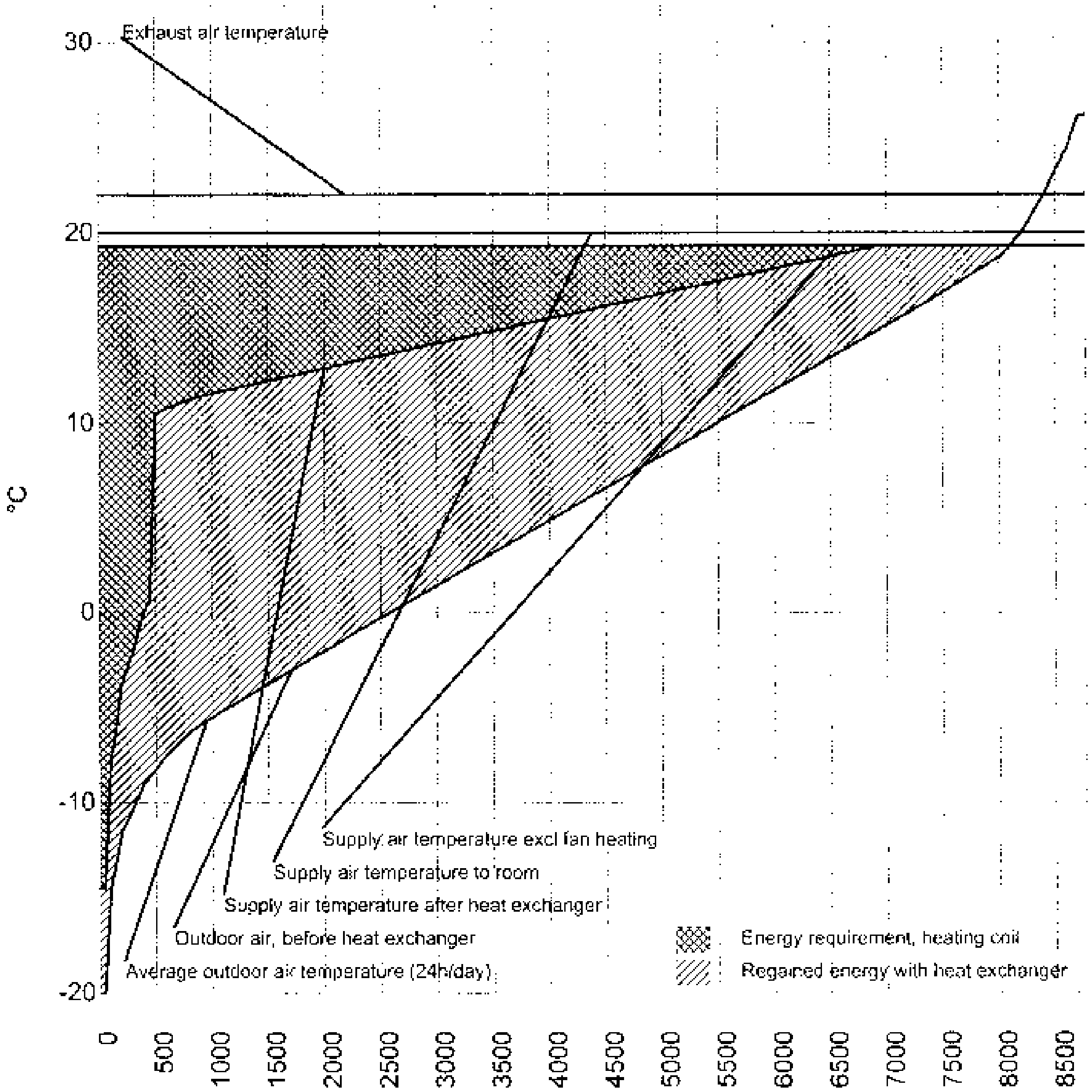


Figure 6.6 – Duration diagram for Sweden for a freeze protection system with bypass damper

Comparing figures 6.5 and 6.6 it's clear that a freeze protection system with pre-heater requires less energy.

Technical data for plate heat exchangers usually specify the temperatures and efficiencies at design outdoor- and extract temperatures during winter operation. As the next graph shows the efficiency at winter design condition is the maximum theoretical value if no freeze control is required.

This maximum value is defined as the total efficiency of the system η_{tot}. During dry operation (no dehu-midification of exhaust air), the effectiveness is lower and defined as the dry heat recovery efficiency η_t. When the freeze protection is activated the efficiency achieves its maximum practical value η_{max}. Between commencement of condensation and initiation of freeze protection control the efficiency between the two extremes (η_t and η_{tot}).

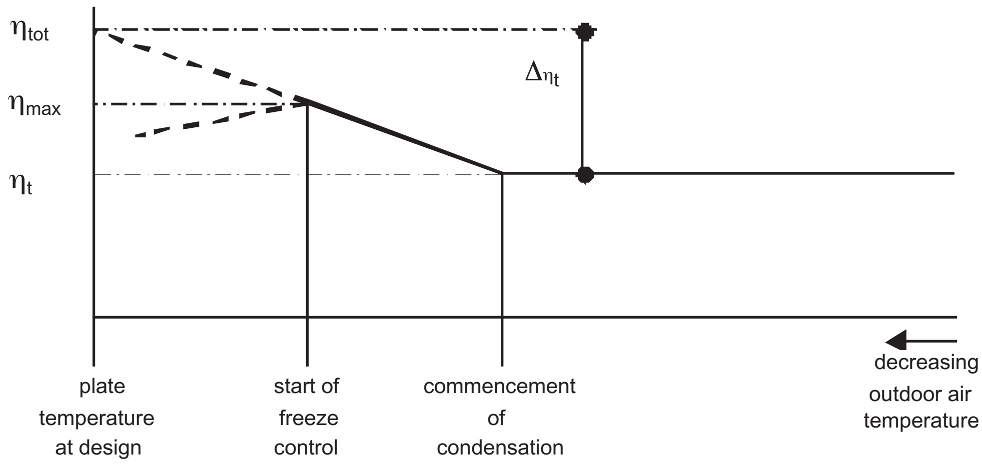


Figure 6.7 — Simplified relationship between the efficiency and the lower temperature of the plates.

The seasonal efficiency used for energy calculation purposes has to represent the mean value that will be achieved during the whole season when the unit is running; taking into account:
¥ frequency distribution of outdoor temperature
¥ humidity of the extracted air
¥ minimum outdoor temperature.

For summer operation the seasonal effectiveness will be equal to the dry efficiency since the plate heat exchanger operates under dry conditions.
During winter operation the efficiency may vary; depending on outdoor temperature.
Table 6.3 specifies seasonal heat recovery efficiencies to be used for energy recovery calculations in common installations. (this is equalin the seasonal efficiency of run around coils in table 6.1 the difference will be the absolute values of the efficiencies).

Table 6.3 Seasonal effectiveness of systems with plate heat exchangers	
Kind of operation and installation type	Seasonal efficiency $\eta_{t,s}$
Winter operation	
♦ Without humidification	η_t
♦ With humidification	
• No freeze control required	$(2 \cdot \eta_t + \eta_{tot}) / 3$
• Freeze protection required (pre-heater)	$(2 \cdot \eta_t + \eta_{tot}) / 3$
• Freeze protection required (sectionalised)	$(3 \cdot \eta_t + \eta_{tot}) / 4$
• Freeze protection required (bypass)	$(4 \cdot \eta_t + \eta_{tot}) / 5$
Summer operation	η_t

As it shows in table 6.3 the extra losses because of the freeze protection with a pre-heater are negligible in relation to the total amount of recovered energy. This is especially true for lower efficiencies.

6.3.4 ENERGY RECOVERY

For any heat recovery system not only the thermal energy savings, but also the additional energy consumption shall be taken into account. The difference between thermal energy savings and additional electric energy consumption on an annual basis, has to be sufficiently high to compensate the extra investment of the plate heat exchanger. The profits of the thermal energy savings can be derived from the equations and tables in chapter 8 and the information on energy costs in chapter 5. To calculate the net annual saved energy see chapter 6.2.3

6.4 ROTARY HEAT EXCHANGER

6.4.1 GENERAL

In the rotary heat exchanger, sensible and latent heat is transferred by means of a matrix structure with micro channels. This structure rotates, between the supply air and exhaust air ducts. Heat exchangers are regenerative and operate according to a counter-flow airstreams, which gives them very high recovery ratio. The matrix must contain a large heat and mass transfer surface and at the same time should provide a low pressure drop as air passes through it. The most common material used in rotors is aluminium.

Performance, leakage and carry-over tests are made according to EN 308 category IIIa and IIIb.

Design

The rotor rotates inside a casing equipped with sealing strips that separate the supply air and exhaust air passages. A drive motor with drive belt is required for operating the section. Changing its speed of rotation can control the recovery ratio of the rotor.

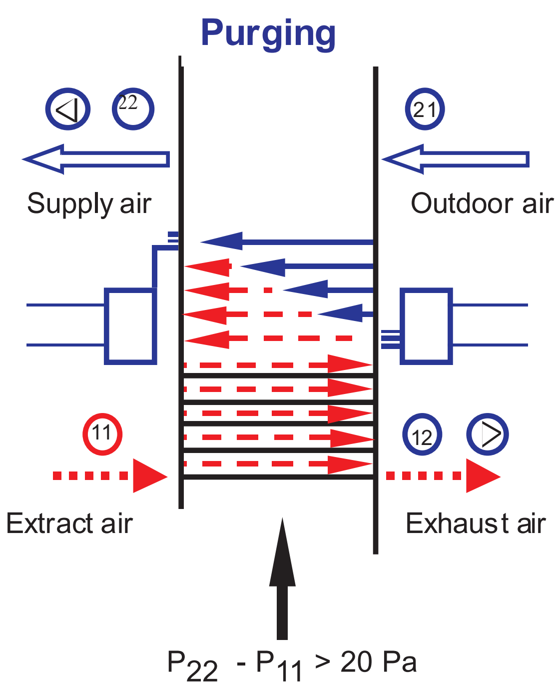


Figure 6.8 —Airflows through rotary heat exchanger (winter operation)

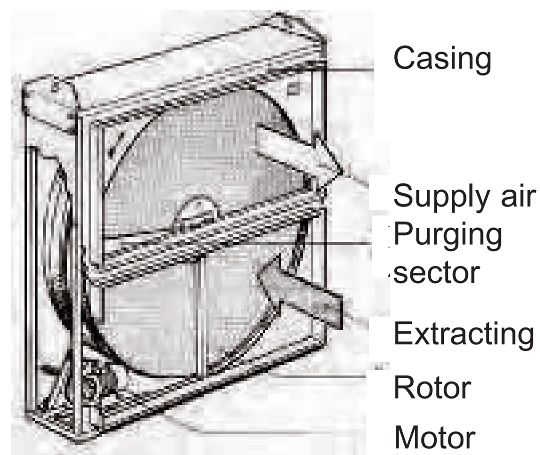


Figure 6.9 — Parts in a rotary heat exchanger

To prevent the polluted extract air from being carried over to the supply air side, most rotary heat recovery sections are equipped with a purging sector. The sector opens up an air passage between outdoor air and extract air and by means of a pressure differential, the rotor passages can be emptied of extract air before they pass into the supply air duct. The angle of the sector is adjusted according to the rotor speed and the pressure differential between outdoor air and extract air.

Regarding ability to recover moisture three different types of rotors can be defined, see chapter 7.5 for details.

6.4.2 HEAT RECOVERY EFFICIENCY

Ratios

The ratios are defined as:

➤ Winter operation (cold side):

Temperature ratio:

$$\eta_{t2} = \frac{t_{22} - t_{21}}{t_{11} - t_{21}} \tag{6.12}$$

where

- η_{t2} = heat recovery efficiency factor (temperature ratio) for air stream being heated
- t_{22} = supply air temperature (outlet colder air stream) in ¡C
- t_{21} = outdoor air temperature (inlet colder air stream) in ¡C
- t_{11} = extract air temperature (inlet warmer air stream) in ¡C

Humidity ratio:

$$\eta_{x2} = \frac{x_{22} - x_{21}}{x_{11} - x_{21}} \tag{6.13}$$

where

- η_{x2} = moisture recovery efficiency factor (humidity ratio) for air stream being humidified
- x_{22} = supply air moisture content (outlet colder air stream) in kg/kg
- x_{21} = outdoor air moisture content (inlet colder air stream) in kg/kg
- x_{11} = extract air moisture content (inlet warmer air stream) in kg/kg

➤ Summer operation (warm side):

Temperature ratio:

$$\eta_{t1} = \frac{t_{11} - t_{12}}{t_{11} - t_{21}} \tag{6.14}$$

where

- η_{t1} = heat recovery efficiency factor (temperature ratio) for air stream being cooled
- t_{11} = outdoor air temperature (inlet warmer air stream) in ¡C
- t_{12} = supply air temperature (outlet warmer air stream) in ¡C
- t_{21} = extract air temperature (inlet colder air stream) in ¡C

Humidity ratio:

$$\eta_{x1} = \frac{x_{11} - x_{12}}{x_{11} - x_{21}} \tag{6.15}$$

where

- η_{x1} = moisture recovery efficiency factor (humidity ratio) for air stream being dehumidified
- x_{11} = outdoor air moisture content (inlet warmer air stream) in kg/kg
- x_{12} = supply air moisture content (outlet warmer air stream) in kg/kg
- x_{21} = extract air moisture content (inlet colder air stream) in kg/kg

The rotary heat recovery section has an extremely high temperature ratio. In a sorption design, it can also transfer moisture (latent heat) at approximately the same ratio, which in certain applications is of great value. Condensation rotors (non hygroscopic) and enthalpy rotors (hygroscopic) are, to a certain extent, also capable of moisture transfer; depending on the inlet and outlet conditions.

With increasing supply air flow at a constant exhaust air flow the heat recovery ratios decrease. The reduction is between 4 to 5 percent units per 10 percent supply air increase.

Anti-frosting protection

A combination of high moisture content in the extracted air and low outdoor temperature can give rise to an excess amount of water in the rotor. If the mean temperature of the rotor during one revolution is below zero, the water may freeze and form ice in the rotor. After a while, this will cause unacceptable high pressure drop across the rotor and a poor energy recovery.

Excess water

It is hard to specify general and precise limits for conditions that do not give rise to excess water forming inside the rotor. The following approximate limits can be used:

Hygroscopic rotor

If the heat exchanger is fitted with a fully hygroscopic rotor, excess water will form as soon as the interconnecting line between the inlet conditions for the extract air and outdoor air intersects the saturation line in the Mollier diagram.

We can define this by indicating maximum moisture content in the exhaust air as a function of the extract air and outdoor air temperature.

$$x_{ea,max} = f_1\ (t_{oa}, t_{ea}) \tag{6.16}$$

If $x_{ea} < x_{ea,max}$ excess water and frosting can be avoided

Non-hygroscopic rotor

A non-hygroscopic rotor transfers moisture only by means of condensation/evaporation and reaches a state of excess water on its surfaces at a lower moisture content in the extract air than a hygroscopic rotor.

$$x_{ea,max} = f_2\ (t_{oa}, t_{ea}) \tag{6.17}$$

If $x_{ea} < x$ excess water and frosting can be avoided.

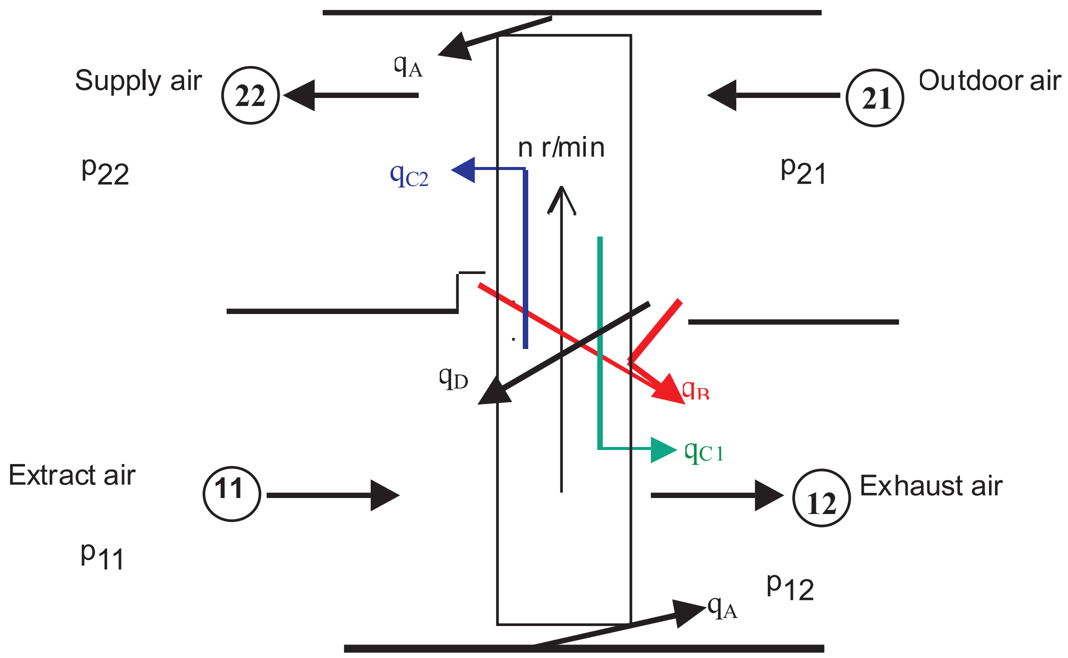
Sorption rotor
A sorption rotor is hardly sensitive to excess water since the rotor transfers moisture in vapour phase. Only extreme conditions may lead to excess water.

Consequences for LCC calculation

There is no basic data for calculating how much heat recovery is annually lost due to defrosting. It is probably insignificant in comparison with the amount heat recovered. On the other hand, however, it is highly significant for the heat power that has to be installed, since a heating coil should make up for the power that is temporarily lost in the heat recovery section.
In a normal comfort ventilation situation there is no need for defrosting, however, if the exhaust air rela-

tive humidity is above 40 % and the outdoor temperature is below -15 ¡C the rotor may need defrosting. During the defrosting period, the temperature ratio of the rotor is only 20 — 30 percent. A typical defrosting time is 15 — 20 minutes and the time between two defrosting periods is several hours. The defrosting impact on seasonal efficiency is neglible.

6.4.3 LEAKAGE



q_A	=	By-pass flows
q_B	=	Leakage flow (throw sealings etc.)
q_{C1}	=	Transferred flow by rotation (on the opposite side of the purge sector)
q_{C2}	=	Carry-over flow(from extract to supply air side)
q_D	=	Purging flow (throw the purging sector)

Figure 6.10 — Leakage in a rotary heat exchanger

By-pass flows lower the temperature and humidity ratios. There are no other disadvantages. They are indirectly included in the ratios indicated by the supplier.

The amount of leakage flow depends on the size of the rotor, the quality of the seals, the differential pressure $p_{21} - p_{12}$ and the pressure drops across the rotor. To provide the leakage from supply air to extract air, the pressure differential between the supply air and extact air ducts, $p_{22} - p_{11}$ should for all operating conditions be at least 20 Pa. It may be necessary to utilize a balancing damper in the extract air duct in order to accomplish this.

The approximate leakage flow can be calculated as follows:

$$q_B = D \cdot (k_1 + k_2 \cdot (p_{21} - p_{11}) \cdot 10^{-3}) \qquad (6.18)$$

where

q_B	=	leakage flow in m^3/s
D	=	rotor diameter in m
k_1 and k_2	=	constants depending on the quality of the seals and adjustment of the rotor

k_1 = 0.02 - 0.04
k_2 = 0.08 - 0.15

The transfered flow q_{C1} from the supply air to the exhaust air side can be expressed as follows:

q_{C1} = „ · (D^2 — d^2)/4 · n /60 · f_p · L (6.19)

where

q_{C1} = transfered flow in m^3/s
D = rotor diameter in m
d = hub diameter in m
n = rotor speed in r/min
f_p = porosity of the rotor structure
L = length of the rotor in the direction of air flow in m

The carry-over flow q_{C2} from extract to supply air side will be zero if the purging sector works in an ideal manner. If the heat recovery section lacks a purging sector then $q_{C2} = q_{C1}$

The angle of the purging sector should be adjusted so that $q_{C2} = 0$ whenever the rotor rotates at full speed $n = n_{max}$. The angle that gives a zero carry-over flow at full speed n_{max} is

$$\theta_0 = \frac{n_{max}}{60} \cdot 360 \cdot \frac{L}{v_{front}} \cdot f_p \qquad (6.20)$$

where

θ_0 = angle of purging sector to accomplish no carry-over flow in ¡
n_{max} = maximum rotor speed in r/min
L = length of the rotor in the direction of air flow in m
v_{front} = velocity on the front area of the purging sector in m/s
f_p = porosity of the rotor structure

The pressure drop coefficient $k = \frac{p_{21} - p_{11}}{(v_{front})^f}$ gives if f ‡ 1:

$$\theta_0 = \frac{n_{max}}{60} \cdot 360 \cdot \frac{L}{\left((p_{21} - p_{11})/k\right)^{1/f}} \cdot f_p \qquad (6.21)$$

where

θ_0 = angle of purging sector to accomplish no purging flow in ¡
n_{max} = maximum rotor speed in r/min
L = length of the rotor in the direction of air flow in m
p_{21} = pressure at outdoor air side in Pa
p_{11} = pressure at extract air side in Pa
k = pressure drop coefficient
f = power factor that characterize the connection between pressure drop and velocity
f_p = porosity of the rotor structure

Without purging sector or an angle of the purging sector, $\theta < \theta_0$ polluted exhaust air will be carried over to the supply air side, which must be avoided.

Consequences for LCC calculations

The exhaust air fan performance shall include the leakage flow q_B
The exhaust air fan shall also include the transfered flow q_{C1} and the purging flow q_D reduced with the

carry over flow q_{C2}. Here is assumed that the fans are located downstream the rotor.

The air leakage flows q_{C1}, q_{C2} and q_D are all dependant of the speed of the rotor.
However it can be shown that the total leakage to the exhaust air side, $q_B + q_{C1}$ —$q_{C2} + q_D$,is independent of the speed of the rotor and can be written:

$$q_{v,leak} = q_B + \frac{\theta}{180} \cdot \left(\frac{p_{21} - p_{11}}{p_{21} - p_{22}} \right)^{\frac{1}{f}} \cdot q_{22} \qquad (6.22)$$

where

$q_{v,leak}$	=	total leakage flow in m³/s
q_B	=	leakage flow throw sealings etc. in m³/s
θ	=	angle of purging sector in ¡
p_{21}, p_{11}, p_{22}	=	air pressures according to fig 6.10 in Pa
f	=	power factor that characterize the connection between pressure drop and velocity. For normal rotor structures the power factor is very close to 1.
q_{22}	=	supply air flow rate in m³/s

If a balancing damper is needed, its pressure drop will burden the exhaust air fan.

Rotor leakage and balancing damper pressure drop should be included in the manufactures selection programs, see chapter 4 (Fans)

6.4.4 ENERGY RECOVERY

Heating recovery

$$Q_{REC,H} = 1.2 \cdot 1.0 \cdot q_{v,sa} \cdot \frac{t_{op}}{8760} \cdot \sum_{i=1}^{8760} (t_{ea,i} - t_{oa,i}) \cdot \frac{\eta_t}{100} \qquad (6.23)$$

if $(t_{ea,i}$ —$t_{oa,i})$ † 0 set $(t_{ea,i}$ —$t_{oa,i}) = 0$
if $t_{oa,i} + (t_{ea,i}$ —$t_{oa,i}) \cdot \eta_t/100$ ‡ $t_{sa,i}$ set $(t_{ea,i}$ —$t_{oa,i}) \cdot \eta_t/100 = t_{sa,i}$ —$t_{oa,i}$

where

$Q_{REC,H}$	=	recovered heating energy in kWh/a
t_{op}	=	annual operation time in h/a
$tq_{v,sa}$	=	supply air flow rate in m³/s
t_{ea}	=	temperature of extract air in °C
t_{oa}	=	temperature of outdoor air in °C
η_t	=	temperature ratio in %
t_{sa}	=	temperature of supply air in °C

Cooling recovery

$$Q_{REC,C,sens} = 1.2 \cdot 1.0 \cdot q_{v,sa} \cdot \frac{t_{op}}{8760} \cdot \sum_{i=1}^{8760} (t_{oa,i} - t_{ea,i}) \cdot \frac{\eta_t}{100} \qquad (6.24)$$

if $(t_{oa,i}$ —$t_{ea,i})$ † 0 set $(t_{oa,i}$ —$t_{ea,i}) = 0$

$$Q_{REC,C,lat} = 1.2 \cdot q_{v,sa} \cdot \frac{t_{op}}{8760} \cdot 2500 \cdot \sum_{i=1}^{8760} (x_{oa,i} - x_{ea,i}) \cdot \frac{\eta_x}{100} \qquad (6.25)$$

if $(x_{oa,i}$ —$x_{ea,i})$ † 0 set $(x_{oa,i}$ —$x_{ea,i}) = 0$
Here is the sensible heat of the steam neglected
If non-hygroscopic rotor $Q_{REC,C,lat} = 0$

where

$Q_{REC,C,sens}$	=	recovered sensible cooling energy in kWh/a
$Q_{REC,C,lat}$	=	recovered latent cooling energy in kWh/a
t_{op}	=	annual operation time in h/a
$q_{v,sa}$	=	supply air flow rate in m^3/s
t_{oa}	=	temperature of outdoor air in °C
x_{oa}	=	moisture content of outdoor air in kg/kg
t_{ea}	=	temperature of extract air in °C
x_{ea}	=	moisture content of extract air in kg/kg
η_t	=	temperature ratio in %
η_x	=	humidity ratio in %

For more accurate calculation the day may be split up in day-time operation and night-time operation.

Calculations of heating and cooling demands at different scenarios and locations are done according to chapter 8.

6.4.5 ENERGY COSTS

Power consumptions for rotary heat exchangers

a. Extra fan power to overcome the pressure drops in the rotor

$$P_{el,pressure} = (q_{v,sa} \cdot \Delta p_{sa}) / (\eta_{sa} \cdot 1000) + (q_{v,ea} \cdot (\Delta p_{ea} + \Delta p_{damper})) / (\eta_{ea} \cdot 1000) \quad (6.26)$$

where

$P_{el,pressure}$	=	extra fan power to overcome the pressure drops in the rotor in kW
$q_{v,sa}$	=	supply air flow in m^3/s
Δp_{sa}	=	pressure drop over rotor on supply air side in Pa
η_{sa}	=	total efficiency of fan, supply air side
$q_{v,ea}$	=	exhaust air flow in m^3/s
Δp_{ea}	=	pressure drop over rotor on exhaust air side in Pa
Δp_{damper}	=	pressure drop over balancing damper on extract air side in Pa
η_{ea}	=	total efficiency of fan, exhaust air side

b. Fan power to exhaust fan due to rotor leakage

$$P_{el,leak} = (q_{v,leak} / q_{v,ea}) \cdot P_{el} \quad (6.27)$$

where

$P_{el,leak}$	=	fan power to exhaust fan due to rotor leakage in kW
$q_{v,leak}$	=	leakage flow in rotor from supply air side to exhaust air side in m^3/s
$q_{v,ea}$	=	exhaust air flow in m^3/s
P_{el}	=	fan power to exhaust fan without rotor leakage in kW

c. Rotor motor drive

$$P_{el,drive} = k \cdot D^2/1000 \quad (6.28)$$

where

$P_{el,drive}$	=	power to rotor motor drive in kW
k	=	constant, 25 — 35 depending on type of seal
D	=	rotor diameter in m

Normally the electrical energy to the drive motor during a year is less than 0.5 % of recovered heating energy.

The annual energy consumptions can be calculated as:

$$W = (P_{el,pressure} + P_{el,leak}) \cdot t_{op,AHU} + P_{el,drive} \cdot t_{op,rotor} \qquad (6.29)$$

where

W	=	annual additional energy consumption in kWh
$P_{el,pressure}$	=	extra fan power to overcome the pressure drops in the rotor in kW
$P_{el,leak}$	=	fan power to exhaust fan due to rotor leakage in kW
$P_{el,drive}$	=	power to rotor motor drive in kW
$t_{op,AHU}$	=	annual operation time for the AHU in h
$t_{op,rotor}$	=	annual running time for the rotor in h

7 HUMIDIFICATION

7.1 GENERAL

The function of a humidifier in an air handling unit is to increase the moisture content of the supplied fresh air. A moisture rise is required if the prevailing moisture content in the outdoor air is insufficient to achieve the desired relative humidity in the building.
Detailed calculations (not in the scope of these recommendations) should establish the relationship between desired relative humidity in the building and required absolute moisture content of the supply air; depending on outdoor condition, building occupancy and building characteristics.
In this chapter a uniform calculation procedure, based on a pre—selected moisture scenario, will be presented.

7.2 HUMIDIFIER TYPES

Humidifiers can be classified accordingly the humidification process (steam or water) and/or depending on the principle of operation.
Particularly for water humidification a great variety of humidifier types has been developed since the need for comfort humidification has been acknowledged.
This Recommendation only deals with (modern) types of humidifiers, frequently applied nowadays in air handling units.

- ➢ Steam humidifiers:
  - ▪ life steam humidifier with (steam jacketed) dispersion pipes
  - ▪ electric steam generator with steam distribution pipe(s)
  - ▪ gas fired steam generator with steam distribution pipes
- ➢ Water humidifiers:
  - ▪ air washers
  - ▪ atomising humidifiers:
    - water/compressed air nozzles
    - ultrasonic
    - high pressure nozzles

7.3 ENERGY CONSUMPTION

For humidification purposes three kinds of energy are consumed:

① energy consumption on primary side (if appropriate)
② energy consumption on air side
③ thermal energy consumption

7.3.1 ENERGY CONSUMPTION ON PRIMARY SIDE

Energy consumption on the primary side of a humidifier is considered to be the energy required to operate the humidifier; excluding the energy to evaporate the water (thermal energy).
Usually electrical energy is consumed to run the humidifier (pumps, compressor, etc.).
As a basis for computation the absorbed power consumption at rated performance shall be taken from supplier s specification.
Where no actual data on power consumption is available, the specified values in this chapter shall be used. In all cases a proportional ratio between primary energy consumption and humidifier performance shall be assumed!

- *Life steam humidifier*
  The primary energy consumption is basically the required energy to transport the steam from the steam boiler to the air handling unit, including heat losses of the pipe network.
  The steam pressure in the boiler induces the steam and condensate flow. All the energy consumption involved, however, is included in the seasonal effectiveness of the energy source (boiler).

- *Electric steam generator*
  Although the steam is electrically generated primary side energy losses of electric steam generators incorporate only losses caused by heat generation in the power supply cables (identical to electric heating coils).

Table 7.1 Primary energy losses in power supply cables to electric steam generators in % of generator capacity

GENERATOR CAPACITY	DISTANCE BETWEEN POWER PANEL AND ELECTRIC STEAM GENERATOR					
kW	10m	16m	25m	40m	63m	100m
10	0,7	1,2	1,8	2,9	4,6	5,0
12,5	0,7	1,2	1,8	2,8	4,4	5,0
16	0,7	1,1	1,8	2,8	4,4	5,0
20	0,6	0,9	1,4	2,2	3,5	5,0
25	0,5	0,7	1,1	1,8	2,9	4,6
31,5	0,4	0,6	0,9	1,4	2,2	3,5
40	0,3	0,5	0,7	1,1	1,7	2,7
50	0,3	0,4	0,7	1,1	1,7	2,7
63	0,3	0,4	0,7	1,1	1,7	2,7
80	0,2	0,3	0,4	0,7	1,1	1,8
100	0,2	0,3	0,4	0,6	1,0	1,6

These energy losses are proportional to the length of the power supply cables and the capacity of the generator. For extensive systems with many electric steam generators, values from the following table may be used.

The consumed annual primary energy for an electric steam generator is calculated with the formula:

$$W_{el,hum} = \left(PEL/100\right) \cdot Q_{hum} \qquad (7.1)$$

where

$W_{el,hum}$ = annual electric energy consumption on the primary side of an electric steam generator in kWh/a

PEL = primary energy loss according to table 9 in %

Q_{hum} = annual thermal energy consumption for humidification in kWh/a (see chapter 7.3.3)

Where no particular installation data is available, a PEL—value of 1% may be presumed!

Gas fired steam generator

Gas fired steam generators have been developed as stand—alone units for local application on single air handling units, as an alternative for electric steam generators.

As the steam output is relatively small, the generators are equipped with atmospheric burners.

Energy consumption on the primary side is essentially the energy required to transport natural gas from the gas meter to the burner. The gas main supply pressure will maintain a continuous gas flow on demand. Hence, no additional energy on the primary side is consumed (electric power supply to control box is negligible)!

• *Air washers*

Primary side energy consumption for air washers is the energy consumption required to operate the spray water pump. The annually consumed electric energy of the spray water pump is computed with the equation:

$$W_{circ} = \left(P_{circ} / 1000\right) \cdot t_{eq,op} \tag{7.2}$$

where

W_{circ} = consumed annual electric energy to run the spray water pump in kWh/a

P_{circ} = absorbed electric power of spray water pump in W

$t_{eq,op}$ = equivalent annual running time of pump at rated load in h/a

The equivalent annual running time can be derived from the yearly evaporated amount of water and the evaporated water rate per unit of time at design conditions. Assuming an optimised moisture control strategy, where the power consumption of the pump is proportional to the evaporated water quantity, the equivalent running time is calculated with:

$$t_{eq,op} = \frac{\dfrac{Q_{hum} \cdot 3600^{*}}{2500}}{\left(x_{sa} - x_{oa}\right) \cdot q_v \cdot \rho \cdot 3600^{**}} = \frac{Q_{hum}}{2500 \cdot \left(x_{sa} - x_{oa}\right) \cdot q_v \cdot \rho} \tag{7.3}$$

where

$t_{eq,op}$ = equivalent annual running time of pump at rated load in h/a

Q_{hum} = annual thermal energy consumption for humidification in kWh/a

3600^{*} = conversion factor in kJ/kWh

2500 = evaporation heat of water at moderate wet bulb outlet temperature in kJ/kg

x_{sa} = moisture content of supply air at design condition in kg/kg

x_{oa} = moisture content of entering/outdoor air at design condition in kg/kg

q_v = air flow rate through humidifier in m^3/s

ρ = density of the considered air flow rate in kg/m^3

3600^{**} = conversion factor in s/h

The rated absorbed electric power of the pump (P_{circ}) should be specified in the technical data of the air washer.

Where no actual data in the quotation is provided, the absorbed power of the spray water pump is calculated with the formula:

$$P_{circ} = \Phi_v \times p_{man} / \eta_p \tag{7.4}$$

where

P_{circ} = absorbed electric power of spray water pump in W

q_v = spray water flow in l/s

p_{man} = manometric head of spray water pump in kPa

η_p = total efficiency of pump and electric motor in %/100

Table 7.2 Efficiency guidelines for spray water pump–units for air washers	
EFFICIENCY OF PUMP UNIT	
$q_v \times p_{man}$ [Watt]	η_p [%]
$100 < Pcirc \times \eta_p \leq 160$	33
$160 < Pcirc \times \eta_p \leq 250$	34
$250 < Pcirc \times \eta_p \leq 400$	39
$400 < Pcirc \times \eta_p \leq 630$	42
$630 < Pcirc \times \eta_p \leq 1000$	43
$1000 < Pcirc \times \eta_p \leq 1600$	44
$1600 < Pcirc \times \eta_p \leq 2500$	49
$2500 < Pcirc \times \eta_p \leq 4000$	55
$4000 < Pcirc \times \eta_p \leq 6300$	60
$6300 < Pcirc \times \eta_p \leq 10000$	63
$10000 < Pcirc \times \eta_p \leq 16000$	66
$16000 < Pcirc \times \eta_p \leq 25000$	67
$25000 < Pcirc \times \eta_p \leq 40000$	68

The total efficiency for pump and electric motor is size related and can be taken from the table below.

¥*Atomising humidifiers*

→ Water/compressed air nozzles

Atomising humidifiers with water/compressed air nozzles use compressed air to produce finely atomised sprays which evaporate completely to raise the humidity of the passing air.

Besides the power consumption of the control panel; the majority of primary side energy is con sumed to operate the compressor unit. The actual power consumption of the compressor unit will depend on:

- capacity and type of compressor
- efficiency of the electric drive motor
- design of compressed air pipe system
- nozzle type

As a realistic guideline (based on 0,6 m^3/h compressed air per l/h evaporated water) the pri mary energy consumption can be valued at:

$$W_{el} = 0{,}1 \cdot Q_{hum} \tag{7.5}$$

where

W_{el} = yearly consumed electric energy on the primary side of the humidifier in kWh/a

Q_{hum} = annual thermal energy consumption for humidification in kWh/a

→ Ultrasonic humidifiers

The ultrasonic humidifier generates atomised water by means of ultrasonic vibrations of approximately 1,7 MHz. A humidity unit consists basically of a number of small vibration elements. In each element an ultrasonic vibration of a diaphragm below water level introduces small water particles into the air stream.

The electrical energy required to generate the ultrasonic vibrations, is considered to be the energy consumption on the primary side.

With sufficient accuracy the primary energy consumption for this type of humidifier is appraised at:

$$W_{el} = 0{,}075 \cdot Q_{hum} \tag{7.6}$$

where

W_{el} = yearly consumed electric energy on the primary side of the humidifier in kWh/a

Q_{hum} = annual thermal energy consumption for humidification in kWh/a

→ <u>High pressure nozzles</u>

Humidifiers using high pressure nozzles to atomise the water, are equipped with a high pressure pump unit with control panel and electric motor. The power input to the pump unit results in a yearly energy consumption on the primary side.
Most of the high pressure nozzle humidifiers (often called cold fog humidifiers) operate with a surplus of water. The ratio between the amount of atomised and evaporated water mainly depends on the required relative humidity at the humidifier outlet. Nozzle and humidifier type are also influencing factors.
In spite of the great variety of affecting parameters; the primary energy consumption of a cold fog humidifier can be reliably computed with the equation:

$$W_{el} = 0{,}02 \cdot Q_{hum} \tag{7.7}$$

where

W_{el} = yearly consumed electric energy on the primary side of the humidifier in kWh/a

Q_{hum} = annual thermal energy consumption for humidification in kWh/a

7.3.2 ENERGY CONSUMPTION ON AIR SIDE

The required air flow through the humidifier is maintained by a pressure difference across the section. The supply fan in the air handling unit sustains the required differential pressure across the component. The energy consumption involved is included in the calculated consumption of electric energy of the fan.

7.3.3 CONSUMPTION OF THERMAL ENERGY

All humidifiers consume thermal energy to evaporate the water required to raise the moisture content of the passing air. Steam humidifiers produce or use steam which is introduced into the air stream in the humidifier. The air temperature will slightly increase, caused by the steam distribution pipes and the down—cooling of the steam to air temperature.
In a water humidifier the water evaporates in the passing air stream. The required evaporation heat is extracted from the air flow which leads to a substantial reduction of the air temperature.
Similar as for the calculation of thermal energy consumption for heating and cooling; a (moisture) scenario is needed to compute the annual energy consumption for humidification. The moisture scenario for humidification is independent of the installation type. An example is depicted in the next graph. Humidification is required as long as the moisture content of the outdoor air is below the desired moisture content of the supply air. During the humidification season a constant set point condition of the supply air has been assumed!
In this graph the set point for the supply air is 6 g/kg. It has been presumed that the moisture production in the building results into a moisture gain in the extract air of 1g/kg during the entire year. When the humidification stops, the moisture content of the supply air and the extract air will follow the outdoor condition until the cooling coil commences to dehumidify.

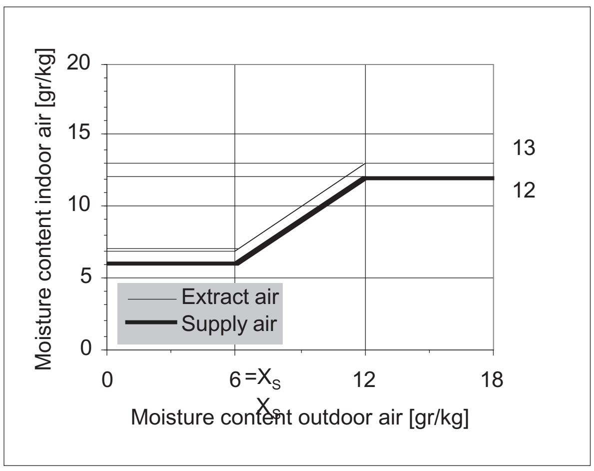


Figure 7.1 — Moisture scenario

The momentary energy consumption for humidification is calculated with the equation:

$$Q_{mom} = q_V \cdot \rho \cdot (x_{out} - x_{in}) \cdot 2500 \cdot t_s \quad (7.8)$$

where

Q_{mom}	=	energy consumption for a time period t_S in kJ
q_V	=	air flow rate through the humidifier in m^3/s
ρ	=	density of the considered air flow rate in kg/m^3
x_{out}	=	moisture content of the air at the humidifier outlet in kg/kg
x_{in}	=	moisture content of the air at the humidifier inlet in kg/kg
2500	=	evaporation heat of water at 0¡C in kJ/kg
t_s	=	period of time with steady in- and outlet conditions in s

The annual thermal energy consumption of any humidifier shall be established taking into account the following factors:
- geographic location of the air handling unit with matching outdoor conditions
- seasonal latent effectiveness of any energy recovering device, characterised by its moisture recovery efficiency
- the selected set point of the supply air leaving the humidifier
- the operating time of the unit

A correct valuation of the aforementioned factors enables an accurate calculation of the momentary thermal energy consumption for the duration of any outdoor condition during the operating time of the unit.
The annual energy consumption of the humidifier then is the sum of the momentary energy consumptions for humidification, according to:

$$Q_{hum} = \sum_{i = Q_{mom,1}}^{i = Q_{mom,n}} \left(Q_{mom,i} / 3600 \right) \quad (7.9)$$

where

Q_{hum}	=	annual thermal energy consumption of the humidifier in kWh/a
$Q_{mom,i}$	=	energy consumption during time $t_{S,i}$ in kJ/a
$t_{s,i}$	=	periods of time with fixed in- and outlet conditions during annual operation time of unit in s
3600	=	conversion factor in kJ/kWh

For further information on this topic see also chapter 8.

7.4 COSTS OF THERMAL ENERGY FOR HUMIDIFICATION

The cost price of thermal energy for humidification shall be established at the location of the considered air handling unit and shall include the overall efficiency for steam or heat production.
In all cases, the fuel price and/or price for electricity must be provided by the public utility, responsible for delivery!

7.4.1 COSTS OF THERMAL ENERGY FOR STEAM HUMIDIFIERS

- *Life steam*
  Life steam humidifiers obtain steam from a central steam boiler in a plant room. For the seasonal efficiency and energy costs we refer to chapter 5.2.
- *Electric steam generators*
  The seasonal efficiency of an electric steam generator is affected by:

- drain frequency of the cylinder(s), depending on water quality
- heat loss of the steam cylinder(s)
- stand by periods

Unless otherwise specified, a seasonal efficiency of 95% for moderate use of local steam gen erators may be assumed. The costs for thermal energy follow the equation:

$$p_{hum} = \frac{E_{el}}{\eta_{s,hum}} \qquad (7.10)$$

where

p_{hum} = costs of thermal energy for humidification in €/kWh
E_{el} = price for electricity in €/kWh
$\eta_{s,hum}$ = seasonal efficiency of humidifier in %/100

• *Gas fired steam generator*

For the seasonal efficiency of a local gas fired steam generator, the same considerations can be made as for an electric steam generator. The seasonal efficiency is valued at 80% unless stated otherwise.

The costs for thermal energy can be calculated with the formula:

$$p_{hum} = \frac{E_{el}}{\eta_{s,hum}} \qquad (7.11)$$

where

p_{hum} = costs of thermal energy for humidification in €/kWh
E_f = fuel price in €/m^3
3600 = conversion factor in kJ/kWh
H_h = higher heating value of gas in kJ/m^3
$\eta_{s,hum}$ = seasonal efficiency of humidifier in %/100

7.4.2 COSTS OF THERMAL ENERGY FOR WATER HUMIDIFICATION

As described in 7.3.3, the necessary heat to evaporate the water is extracted from the air stream.
To maintain the desired air supply temperature, an additional heat input in the heating coil is required; equal to the extracted evaporation energy.
Hence, the thermal energy for water humidification is coming from the same source as the thermal energy for heating. The calculation procedure therefore is the same.
For seasonal efficiency and energy costs see chapter 5.2.

7.5 MOISTURE RECOVERY

The only energy recovering device (within the scope of this Recommendation) that also recovers moisture during the humidification season, is a rotary heat exchanger (or any other recovery device, operating according to the same physical principle).
Depending on the rotor type, the moisture recovery efficiency will vary with the in— and outlet conditions of the air streams. For a proper assessment of the seasonal efficiency for moisture recovery, three different rotor types have to be considered:

• *Condensation rotor*

A condensation rotor (non hygroscopic rotor) will transfer moisture from the extract air to the supply air as long as water vapour condensates on the rotor material in the extract air.
If and how much water vapour condensates, depends on the moisture content in the extract air and the moisture content of the outdoor air at saturation.

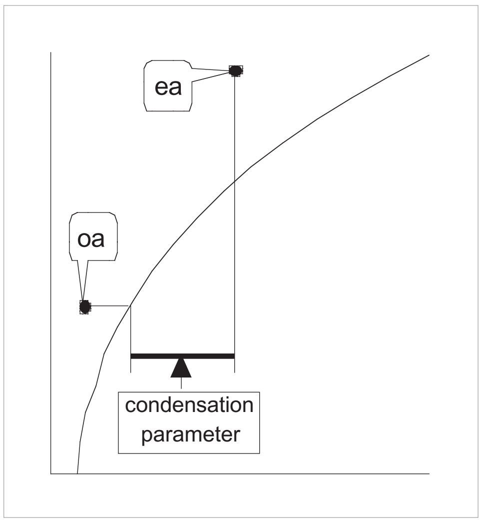


Figure 7.2 — Condition for condensation

The diagram shows that the temperature of the outdoor air (oa) must be lower than the dew point temperature of the extract air (ea).
It is obvious that for a fixed extract air condition the efficiency varies with the outdoor condition.
Accurate calculations demand a computer program linked to selection software from rotor suppliers, where for each outdoor condition in the Mollier diagram the efficiency and accompanying moisture recovery are established.
A summation of all the individual computer calculations results in the total amount of recovered moisture during the humidification season.
As a guidance for comparative calculation purposes, the seasonal moisture efficiencies (as a percentage of the latent efficiency at desigN conditions) have been tabulated in table 7.3. The efficiencies are sufficiently reliable if the conditions, mentioned below the table, comply more or less with the actual situation!

• *Hygroscopic rotor*
A hygroscopic rotor (enthalpy rotor) is basically a condensation rotor with a porous layer on the rotor material. This surface treatment is accomplished by submerging the entire rotor in a chemi cal liquid.
Due to the hygroscopic layer, moisture is not only transferred by condensation but also by absorption of moisture in the extract air. Condensed and adsorbed moisture are released in the supply air.
For the impact of condensation on moisture transfer, the same fundamentals as for condensa tion rotors apply.
The contribution of the hygroscopic layer to the moisture recovery strongly depends on the applied chemical process; hence on the rotor manufacture. Now even more a computer pro gram, as described before, is required to compute accurate seasonal moisture recovery.
For comparison purposes, however, within the scope of this Eurovent Recommendation; a sea sonal moisture efficiency (as a percentage of the latent efficiency at design conditions) has been tabulated in table 7.3. The same restrictions as for condensation rotors are valid; in gener al results may diverge more, depending on the quality of the hygroscopic layer!

• *Sorption rotor*
A sorption rotor has a moisture efficiency almost independent of the entering air conditions. The rotor material has a solid desiccant (e.g. silica gel) on the surface, which will adsorb water vapour from the extracted air stream. Desorption of an equal amount of water vapour in the out door air stream raises the humidity of the supply air. Adsorption and desorption are induced by changes in relative humidity of the air during the rotor passage.
The seasonal efficiencies of the sorption rotor, as submitted in table 7.3, are in fact reduction factors (as a percentage of the latent efficiency at design conditions) to compensate the inevitable surplus of moisture recovery when the absolute humidity of the outdoor air is close to the desired value of the supply air.
This phenomenon only occurs in case of higher latent efficiency in combination with a moisture gain in the extract air. In reality a speed control on the rotor will reduce the efficiency to the appropriate level.
The lower the humidifier set point, the more calculation compensation is needed!
Precise calculation results can only be accomplished with a computer program as described before.

The accuracy of calculated results obtained with the figures of table 7.3, depends on how close the stipulated conditions below the table are met!

Table 7.3 Seasonal rotor efficiencies[1)] for moisture recovery							
Rotor type	Humidifier set point X_s of the supply air						
	3 g/kg	4 g/kg	5 g/kg	6 g/kg	7 g/kg	8 g/kg	9 g/kg
Condensation rotor	51%	44%	43%	44%	47%	50%	53%
Hygroscopic rotor	78%	71%	67%	64%	63%	63%	63%
Sorption rotor	82%	87%	90%	93%	94%	94%	95%

[1)] As a percentage of the latent efficiency at design conditions!
Following design conditions have been assumed:
outdoor air : —10 °C at 1 g/kg
pressure drop rotor: –150 Pa
moisture gain: 1 g/kg
Figures valid for western European climates

Percentages have been derived from western European outdoor conditions.
To achieve best possible accuracy, it is strongly recommended to use the same design conditions.
Values for intermediate set points may be interpolated!

The profits of a rotary heat exchanger (or similar device), recovering moisture, can be derived from the equations and tables in chapter 8 and the data on thermal energy costs for humidification in chapter 7.4. Additional energy consumption to operate the energy recovery device (pressure drop, drive system) shall be taken into account with the application (heating, cooling or humidification) where the highest savings are achieved (usually heating).
To compute the financial benefits of the moisture recovery device, the following procedure should be conformed to:
1) Calculate the total annual thermal energy demand for humidification without moisture recovery by substituting $\eta_x = 0$ in the relevant equations of chapter 8.
2) Carry out the computation again with moisture recovery, using the seasonal efficiency as established in this chapter.
3) Subtract both energy demands 1) & 2) and multiply the result with the costs for thermal energy for humidification in accordance with chapter 7.4
4) Deduct the costs for additional energy, consumed to operate the system (if applicable).

7.6 HUMIDIFIER WATER ECONOMY

The costs for water consumption and water treatment (if appropriate) are relatively low compared to the thermal energy costs for humidification. However to evaluate the application of different types of humidifiers in all it s aspects; this chapter gives appraising guidelines on expenditures for water consumption and water treatment.

7.6.1 WATER CONSUMPTION

All humidifiers will consume more city water than the amount of water required to humidify the air. The ratio between the two depends on the humidifier type, controls and required water quality!
In general, humidifiers demand the following water quality:

➤ Steam humidifiers : city water or softened water
➤ Water humidifiers
  ¥ air washers : city water or softened water
  ¥ atomising humidifiers : demineralised (reverse osmosis) water

Exceptions for special applications have been allowed for. The quality of the required feed water shall be specified in manufacturer s quotation!

A better water quality on the one hand reduces the drain rate or bleed off rate of the humidifier, but on the other hand raises the amount of city water needed for a proper economic water treatment.
The following table enables a quick appraisal of the water consumption for different types of humidifiers, depending on the required water quality.
The table has been based on the following assumptions:

→ Medium hard city water (14¡D) is supplied by the public utility
→ Feed water factor is the assessed ratio between totally supplied feed water to the humidifier and annually evaporated water quantity in the air stream
→ FWQ is the demanded feed water quality to be considered
→ Ratio between soft water supply and demineralised water production is 1,5 (due to reject and flush water for R.O. unit).
→ Ratio between consumed city water and produced soft water is 1,05 (due to drain water for regener ation)

Table 7.4 Water consumption ratios for different types of humidifiers

Humidifier type	Feed water factor	Demineralised (RO)˚water	Softened water	City water
Steam humidifiers	1,10	—	—	FWQ
	1,05	—	FWQ	1,05
	1,00	FWQ	1,50	1,05
Air washers	2,00	—	—	FWQ
	1,50	—	FWQ	1,05
	1,10	FWQ	1,50	1,05
Atomising humidifiers*	1,10	FWQ	1,50	1,05

*Lower water qualities are strongly dissuaded!

City water consumption is computed by multiplication of the yearly evaporated water quantity with the factors, valid for the applied water quality (indicated FWQ).

The yearly evaporated amount of water or steam supplied into the air stream, is calculated with the equation:

$$Q_w = \frac{Q_{hum} \cdot 3600}{2500} \qquad (7.12)$$

where

Q_w = yearly evaporated amount of water or absorbed steam in the outdoor air stream in kg/a
Q_{hum} = annual thermal energy consumption for humidification in kWh/a
3600 = conversion factor in kJ/kWh
2500 = evaporation heat of water at moderate temperatures in kJ/kg

Example:
What is the annual water consumption of an atomising humidifier which consumes 120.000 kWh/a of thermal energy?
Evaporated amount of water in the air stream: Q_w = (120.000 x 3600) / 2500 = 172.800 kg/a
- demineralised water consumption : 172,8 x 1,1 = 190,1 m^3/a
- soft water consumption : 172,8 x 1,1 x 1,50 = 285,1 m^3/a
- city water consumption : 172,8 x 1,1 x 1,50 x 1,05 = 299,4 m^3/a

The price for city water must be provided by the public utility, responsible for delivery.
Prices for treated water should be derived from actual technical specifications and data of the R.O. installation.

7.6.2 EXPENDITURES FOR WATER TREATMENT

When prices for treated water are not available (which usually is the case), the additional costs for water treatment have to be assessed; based on the information in this chapter.
The brine tank of a water softener has to be filled with salt for regeneration of the unit.

➤ The salt consumption for soft water production is valued at 0,5 kg/m^3.

The power consumption of a water softener may be disregarded!

A reverse osmosis unit consumes electric energy for the production of demineralised water. The majority of energy is used by a water pump required to force the feed water through a semi—permeable membrane and a booster pump needed to supply the consumer(s) with product water (permeate). The energy consumption for the production and distribution of R.O. water strongly depends on:

- the size of the reverse osmosis unit
- the extension of the pipe network for product water

In general the required water production for humidification purposes is rather low; hence R.O. units with low performances are often applied.

➤ As a guideline 4 kWh of electricity per m^3 R.O. water should be used to compute the energy costs for the production of demineralised water.

8 GENERAL GUIDELINES FOR THE CALCULATION OF ENERGY DEMANDS IN AHU

8.1 TYPES OF ENERGY USED IN AHU

The thermal and physical treatment of the air in air handling units (AHU) requires different types of energy, depending on the utilised systems and components:

¥heating of air: air can be heated by means of thermal energy (heat exchangers fed with hot water from fuel fired boilers or district heating) or by means of electrical energy (electrical heat exchangers). In addition electrical energy is demanded to run the utilised pumps

¥cooling of air: air can be cooled by means of cooling systems based on compression cycles using electrical energy for running the system, or based on absorption cycles using thermal energy for operating the absorption cycle process or based on evaporation processes like adiabatic cooling. In addition electrical energy is demanded to run the utilised pumps

¥humidification of air: air can be humidified by means of water (evaporation humidification) or vapour (steam humidification). In case of steam humidification, thermal energy is required for the generation of the steam. In case of evaporation humidification, electrical energy is demanded to run the injection pumps. Evaporation heat is withdrawn from the passing air (thermal energy)

¥ air transport: in AHU air is transported by means of fans, using electrical energy

¥auxiliary devices for AHU: to operate AHU properly, a number of auxiliary devices such as damper motors, control equipment, lighting systems and pumps are needed. All of these devices require electrical energy

The above list shows, that the overall demand of energy for AHU«s can be summarised into two classes of energy:

¥ electrical energy
¥ thermal energy

Further information on the demand of energy, the utilised classes of energy and the different systems and components for air treatment including efficiencies and systems effects have been given in chapters 4 to 7 in this recommendation.

The price levels for fuel and electrical energy are very much depending on national, international and temporary circumstances due to politics and economics. Therefore reliable prices for 1 kWh of energy (fuel or electricity) must be ascertained individually.

8.2 BOUNDARY CONDITIONS FOR THE CALCULATION OF ENERGY CONSUMPTION

8.2.1 CRITERIA REFERRING TO PHYSICAL PROPERTIES

density: The density of the air is fixed to ρ = 1,2 kg/m^3

air flow rate: The air flow rate q_v [m^3/s] shall always be given under standard conditions according to EN 13053.

heat recovery: The temperature ratio η_t and humidity ratio η_x [%] shall always be given under standard conditions according to EN 308.

8.2.2 CRITERIA REFERRING TO LEVELS OF AIR TEMPERATURE AND MOISTURE CONTENT

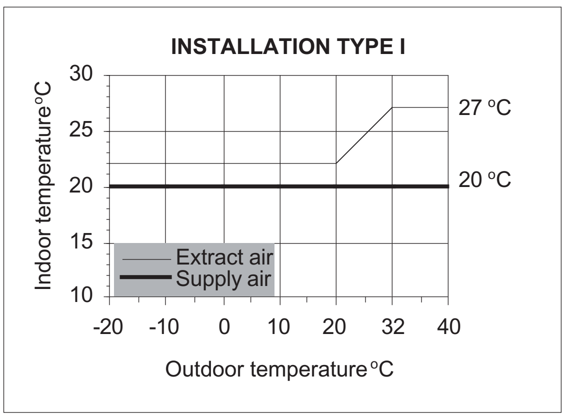
The temperatures of the air in the AHU play the most important role, as the calculation of the annual energy demand for thermal air treatment is very much linked to the question, how much the temperature of the outdoor air (oa) has to be changed by means of heating or cooling, to fit to the requirements of the supply air (sa). In case of heat recovery, the extracted air (ea) supports the thermal treatment of the outdoor air. So the temperature levels of the extract air, which are very much linked to the room air temperatures, are important to know.

As each AHU is operating under different ambient conditions with different demands concerning the room air temperatures, overall valid temperature levels cannot be fixed. Nevertheless general scenarios for temperature levels can be defined in principle. These scenarios show the temperatures of the supply air and of the extract air on the air handling system depending on the actual outdoor air temperatures.

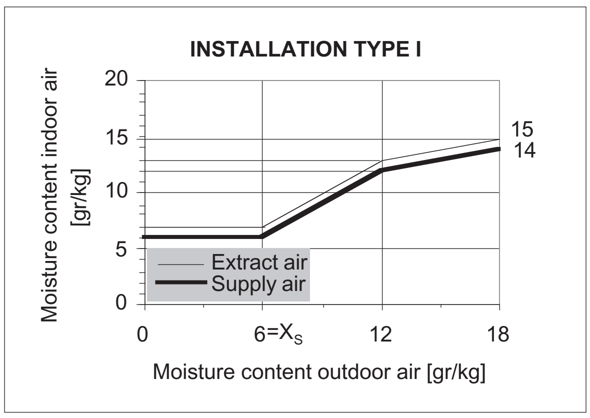
Three basic supply and extract air temperature and moisture scenarios are presented below for three different types of installation.

These installation types have been explained in chapter 5.

<u>*Installation type I*</u>
- Central ventilation system
- Additional local heating and/or cooling in case of heating and/or cooling load



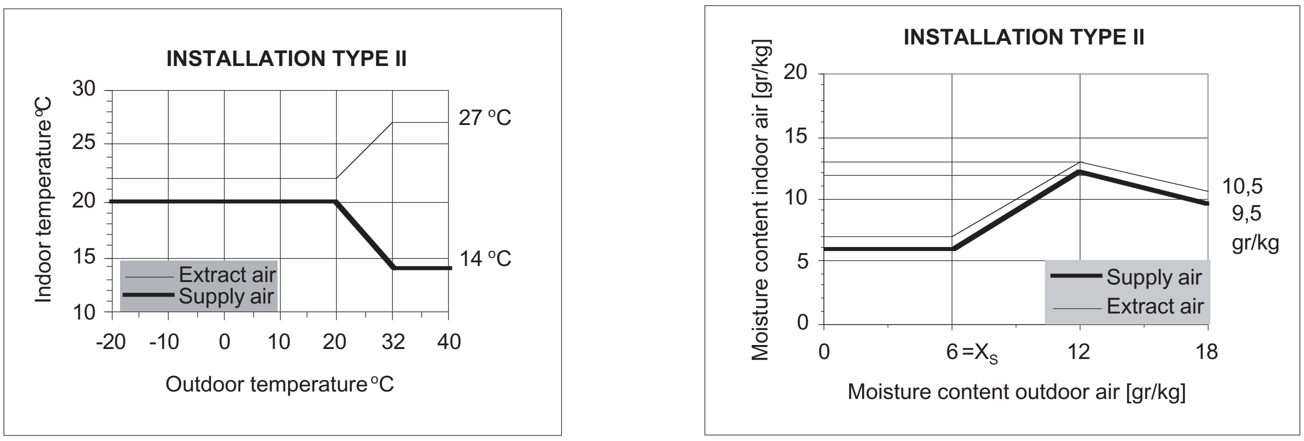
Temperature scenario



Moisture scenario

Figure 8.1 —Temperature and moisture scenario, installation type I

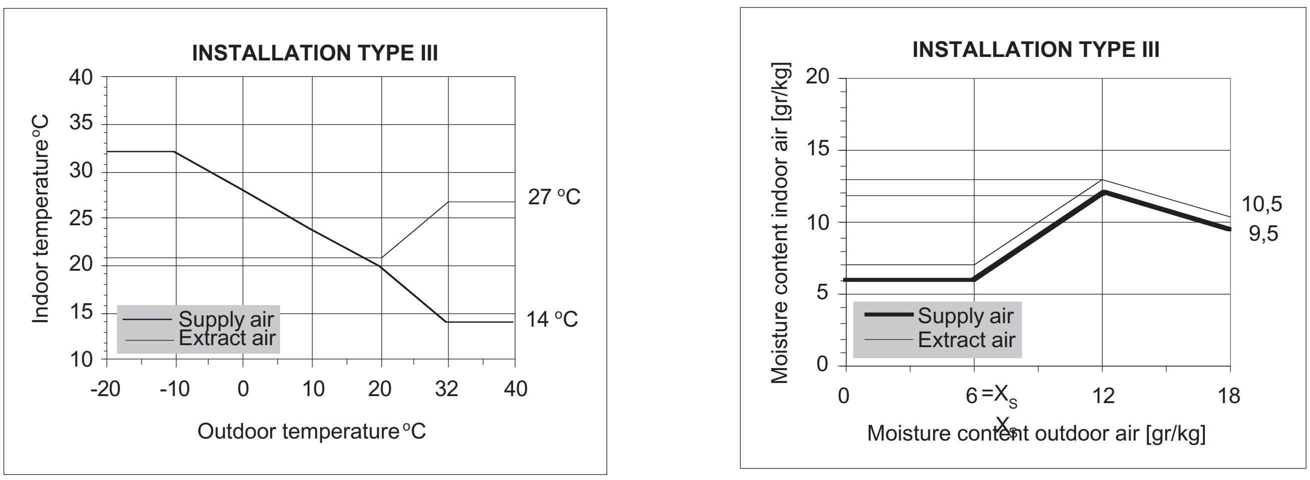
Installation type II
- Central ventilation and cooling system
- Additional local heating in case of heating load
- Additional local cooling in case of high cooling load



Temperature scenario ***Moisture scenario***

Figure 8.2 —Temperature and moisture scenario, installation type II

Installation type III
- Central air conditioning system for ventilation, heating and cooling
- Additional local heating in case of high heating load
- Additional local cooling in case of high cooling load



Temperature scenario ***Moisture scenario***

Figure 8.3 —Temperature and moisture scenario, installation type III

In these installation types it is assumed that the heating period is at outdoor temperatures below 20 …C and cooling period at outdoor temperatures above 20 …C.

In the heating and cooling calculation method in chapter 8.4 and 8.5 the supply and extract temperatures and moisture contents are variables which can be changed. Nevertheless values according to the scenarios above are suitable default values.

More detailed information on the different temperature and moisture levels is given in chapter 5.

8.2.3 CRITERIA REFERRING TO LEVELS OF AIR HUMIDITY

The actual humidity s of the outdoor air, the supply air and the extract air influence the operation conditions and the demand of the energy for thermal air treatment in the air handling system. Thereby not the relative humidity φ [%] but the moisture contents of the air x [kg H_2O/kg dry air] is of importance.

In case of the utilization of humidification systems and/or dehumidifying cooling coils in the AHU the actual moisture scenarios have to be taken into account. Thereby the relative humidity φ is the leading parameter for the humidification of the supply air. Depending on the utilization of the room or building, the relative humidity of the supply air (φ_{sa}) has to be within a well defined range. For human comfort, the relative humidity of the room air should be between 30% and 60%, with a moisture content not higher than 13 g H_2O/kg air . In industrial processes, the relative humidity of the room air depends very much on the type of process and can vary in principal between 0% and 100%.

For energy calculations the moisture content of the supply air (x_{sa}) is important, and can be calculated using the relation:

$$x_{sa} = 0{,}622 \cdot \frac{\frac{\varphi_{sa}}{100} \cdot p_{s,sa}}{101300 - \frac{\varphi_{sa}}{100} \cdot p_{s,sa}} \qquad (8.1)$$

where

x_{sa}	=	moisture content of supply air in kg vapour/kg dry air
φ_{sa}	=	relative humidity of the supply air in %
$p_{s,sa}$	=	saturation pressure of the water vapour in Pa
0,622	=	ratio between molecule weight of water and dry air
101300	=	atmospheric pressure at sea level in Pa

Thereby, the saturation pressure of the water vapour in the air is depending on the supply air temperature and can be calculated by:

$$p_{s,sa} = 288{,}68 \cdot \left(1{,}098 + \frac{t_{sa}}{100}\right)^{8{,}02} \qquad 0 \dagger t_{sa} \dagger 35\ldots C \qquad (8.2)$$

where

$p_{s,sa}$	=	saturation pressure in Pa
t_{sa}	=	supply air temperature in ¡C

The moisture content of the extract air (x_{ea}) is linked to the moisture content of the supply air by taking into account the vapour loads of the room or building. The vapour loads () are the amounts of water vapour which are emitted into the room air by persons, plants, areas of liquid water inside of the room (swimming pools, indoor ponds, etc.) and production processes:

$$\overset{¥}{v} = \overset{¥}{v}_{person} + \overset{¥}{v}_{process} + \overset{¥}{v}_{water\ areas} + \overset{¥}{v}_{plants} \qquad (8.3)$$

where

$\overset{¥}{v}$	=	total vapour load of the room or building in g/h
$\overset{¥}{v}_{person}$	=	vapour emitted into the room air by persons in g/h
$\overset{¥}{v}_{process}$	=	vapour emitted into the room air by processes in g/h
$\overset{¥}{v}_{water\ areas}$	=	vapour emitted into the room air by areas of liquid water in g/h
$\overset{¥}{v}_{plants}$	=	vapour emitted into the room air by plants in g/h

Concerning common building technologies, vapour diffusion through exterior walls can be ignored. The vapour loads induced by persons are depending on the degree of physical work and can be assumed as follows:

$$\dot{V}_{person} = 50 \ \frac{g\,vapour}{h}$$

$$\dot{V}_{person} = 130 \ \frac{g\,vapour}{h}$$

Taking into account the vapour loads, the moisture content of the extract air can be calculated as:

$$x_{ea} \approx x_{sa} + \frac{\dot{V}}{q_V \cdot 1{,}2} \qquad (8.4)$$

where

x_{ea}	=	moisture content of the extract air in g H_2O/kg air
x_{sa}	=	moisture content of the supply air in g H_2O/kg air
$\dot{V}$	=	total vapour loads of the room or building in g/h
q_V	=	air flow rate in m^3/h

For a typical office building, the calculation indicates an increase of the moisture contents between supply air and extract air by Δx = 1 g H_2O /kg dry air.

More information on the humidification and on the influence of humidification on the overall energy consumption has been given in chapter 7 of this recommendation.

8.2.4 CRITERIA REFERRING TO THE DESIGN OF THE AHU

Casing air leakage:

The air leakage of the casing can be neglected in the calculation of the energy consumption of the AHU if the leakage class is equal to or better than class L3 according to EN 1886.

Thermal transmittance of the casing:

The thermal transmittance of the casing can be neglected in the calculation of the energy consumption of the AHU if the thermal transmittance is equal to or better than class T4 according to EN 1886. If the thermal transmittance is of class T5, it has to be taken into account.

Airside pressure drops of the components in the AHU:

The airside pressure drops of the components in the AHU have to be taken into account under the design conditions of the unit:

- air filters are taken under average design conditions, depending on the filter class:
- filter class G1 to G4: $$\Delta p_{des} = \frac{\Delta p_{ini} + 150}{2} \qquad (8.5)$$
- filter class F5 to F7: $$\Delta p_{des} = \frac{\Delta p_{ini} + 200}{2} \qquad (8.6)$$
- filter class F8 and higher: $$\Delta p_{des} = \frac{\Delta p_{ini} + 300}{2} \qquad (8.7)$$

where

Δp_{des} = design pressure drop of filter in Pa

Δp_{ini} = initial pressure drop of filter in Pa

heat exchangers are taken under average design conditions:

$$\Delta p_{des} = \frac{\Delta p_{dry} + \Delta p_{wet}}{2} \qquad (8.8)$$

where

Δp_{des} = design air side pressure drop of heat exchanger in PA

Δp_{dry} = air side pressure drop of dry heat exchanger, i.e. without condensing water on fins in Pa

Δp_{wet} = air side pressure drop of wet heat exchanger, i.e. with condensing water on fins in Pa

¥ humidifiers are assumed to work under design conditions

Fans:

The performance of the fans and the fan motors shall always be given under the condition that they are installed inside of the casing of the AHU.

8.2.5 CRITERIA REFERRING TO THE OPERATION CONDITIONS OF THE AHU

Operation time:

The operation time t_{op} [h/a], i.e. the annual running time of the AHU, is very much dependent upon the application of the air handling system. Apart from the overall annual operation time, the ratio of daytime to night-time operation of the AHU is important to know. Therefore the annual operation time should be defined in 2 parameters:

¥ $t_{op,d}$ [h/a]: annual operation time during daytime from 6.00 to 18.00
¥ $t_{op,n}$ [h/a]: annual operation time during night-time from 18.00 to 6.00

If the actual annual operation time is not known, following figures can be assumed:

Table 8.1 Assumed annual operation time

type of building	annual operation time during daytime from 6.00 to 18.00 $t_{op,d}$ [h/a]	annual operation time during night-time from 18.00 to 6.00 $t_{op,n}$ [h/a]	total annual operation time t_{op} [h/a]
office buildings (open from 6.00 to 20.00)	2800 h/a	400 h/a	3200 h/a
office buildings (open 24 h a day)	3100 h/a	3100 h/a	6200 h/a
industrial plants (1 shift; 5 day a week)	2200 h/a	400 h/a	2600 h/a
industrial plants (2 shifts, 5 days a week)	3100 h/a	1600 h/a	4700 h/a
industrial plants (3 shifts, 5 days a week)	3150 h/a	3150 h/a	6300 h/a
shops	3750 h/a	1550 h/a	5300 h/a
restaurants	1900 h/a	3400 h/a	5300 h/a
hotels and hospitals	4380 h/a	4380 h/a	8760 h/a

Variable air volume (VAV):

When using VAV, the total consumption of energy for thermal treatment and air transportation will be ¥ ¥ ¥ reduced by:

¥ reduction of air flow to be cooled and / or heated
¥ reduction of air flow to be transported
¥ reduction of airside pressure losses due to reduction of air flow velocities

To take into account the reduced energy consumption when using VAV, reduction factors shall be introduced in the calculations:

¥ $\eta_{VAV,HC}$: reduction factor for heating and cooling energy
¥ $\eta_{VAV,fan}$: reduction factor for fan energy consumption

If no actual data is available, the following default values can be used:

¥ average reduction of the air flow rate by 35% over the year
($q_{V,VAV} = 0{,}65 \cdot q_{V,CAV}$)
¥ $\eta_{VAV,HC} = 0{,}65$
¥ $\eta_{VAV,fan} = 0{,}42$

If the AHU operates with constant air volume (CAV), the reduction factors are equal to 1:

¥ $\eta_{VAV,HC} = \eta_{CAV,HC} = 1{,}0$
¥ $\eta_{VAV,fan} = \eta_{CAV,fan} = 1{,}0$

8.2.6 CRITERIA REFERRING TO THE AUXILIARY DEVICES

Energy consuming auxiliary devices for AHU are:

¥ pumps for humidifiers, heating coils, cooling coils and heat recovery systems
¥ damper motors
¥ motors for rotary heat exchanger systems
¥ lighting of the casing
¥ control equipment
¥ defrosting devices

The consumption of electrical energy for damper motors, lighting and control equipment can be neglected. All other devices have to be taken into account with respect to the nominal power input, the load function and the annual operation time of the device.

8.3 AMBIENT CONDITIONS ON SITE

The demand of heating and cooling energy for the thermal treatment of the air is very much depending on the ambient conditions on site such as temperature and humidity of the outdoor air. As temperature and humidity vary over the year, the annual energy consumption for heating and cooling has to be determined by taking into account the annual frequency of the outdoor conditions, as defined in the reference year of the location. The reference year provides typical temperatures and humidities for each hour of a year at the location of interest.

With these data the specific enthalpies of the outdoor air (h_{oa}) [kJ/kg]) can be determined for each hour of the year:

$$h_{oa,i} = 1{,}0 \cdot t_{oa,i} + x_{oa,i} \cdot \left(2500 + 1{,}86 \cdot t_{oa,i}\right) \tag{8.9}$$

where

$h_{oa,i}$:	=	specific enthalpy of the outdoor air at the hour i of the reference year in kJ/kg
$t_{oa,i}$:	=	temperature of the outdoor air at the hour i of the reference year in …C
$x_{oa,i}$:	=	moisture content of the outdoor air at the hour i of the reference year in kg/kg
1,0:	=	specific heat of dry air in kg/kg›K
2500:	=	evaporation heat of water at 0¡C in kJ/kg
1,86:	=	specific heat of moisture in kg/kg›K

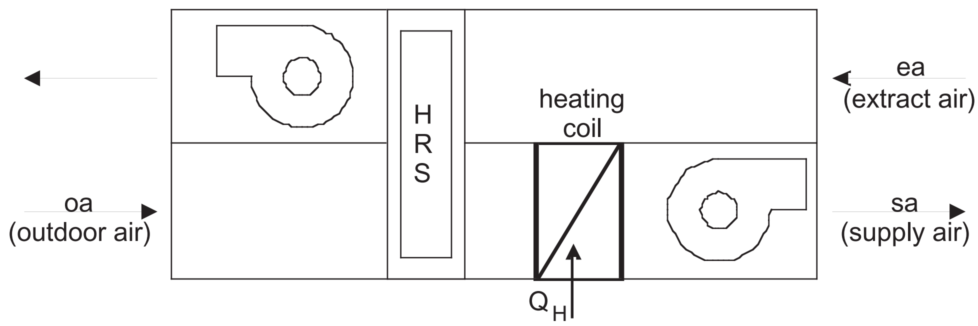
8.4 CALCULATION OF THE ANNUAL CONSUMPTION OF HEATING ENERGY

8.4.1 ASSUMPTIONS FOR CALCULATION PROCEDURE

For the calculation of the annual consumption of heating energy (Q_H) the following assumptions are made:
¥The heating period of the year is defined by outdoor air temperatures (t_{oa}) below 20¡C:

t_{oa} < 20 ¡C

¥ For the standard calculation scheme the following set-up of the AHU is taken as a basis:
Figure 8.4 — Air Handling Unit assumed in the calculation of heating energy



The heating coil is located between the heat recovery system (HRS) and the supply air fan.
The supply air fan induces a temperature rise of Δt_{fan} ¯ 1K of the air flow.
The exhaust air fan is located downstream of the heat recovery system and does not affect the thermal treatment of the supply air
The heat recovery system is defined by the temperature ratio (η_t), which has to be determined under standard conditions according to EN 308. The temperature ratio is taking into account system factors and system effects like defrosting etc. More information on this is given in chapter 6 of this recommendation.

¥The moisture content of the air (x) is defined not to vary in the air handling system. This means that the moisture content of the supply air (x_{sa}) is equal to the moisture content of the outdoor air (x_{oa})

$$x_{oa} = x_{sa}$$

In case of humidification of the supply air, the calculation procedure for the additional annual energy consumption for humidification is given in chapter 8.6 of this recommendation.
The supply air temperature (t_{sa}) depends on the applied heating scenario. In general, these heating scenarios are defined by:

maximum supply air temperature $t_{sa,max}$ at t_{oa} = -10¡C
minimum supply air temperature $t_{sa,min}$ at t_{oa} = 20¡C

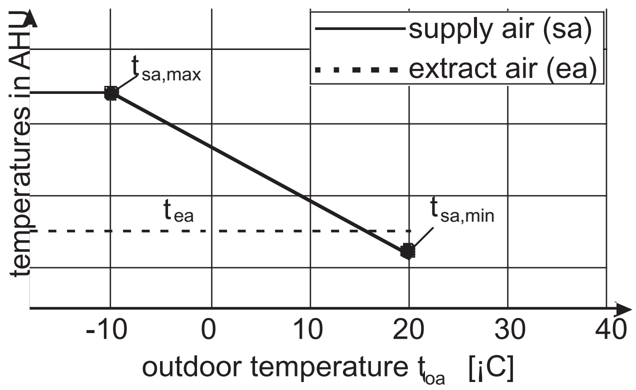


Figure 8.5— Temperature scenario, heating

In order to calculate the annual demand of heating energy, the supply air temperature shall be taken as a function of the outdoor air temperature:

¥t_{oa} < -10¡C: $$t_{sa} = t_{sa,max} \quad (8.10)$$

¥ -10¡C ≤ t_{oa} ≤ 20¡C: $$t_{sa} = t_{sa,max} - \left(\frac{t_{sa,max} - t_{sa,min}}{30}\right) \cdot \left(t_{oa} + 10\right) \quad (8.11)$$

where

t_{sa}	=	temperature of the supply air in ¡C
$t_{sa,max}$	=	max. temperature of the supply air in ¡C
$t_{sa,min}$	=	min. temperature of the supply air in ¡C

¥The extract air temperature (t_{ea}) is constant below 20°C outdoor air temperature and fixed at 22¡C in the manual calculation method.

8.4.2 GENERAL CALCULATION OF ANNUAL HEATING ENERGY DEMAND

Taking into account those assumptions the demand of heating power is calculated as:

$$P_H = \eta_{VAV,HC} \cdot c_p \cdot \rho \cdot q_v \cdot \left[\left(t_{sa} - 1\right) - t_{oa} - \frac{\eta_t}{100} \cdot \left(t_{ea} - t_{oa}\right)_{if>0}\right]_{if>0} \quad (8.12)$$

if no heat recovery: $\eta_t = 0$

where

P_H	=	demand of heating power in kW
ρ	=	density of the air; ρ = 1,2 in kg/m³
c_p	=	specific heat of air; c_p = 1,0 in kJ/kg/…C
q_v	=	air flow rate in m³/s
t_{oa}	=	temperature of the outdoor air in …C
t_{sa}	=	temperature of the supply air in …C
t_{ea}	=	temperature of the extract air in …C

η_t = heat recovery temperature ratio (see chapter 6) in %
$\eta_{VAV,HC}$ = reduction factor for variable air volume (VAV)

To get the annual demand of heating energy, the total sum of the individual heating energies for each hour of the reference year (mean value over a period of 10-30 years) has to be calculated, taking into account the annual operation time of the air handing system during daytime ($t_{op,d}$)and during night-time ($t_{op,n}$):

$$Q_H = \left(\sum_{i=1}^{4380} P_{H,d,i} \cdot 1 \right) \cdot \frac{t_{op,d}}{4380} + \left(\sum_{i=1}^{4380} P_{H,n,i} \cdot 1 \right) \cdot \frac{t_{op,n}}{4380} \qquad (8.13)$$

where
Q_H = demand of annual heating energy in kWh/a
$P_{H,d,i}$ = demand of heating power during daytime at the hour i in kW
$P_{H,n,i}$ = demand of heating power during night-time at the hour i in kW
$t_{op,d}$ = annual operation time during daytime in h/a
$t_{op,n}$ = annual operation time during night-time in h/a
1 = time in h

Total annual costs for heating:

The total annual energy costs for heating (E_H) can be calculated as:

$$E_H = Q_H \cdot p_{heating} \qquad (8.14)$$

where
E_H = total annual energy costs for heating in €/a
Q_H = total demand of annual heating energy in kWh/a
$p_{heating}$ = price for heating energy [€/kWh] taking into account system effects and efficiencies (see chapter 5)

8.4.3 CALCULATION METHODS

The calculation of annual heating demand can be done either by means of the software program EUROVENT-AHU (see Annex C) or manually according to method below.

Accuracies: Manual method – 5 % in relation to software program.

In case of high temperature ratio and low supply air temperature the deviation may be higher.

8.4.4 MANUAL CALCULATION PROCEDURE FOR ANNUAL HEATING DEMAND

To take into account the significant difference of the heating energy demands for buildings with respect to the daily operation times it is recommended to calculate the annual energy demands for two periods each day:
¥ daytime operation: 6.00 — 18.00
¥ night-time operation : 18.00 — 6.00

Each air handling system will have an individual mix of daytime and night-time operation, depending on the type and utilization of the building. Therefore it is necessary first to calculate the general heating energy demands for daytime operation ($Q_{H,d}$) and night-time operation ($Q_{H,n}$) and then to add those figures to take into account the typical operation times of the system.

Annual demand for heating energy during daytime:

Introducing the following correlation factors for heating during daytime:

¥ $H_{sa,d} = \sum_{i=1}^{4380} \left(t_{sa,i} - 1\right)$ for $t_{oa,i} \leq (t_{sa,i} — 1)$ (8.15)

¥ $H_{oa,d} = \sum_{i=1}^{4380} t_{oa,i}$ for $t_{oa,i} \leq (t_{sa,i} — 1)$ (8.16)

¥ $(H_{ea} - H_{oa})_d = \sum_{i=1}^{4380} (t_{ea,i} - t_{oa,i})$ for $(t_{sa,i} — 1) — t_{oa,i}$ ‡ $0.38\ (t_{ea,i} — t_{oa,i})$ (8.17)

where

¥ $H_{sa,d}$ = heating correlation factor, supply air, daytime in K·h/a

¥ $H_{oa,d}$ = heating correlation factor, outdoor air, daytime in K·h/a

¥ $(H_{ea}$-$H_{oa})_d$ = heating correlation factor, difference between extract air and outdoor air, daytime in K·h/a

$t_{sa,i}$ = temperature of supply air at the hour i in ¡C

$t_{oa,i}$ = temperature of outdoor air at the hour i in ¡C

$t_{ea,i}$ = temperature of extract air at the hour i in ¡C

0.38 = a summation limiting factor to increase the accuracy.

The annual demand of heating energy during daytime operation can be calculated as:

$$Q_{H,d} = 1{,}2 \cdot \eta_{VAV,HC} \cdot q_v \cdot c_p \cdot \left[H_{sa,d} - H_{oa,d} - \frac{\eta_{t,s}}{100}(H_{ea} - H_{oa})_d\right] \cdot \frac{t_{op,d}}{4380} \qquad (8.18)$$

where

$Q_{H,d}$ = demand of annual heating energy during daytime in kWh/a

q_v = air flow rate in m^3/s

$\eta_{VAV,HC}$ = reduction factor in case of VAV

c_p = specific heat of air; c_p = 1.0 in kJ/kg›K

$H_{sa,d}$ = heating correlation factor, supply air, daytime in K·h/a

$H_{oa,d}$ = heating correlation factor, outdoor air, daytime in K·h/a

$(H_{ea} - H_{oa})_d$ = heating correlation factor, difference between extract air and outdoor air, daytime in K·h/a

$t_{op,d}$ = annual operation time during daytime from 6.00 to 18.00 in h/a

$\eta_{t,s}$ = seasonal temperature ratio in %

Note that the correlation factors for heating during daytime; $H_{sa,d}$, $H_{oa,d}$ and $(H_{ea} - H_{oa})_d$ are dependent upon the location of the AHU and on the supply air temperature scenario.

Annual demand for heating energy during night time:

Introducing the following correlation factors for heating during night-time:

• $H_{sa,n} = \sum_{i=1}^{4380} \left(t_{sa,i} - 1\right)$ for $t_{oa,i} \leq (t_{sa,i} — 1)$ (8.19)

- $H_{oa,n} = \sum_{i=1}^{4380} t_{oa,i}$ for $t_{oa,i} \leq (t_{sa,i} - 1)$ (8.20)

- $(H_{ea} - H_{oa})_n = \sum_{i=1}^{4380} (t_{ea,i} - t_{oa,i})$ for $(t_{sa,i} - 1) - t_{oa,i} \geq 0.38\ (t_{ea,i} - t_{oa,i})$ (8.21)

where

$H_{sa,n}$	=	heating correlation factor, supply air, night-time in K·h/a
$H_{oa,n}$	=	heating correlation factor, outdoor air, night-time in K·h/a
$(H_{ea} - H_{oa})_n$	=	heating correlation factor, difference between extract air and outdoor air, night-time in K·h/a
$t_{sa,i}$	=	temperature of supply air at the hour i in ¡C
$t_{oa,i}$	=	temperature of outdoor air at the hour i in ¡C
$t_{ea,i}$	=	temperature of extract air at the hour i in ¡C
0.38	=	a summation limiting factor to increase the accuracy.

The annual demand of heating energy during night-time operation can be calculated as:

$$Q_{H,n} = 1{,}2 \cdot \eta_{VAV,HC} \cdot q_v \cdot c_p \cdot \left[H_{sa,n} - H_{oa,n} - \frac{\eta_{t,s}}{100} (H_{ea} - H_{oa})_n \right] \cdot \frac{t_{op,n}}{4380} \qquad (8.22)$$

where

$Q_{H,n}$	=	demand of annual heating energy during night-time in kWh/a
q_v	=	air flow rate in m³/s
$\eta_{VAV,HC}$	=	reduction factor in case of VAV
c_p	=	specific heat of air; c_p = 1.0 in kJ/kg·K
$H_{sa,n}$	=	heating correlation factor, supply air, night-time in K·h/a
$H_{oa,n}$	=	heating correlation factor, outdoor air, night-time in K·h/a
$(H_{ea} - H_{oa})_n$	=	heating correlation factor, difference between extract air and outdoor air, night-time in K·h/a
$t_{op,n}$	=	annual operation time during night-time from 18.00 to 06.00 in h/a
$\eta_{t,s}$	=	seasonal temperature ratio in %

Note that the correlation factors for heating during night-time; $H_{sa,n}$, $H_{oa,n}$ and $(H_{ea} - H_{oa})_n$ are dependent upon the location of the AHU and on the supply air temperature scenario.

In Annex A the correlation factors for heating $H_{sa,d}$; $H_{oa,d}$; $(H_{ea} - H_{oa})_d$; $H_{sa,n}$; $H_{oa,n}$; $(H_{ea} - H_{oa})_n$ as a function of the supply air temperature scenario are given for a number of European locations.

Total annual demand for heating energy:

The total annual demand for heating energy is calculated by adding the annual demand during daytime and during night-time:

$$Q_H = Q_{H,d} + Q_{H.n} \qquad (8.23)$$

where

Q_H	=	total demand of annual heating energy in kWh/a
$Q_{H,d}$	=	demand of heating energy during daytime in kWh/a
$Q_{H,n}$	=	demand of heating energy during night-time in kWh/a

8.5 Calculation of the Annual Consuption of Cooling Energy

8.5.1 Assumptions for Calculation Procedure

For the calculation of the annual consumption of cooling energy (Q_C) the following assumptions are made:

• The cooling period of the year is defined by outdoor air temperatures t_{oa} above 20°C:

$$t_{oa} > 20°C$$

• For the standard calculation scheme the following set-up of the AHU is taken as a basis:

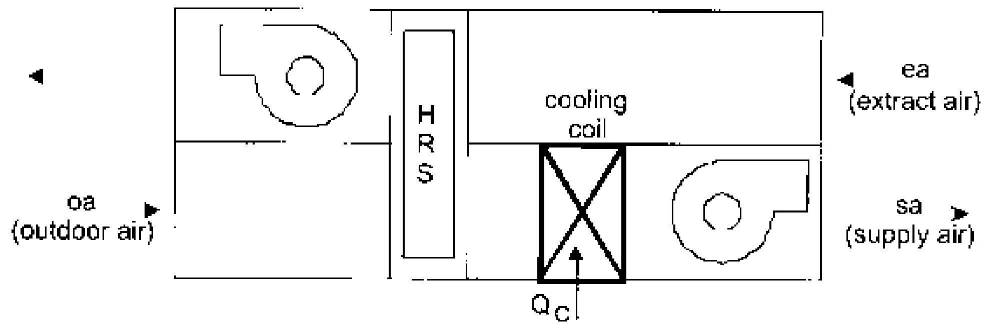


Figure 8.6 – Air Handling Unit assumed in the calculation of cooling energy

The cooling coil is located between the heat recovery system (HRS) and the supply air fan. The supply air fan induces a temperature rise of $\Delta t_{far} \approx$ 1K of the air flow.
The exhaust air fan is located downstream of the heat recovery system and does not affect the thermal treatment of the supply air.
The cooling recovery system is defined by the temperature ratio (η_t) and the humidity ratio (η_x) which has to be determined under standard conditions according to EN 308. More information on this will be given in chapter 6 of this recommendation.

Temperature scenario

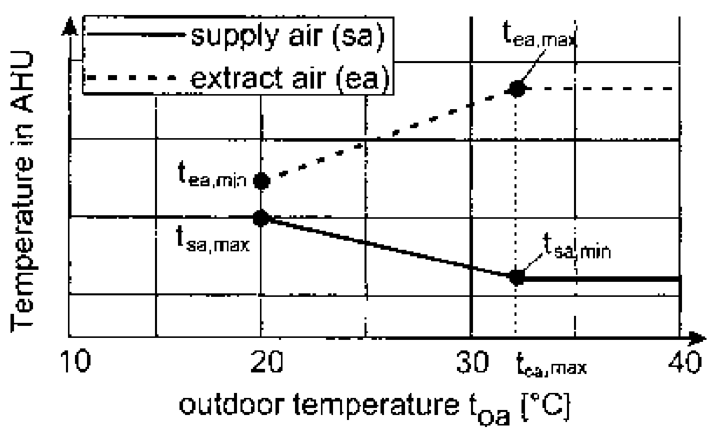


Figure 8.7 – Temperature scenario, cooling

The supply air temperature (t_{sa}) and the extract air temperature (t_{ea}) are depending on the applied cooling scenario. In general, these cooling scenarios are defined by:

- maximum supply air temperature $t_{sa,max}$ at t_{oa} = 20°C
- minimum supply air temperature $t_{sa,min}$ at t_{oa} = maximum outdoor temperature ($t_{oa,max}$)
- maximum extract air temperature $t_{ea,max}$ at t_{oa} = maximum outdoor temperature ($t_{oa,max}$)
- minimum extract air temperature $t_{ea,min}$ at t_{oa} = 20°C
- in the manual calculation method the extract temperatures are fixed at 22°C and 27°C at respectively 20°C and 32°C outdoor temperature

In order to calculate the annual demand of cooling energy, the supply air temperature and the extract air temperature have to be taken as a function of the outdoor air temperature:

supply air temperature:

- $20°C \le t_{oa} \le t_{oa,max}$:

$$t_{sa} = t_{sa\,max} - \frac{(t_{oa} - 20)}{(t_{oa\,max} - 20)} \cdot (t_{sa,max} - t_{sa,min}) \quad (8.24)$$

- $t_{oa} > t_{oa,max}$:

$$t_{sa} = t_{sa,min} \quad (8.25)$$

where

t_{sa} = temperature of the supply air in °C
$t_{sa,max}$ = max. temperature of the supply air in °C
$t_{sa,min}$ = min. temperature of the supply air in °C
t_{oa} = temperature of the outdoor air in °C
$t_{oa,max}$ = max. temperature of the outdoor air in °C

extract air temperature:

- $20°C \le t_{oa} \le t_{oa,max}$:

$$t_{ea} = t_{ea,min} + \frac{(t_{oa} - 20)}{(t_{oa,max} - 20)} \cdot (t_{ea,max} - t_{ea,min}) \quad (8.26)$$

- $t_{oa} > t_{oa,max}$:

$$t_{ea} = t_{ea,max} \quad (8.27)$$

where

t_{ea} = temperature of the extract air in °C
$t_{ea,max}$ = max. temperature of the extract air in °C
$t_{ea,min}$ = min. temperature of the extract air in °C
t_{oa} = temperature of the outdoor air in °C
$t_{oa,max}$ = max. temperature of the outdoor air in °C

<u>Moisture scenario:</u>

Comfort climate:

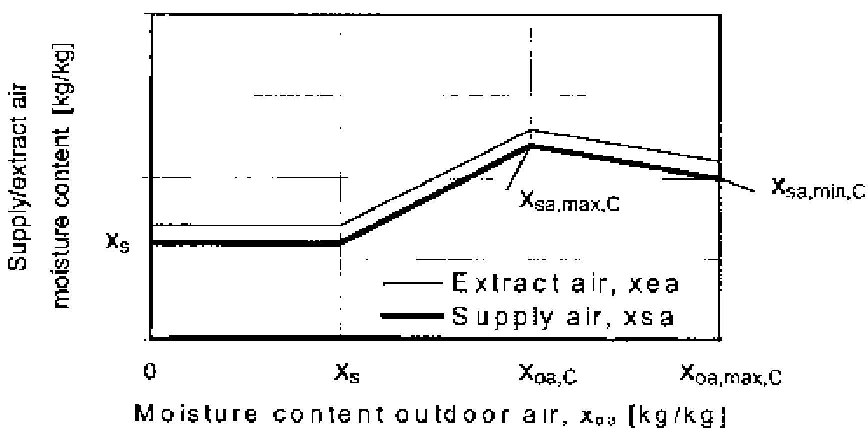


Figure 8.8 – Moisture scenario, comfort climate

The moisture content of supply air (x_{sa}) and its dependence upon moisture content of outdoor air (x_{oa}) can be defined by:

x_s = humidification set point downstream the humidifier. If there is no humidifier and no moisture recovery, the moisture content of the supply air is equal to the moisture content of the outdoor air up to the point $x_{oa,C}$. The line x_{sa} will intersect the origin of the graph.

$x_{oa,C}$ = moisture content of outdoor air when dehumidification commences in the cooling period.

$x_{oa,max,C}$ = maximum moisture content of outdoor air for the location of interest.

$x_{sa,max,C}$ and $x_{sa,min,C}$ = moisture content according to tables below (see also chapter 5 and appendix A): (if no latent cooling recovery $x_{sa,max} = x_{oa,c}$)

Table 8.2 Moisture content in supply air at different max and min supply air temperatures

	$(t_{sa}-1)_{max,C}$								
	22 °C	21 °C	20 °C	19 °C	18 °C	17 °C	16 °C	15 °C	14 °C
$X_{sa,max,C}$ [g/kg]	13,0	12,5	12,0	11,5	11,0	10,5	10,0	9,5	9,0
	$(t_{sa}-1)_{min,C}$								
	19 °C	18 °C	17 °C	16 °C	15 °C	14 °C	13 °C	12 °C	11 °C
$X_{sa,min,C}$ [g/kg]	13,0	12,0	11,5	11,0	10,0	9,5	9,0	8,5	8,0

Between these defined points the moisture content of supply air can be calculated as linear in relation to the moisture content of the outdoor air.

$$x_{ea} = x_{sa} + x_{rl} \quad (8.28)$$

where

x_{ea} = moisture content in extract air in kg/kg

x_{sa} = moisture content in supply air in kg/kg

x_{rl} = expected increase of moisture content due to room loads in kg/kg fixed at 1 g/kg in manual calculation method

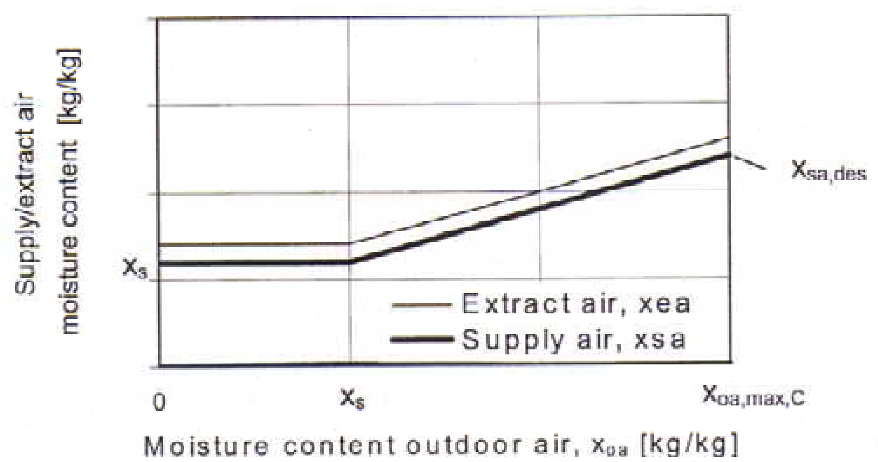


Figure 8.9 – Moisture scenario, industrial climate

The moisture content of supply air (x_{sa}) and its dependence upon moisture content of outdoor air (x_{oa}) can be defined by:

x_s = humidification, set point downstream the humidifier.

$x_{oa,max,C}$ = maximum moisture content of outdoor air for the location of interest.

$x_{sa,des}$ = Moisture content design value of supply air at $x_{oa,max,C}$
This value is always lower than the moisture content of the saturated supply air at minimum temperature

Between these defined points the moisture content of supply air can be calculated as linear in relation to the moisture content of the outdoor air.

8.5.2 GENERAL CALCULATION OF ANNUAL COOLING ENERGY DEMAND

Two operation modes can be defined:

1. **Temperature control:** Outdoor air is cooled down to the required temperature of supply air. The moisture content follows the supply air temperature and and is not controlled. Some dehumidification will occur, depending on selected temperature scenario.

2. **Moisture control:** Outdoor air is cooled down to the required moisture content of supply air. If the supply air temperature is below the required temperature the air is reheated.

The two operation modes in the Mollier-diagram :

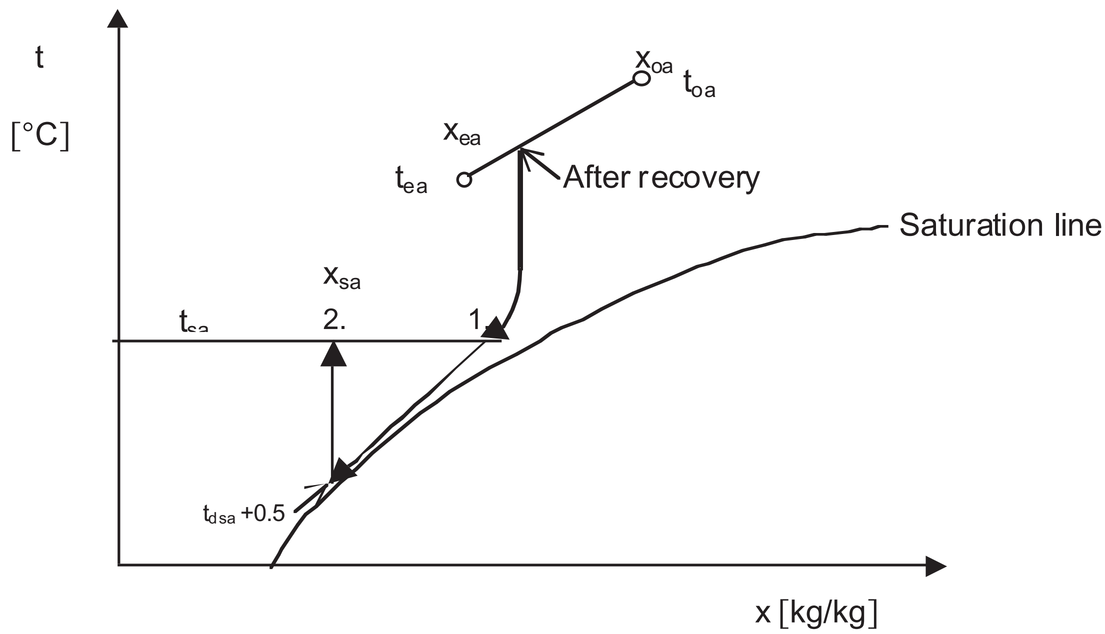


Figure 8.10 — Operation modes in the Mollier-diagram

Taking into account assumptions as above the cooling demands (split up in a sensible and a latent part) and reheating if required, are calculated as below.

Temperature control:

$$P_{C,S} = \eta_{VAV,HC} \cdot \rho \cdot q_v \cdot c_p \cdot \left[t_{oa} - REC_S - (t_{sa} - 1)\right] \tag{8.29}$$

$$P_{C,L} = \eta_{VAV,HC} \cdot \rho \cdot q_v \cdot 2500 \cdot (x_{oa} - REC_L - x_{sa}) \tag{8.30}$$

$$P_{Hre} = 0 \tag{8.31}$$

Moisture control:

$$P_{C,S} = \eta_{VAV,HC} \cdot \rho \cdot q_v \cdot c_p \cdot \left[t_{oa} - REC_S - (t_{dsa} + 0.5)\right] \tag{8.32}$$

$$P_{C,L} = \eta_{VAV,HC} \cdot \rho \cdot q_v \cdot 2500 \cdot (x_{oa} - REC_L - x_{sa}) \tag{8.33}$$

$$P_{Hre} = \eta_{VAV,HC} \cdot \rho \cdot q_v \left[(t_{sa} - 1) - (t_{dsa} + 0.5)\right] \tag{8.34}$$

where

$P_{C,S}$	=	sensible cooling demand in kW
$P_{C,L}$	=	latent cooling demand in kW
P_{Hre}	=	reheating demand in kW
$\eta_{VAV,HC}$	=	reduction factor for variable air volume (VAV)
ρ	=	density of the air; ρ = 1,2 in kg/m³
q_v	=	supply air flow rate in m³/s
t_{dsa} + 0,5	=	dew point temperature at x_{sa}+ 0,5 °C to compensate the saturation deficit (≈95 %) in the outlet of the cooling coil in °C
REC_S	=	sensible cooling recovery in °C $REC_S = c_p \cdot (t_{oa} - t_{ea})_{if>o} \cdot \frac{\eta_t}{100}$ if no cooling recovery: REC_S = 0
REC_L	=	Latent cooling recovery in kg/kg $REC_L = (x_{oa} - x_{ea})_{if>0} \cdot \frac{\eta_x}{100}$ if no latent recovery system: REC_L =0
t_{sa}	=	supply air temperature in °C
t_{ea}	=	extract air temperature in °C
t_{oa}	=	outdoor air temperature and fixed at 22¡C in the manual calculation method in °C
x_{sa}	=	moisture content of supply air in kg/kg
x_{ea}	=	moisture content of extract air in kg/kg
x_{oa}	=	moisture content of outdoor air in kg/kg
η_t	=	temperature ratio in %
η_x	=	humidity ratio in %
c_p	=	specific heat of air; c_p = 1,0 in kJ/kg·K
2500	=	latent heat of moisture in kJ/kg

Here the sensible heat of water vapour is neglected

To get the annual demand of cooling energy, the total sum of the individual cooling energies for each hour of the reference year (mean value over a period of 10-30 years) has to be calculated, taking into account the annual operation time of the air handing system during daytime ($t_{op,d}$) and during night-time ($t_{op,n}$):

$$Q_C = \left(\sum_{i=1}^{4380} (P_{C,S,d,i} + P_{C,L,d,i}) \cdot 1 \right) \cdot \frac{t_{op,d}}{4380} + \left(\sum_{i=1}^{4380} (P_{C,S,n,i} + P_{C,L,n,i}) \cdot 1 \right) \cdot \frac{t_{op,n}}{4380} \tag{8.35}$$

where

Q_C	=	total demand of annual cooling energy in kWh/a
$P_{C,S,d,i}$	=	sensible cooling demand during daytime at the hour i in kW
$P_{C,S,n,i}$	=	sensible cooling demand during night-time at the hour i in kW
$P_{C,L,d,i}$	=	latent cooling demand during daytime at the hour i in kW
$P_{C,L,n,i}$	=	latent cooling demand during night-time at the hour i in kW
$t_{op,d}$	=	annual operation time during daytime in h/a
$t_{op,n}$	=	annual operation time during night-time in h/a
1	=	time in h

Annual reheating:

$$Q_{Hre} = \left(\sum_{i=1}^{4380} P_{Hre,d,i} \cdot 1 \right) \cdot \frac{t_{op,d}}{4380} + \left(\sum_{i=1}^{4380} P_{Hre,n,i} \cdot 1 \right) \cdot \frac{t_{op,n}}{4380} \quad \left[\frac{kWh}{a} \right] \tag{8.36}$$

where

Q_{Hre}	=	total demand of annual reheating energy in kWh/a
$P_{Hre,d,i}$	=	demand of reheating power during daytime at the hour i in kW
$P_{Hre,n,i}$	=	demand of reheating power during night-time at the hour i in kW
$t_{op,d}$	=	annual operation time during daytime in h/a
$t_{op,n}$	=	annual operation time during night-time in h/a
1	=	time in h

<u>total annual costs for cooling:</u>

The total annual energy costs for cooling (E_C) and reheating (E_{Hre})can be calculated as:

$$E_C = Q_C \cdot p_{cooling} \tag{8.37}$$

$$E_{Hre} = Q_{Hre} \cdot p_{heating} \tag{8.38}$$

where

Q_C	=	total demand of annual cooling energy in kWh/a
$p_{cooling}$	=	price for cooling energy [€/kWh] taking into account system effects and efficiencies (see chapter 5)
Q_{Hre}	=	total demand of annual reheating energy in kWh/a
$p_{heating}$:	=	price for heating energy [€/kWh] taking into account system effects and efficiencies (see chapter 5)

8.5.3 CALCULATION METHODS

The calculation of annual cooling demand can be done either by means of the software program EUROVENT-AHU (see Annex C) or manually for temperature controlled systems according to method below.

<u>Accuracies:</u> Manual method – 10 % in relation to software program.

8.5.4 MANUAL CALCULATION PROCEDURE FOR ANNUAL COOLING DEMAND

To take into account the significant difference of the cooling energy demands for buildings with respect to the daily operation times it is recommended to calculate the annual energy demands for two periods each day:

¥ daytime operation: 6.00 — 18.00

¥ night-time operation : 18.00 — 6.00

Each air handling system will have an individual mix of daytime and night-time operation, depending on the type and utilization of the building. Therefore it is necessary first to calculate the general cooling energy demands for daytime operation ($Q_{C,d}$) and night-time operation ($Q_{C,n}$) and then to add those figures to take into account the typical operation times of the system.

<u>annual demand for cooling energy during daytime:</u>

Introducing the following correlation factors for sensible and latent cooling during daytime:

<u>Sensible cooling</u>:

$$C_{S,oa,d} = \sum_{i=1}^{4380} t_{oa,i} \tag{8.39}$$

$$C_{S,sa,d} = \sum_{i=1}^{4380} \left(t_{sa,i} - 1\right) \tag{8.40}$$

$$\left(C_{S,oa} - C_{S,ea}\right)_d = \sum_{i=1}^{4380} \left(t_{oa,i} - t_{ea,i}\right) \tag{8.41}$$

where

$C_{S,oa,d}$	=	sensible cooling correlation factor, outdoor air, daytime K·h/a
$C_{S,sa,d}$	=	sensible cooling correlation factor, supply air, daytime K·h/a
$(C_{S,oa} - c_{S,ea})_d$	=	sensible cooling correlation factor, difference between outdoor air and extract air, daytime K·h/a
$t_{oa,i}$	=	temperature of outdoor air at the hour i in ¡C
$t_{sa,i}$	=	temperature of supply air at the hour i in ¡C
$t_{ea,i}$	=	temperature of extract air at the hour i in ¡C

The annual demand of sensible cooling energy during daytime operation can be calculated as:

$$Q_{C,S,d} = 1{,}2 \cdot \eta_{VAV,HC} \cdot q_v \cdot c_p \cdot \left(C_{S,oa,d} - C_{S,sa,d} - \frac{\eta_{t,s}}{100} \left(C_{S,oa} - C_{S,ea}\right)_d \right) \cdot \frac{t_{op,d}}{4380} \tag{8.42}$$

where

$Q_{C,S,d}$	=	demand of annual sensible cooling energy, daytime in kWh/a
q_v	=	air flow rate in m^3/s
c_p	=	specific heat of air in kJ/kg·K
$\eta_{VAV,HC}$	=	reduction factor in case of VAV
$t_{op,d}$	=	annual operation time during daytime from 6.00 to 18.00 in h/a
$C_{S,oa,d}$	=	correlation factors for sensible cooling during daytime, outdoor air in K·h/a
$C_{S,sa,d}$	=	correlation factors for sensible cooling during daytime, supply air in K·h/a
$(C_{S,oa} - c_{S,ea})_d$	=	correlation factor for sensible cooling, difference between outdoor air and extract air, daytime K·h/a
$\eta_{t,s}$	=	seasonal temperature ratio in %

Note that the correlation factors for sensible cooling during daytime; $C_{S,oa,d}$, $C_{S,sa,d}$ and $(C_{S,oa} - C_{S,ea})_d$ are depending on location of the AHU and on the supply air temperature scenario.

<u>Latent cooling:</u>

$$C_{L,oa,d} = \sum_{i=1}^{4380} x_{oa,i} \quad (8.43)$$

$$C_{L,sa,d} = \sum_{i=1}^{4380} x_{sa,i} \quad (8.44)$$

$$(C_{L,oa} - C_{L,ea})_d = \sum_{i=1}^{4380} (x_{oa,i} - x_{ea,i}) \quad (8.45)$$

where

$C_{L,oa,d}$	=	latent cooling correlation factor, outdoor air, daytime K·h/a
$C_{L,sa,d}$	=	latent cooling correlation factor, supply air, daytime K·h/a
$(C_{L,oa} - C_{L,ea})_d$	=	latent cooling correlation factor, difference between outdoor air and extract air, daytime K·h/a
$x_{oa,i}$	=	moisture content of outdoor air at the hour i in ¡C
$x_{sa,i}$	=	moisture content of supply air at the hour i in ¡C
$x_{ea,i}$	=	moisture content of extract air at the hour i in ¡C
$x_{ea,i}$	=	$x_{sa,i} + 0.001$

The annual demand of latent cooling energy during daytime operation can be calculated as:

$$Q_{C,L,d} = 1{,}2 \cdot \eta_{VAV,HC} \cdot q_v \cdot 2500 \cdot \left(C_{L,oa,d} - C_{L,sa,d} - \frac{\eta_{x,s}}{100}(C_{L,oa} - C_{L,ea})_d \right) \cdot \frac{t_{op,d}}{4380} \quad (8.46)$$

where

$Q_{C,L,d}$	=	demand of annual latent cooling energy, daytime in kWh/a
q_v	=	air flow rate in m^3/s
c_p	=	specific heat of air in kJ/kg·K
$\eta_{VAV,HC}$	=	reduction factor in case of VAV
$t_{op,d}$	=	annual operation time during daytime from 6.00 to 18.00 in h/a
$C_{L,oa,d}$	=	correlation factors for latent cooling during daytime, outdoor air in K·h/a
$C_{L,sa,d}$	=	correlation factors for latent cooling during daytime, supply air in K·h/a
$(C_{L,oa} - C_{L,ea})_d$	=	correlation factor for latent cooling, difference between outdoor air and extract air, daytime K·h/a
$\eta_{x,s}$	=	seasonal humidity ratio in %

Note that the correlation factors for latent cooling during daytime; $C_{L,oa,d}$, $C_{L,sa,d}$ and $(C_{L,oa} — C_{L,ea})_d$ are dependent upon the location of the AHU and on the supply air temperature (moisture) scenario.

<u>Annual demand for cooling energy during night-time:</u>

Introducing the following correlation factors for sensible and latent cooling during night-time:

Sensible cooling:

$$C_{S,oa,n} = \sum_{i=1}^{4380} t_{oa,i} \tag{8.47}$$

$$C_{S,sa,n} = \sum_{i=1}^{4380} (t_{sa,i} - 1) \tag{8.48}$$

$$(C_{S,oa} - C_{S,ea})_n = \sum_{i=1}^{4380} (t_{oa,i} - t_{ea,i}) \tag{8.49}$$

where

$C_{S,oa,n}$	=	sensible cooling correlation factor, outdoor air, night-time K·h/a
$C_{S,sa,n}$	=	sensible cooling correlation factor, supply air, night-time K·h/a
$(C_{S,oa} - C_{S,ea})_n$	=	sensible cooling correlation factor, difference between outdoor air and extract air, night-time K·h/a
$t_{oa,i}$	=	temperature of outdoor air at the hour i in °C
$t_{sa,i}$	=	temperature of supply air at the hour i in °C
$t_{ea,i}$	=	temperature of extract air at the hour i in °C

The annual demand of sensible cooling energy during night time operation can be calculated as:

$$Q_{C,S,n} = 1{,}2 \cdot \eta_{VAV,HC} \cdot q_v \cdot c_p \cdot \left(C_{S,oa,n} - C_{S,sa,n} - \frac{\eta_{t,s}}{100} (C_{S,oa} - C_{S,ea})_n \right) \cdot \frac{t_{op,n}}{4380} \quad (8. \tag{8.50}$$

where

$Q_{C,S,n}$	=	demand of annual sensible cooling energy, night-time in kWh/a
q_v	=	air flow rate in m³/s
c_p	=	specific heat of air in kJ/kg·K
$\eta_{VAV,HC}$	=	reduction factor in case of VAV
$t_{op,n}$	=	annual operation time during night-time from 18.00 to 6.00 in h/a
$C_{S,oa,n}$	=	correlation factors for sensible cooling during night-time, outdoor air in K·h/a
$C_{S,sa,n}$	=	correlation factors for sensible cooling during night-time, supply air in K·h/a
$(C_{S,oa} - C_{S,ea})_n$	=	correlation factor for sensible cooling, difference between outdoor air and extract air, night-time K·h/a
$\eta_{t,s}$	=	seasonal temperature ratio in %

Note that the correlation factors for sensible cooling during night-time; $C_{S,oa,n}$, $C_{S,sa,n}$ and $(C_{S,oa} - C_{S,ea})_n$ are dependent upon the location of the AHU and on the supply air temperature scenario.

Latent cooling:

$$C_{L,oa,n} = \sum_{i=1}^{4380} x_{oa,i} \tag{8.51}$$

$$C_{L,oa,n} = \sum_{i=1}^{4380} x_{oa,i} \tag{8.52}$$

$$C_{L,oa,n} = \sum_{i=1}^{4380} x_{oa,i} \tag{8.53}$$

where

$C_{L,oa,n}$ = latent cooling correlation factor, outdoor air, night-time K·h/a

$C_{L,sa,n}$ = latent cooling correlation factor, supply air, night-time K·h/a

$(C_{L,oa} - C_{L,ea})_n$ = latent cooling correlation factor, difference between outdoor air and extract air, night-time K·h/a

$x_{oa,i}$ = moisture content of outdoor air at the hour i in ¡C

$x_{sa,i}$ = moisture content of supply air at the hour i in ¡C

$x_{ea,i}$ = moisture content of extract air at the hour i in ¡C

$x_{ea,i}$ = $x_{sa,i}$ + 0.001

The annual demand of latent cooling energy during night time operation can be calculated as:

$$Q_{C,L,n} = 1{,}2 \cdot \eta_{VAV,HC} \cdot q_v \cdot 2500 \cdot \left(C_{L,oa,n} - C_{L,sa,n} - \frac{\eta_{x,s}}{100}(C_{L,oa} - C_{L,ea})_n \right) \cdot \frac{t_{op,n}}{4380} \qquad (8.54)$$

where

$Q_{C,L,n}$ = demand of annual latent cooling energy, night-time in kWh/a

q_v = air flow rate in m^3/s

c_p = specific heat of air in kJ/kg·K

$\eta_{VAV,HC}$ = reduction factor in case of VAV

$t_{op,n}$ = annual operation time during night-time from 18.00 to 6.00 in h/a

$C_{L,oa,n}$ = correlation factors for latent cooling during night-time, outdoor air in K·h/a

$C_{L,sa,n}$ = correlation factors for latent cooling during night-time, supply air in K·h/a

$(C_{L,oa} - C_{L,ea})_n$ = correlation factor for latent cooling, difference between outdoor air and extract air, night-time K·h/a

$\eta_{x,s}$ = seasonal humidity ratio in %

Note that the correlation factors for latent cooling during daytime; $C_{L,oa,d}$, $C_{L,sa,d}$ and $(C_{L,oa} — C_{L,ea})_d$ are dependent upon the location of the AHU and on the supply air temperature (moisture) scenario.

In Annex A the correlation factors for cooling $C_{S,sa,d}$; $C_{S,oa,d}$; $(C_{S,oa} — C_{S,ea})_{d;}$; $C_{L.sa,d}$; $C_{L,oa,d}$; $(C_{L,oa} — C_{L\cdot ea})_d$; $C_{S,sa,n}$; $C_{S,oa,n}$; $(C_{S,oa} — C_{S,ea})_{n;}$; $C_{L.sa,n}$; $C_{L.oa,n}$ and $(C_{L,oa} — C_{L,ea})_n$ as a function of the supply air temperature scenario are given for a number of European locations.

<u>Total annual demand for cooling energy:</u>

The total annual demand for cooling energy is calculated by adding the annual demand during daytime and during night-time:

$$Q_C = Q_{C,S,d} + Q_{C,L,d} + Q_{C,S,n} + Q_{C,L,n} \qquad (8.55)$$

where

Q_C = total demand of annual cooling energy in kWh/a

$Q_{C,S,d}$ = demand of annual sensible cooling energy, daytime in kWh/a

$Q_{C,L,d}$ = demand of annual latent cooling energy, daytime in kWh/a

$Q_{C,S,n}$ = demand of annual sensible cooling energy, night-time in kWh/a

$Q_{C,L,n}$ = demand of annual latent cooling energy, night-time in kWh/a

8.6 Calculation of the Annual Thermal Energy Consumption for Humidification

8.6.1 Assumptions for Calculation Procedures

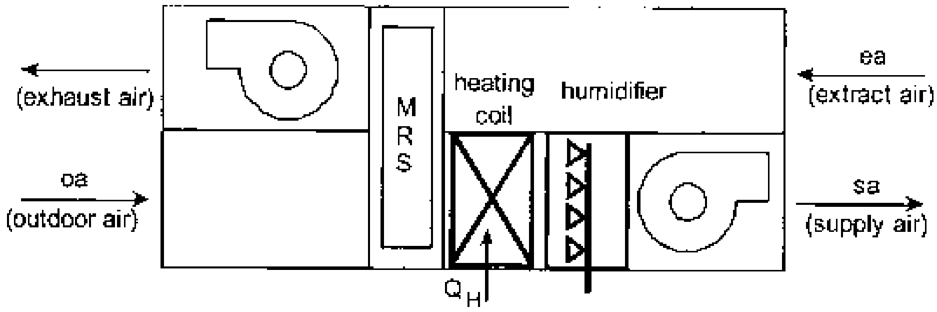


Figure 8.11 – Air Handling Unit assumed in the calculation of energy consumption for humidification

For the calculation of the annual consumption of thermal energy for humidification (Q_{hum}), the following assumptions have been made:

- The thermal energy consumption is considered to be the expended energy to evaporate the water, required to achieve the desired moisture content of the supply air during the humidification season. The definition makes it irrelevant whether steam or water humidifiers are considered!
- The humidification season is defined by the hours during which the moisture content of the outdoor air is below the set point for moisture content of the supply air.

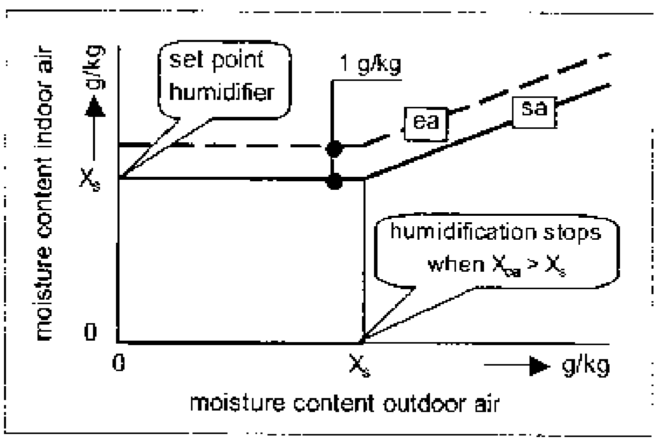


Figure 8.12 – Humidification season

- The moisture production in the building results into a moisture gain in the extract air of 1 g/kg.
- For the standard calculation scheme the following AHU configuration has been taken as a basis:
- The humidifier is located between the heating coil and the supply fan. A moist ure recovery system (MRS, if applied) is always located upstream the heating coil. The performance of any moisture recovery system is defined by the seasonal moisture recovery efficiency $\eta_{x,s}$ as described in chapter 7.
- The moisture scenario for humidification is based on a constant set point condition of the supply air during the humidification season. In order to calculate the annual demand for thermal energy for humidification the moisture content of the supply air shall be taken as a function of the moisture content of the outdoor air. Moisture content of the extract air is always 1 g/kg higher than set point moisture content of supply air.

8.6.2 General Calculation of Annual Energy Demand for Humidification

The momentary (power)consumption of thermal energy for humidification is calculated with:

$$P_{mom} = \eta_{VAV} \cdot q_v \cdot \rho \cdot 2500 \cdot \left[(x_{sa} - x_{oa}) - \frac{\eta_x}{100} (x_{ea} - x_{oa}) \right] \qquad (8.56)$$

where

P_{mom} = momentary thermal energy power demand in kW

η_{VAV} = reduction factor for variable air volume system (VAV)

q_V = air flow rate in m^3/s

ρ = density of the air, established at 1,2 kg/m^3

2500 = evaporation heat of water in kJ/kg

x_{sa} = moisture content of the supply air in kg/kg dry air

x_{oa} = moisture content of the outdoor air in kg/kg dry air

x_{ea} = moisture content of the extract air in kg/kg dry air

η_x = humidity ratio at considered condition in %

To establish the annual consumption of thermal energy for humidification, the total sum of the individual energy demands for each hour of the reference year has to be computed, taking into account the annual operation times of the air handling system during day time ($t_{op,d}$) and during night time ($t_{op,n}$):

$$Q_{hum} = \left(\sum_{i=1}^{4380} P_{mom,d,i} \cdot 1 \right) \frac{t_{op,d}}{4380} + \left(\sum_{j=1}^{4380} P_{mom,n,j} \cdot 1 \right) \frac{t_{op,n}}{4380} \qquad (8.57)$$

where

Q_{hum} = annual thermal energy consumption for humidification in kWh/a

P_{mom} = momentary thermal energy power demand in kW

1 = time in h

Total annual costs for humidification:

The total annual energy costs for humidification (E_{hum}) can be calculated as:

$$E_{hum} = Q_{hum} \cdot p_{hum} \qquad (8.58)$$

where

E_{hum} = total annual energy costs for humidification in €

Q_{hum} = annual thermal energy consumption for humidification in kWh/a

p_{hum} = price for thermal energy for humidification, taking into account system effects and efficiencies (see chapter7) [€/kWh]

8.6.3 CALCULATION METHODS

The calculation of annual energy demand for humidification can be done either by means of the software program EUROVENT-AHU (see Annex C) or manually according to method below.

Accuracies: Manual method – 5 % in relation to software program.

Note that in case of moisture recovery the accuracies can not be specified.

8.6.4 MANUAL CALCULATION PROCEDURE FOR ANNUAL HUMIDIFICATION ENERGY DEMAND

To take into account the different frequency distributions of outdoor moisture content during day time and night time it is recommended to calculate the annual energy demand for two periods a day:
¥ day time operation: 6.00 — 18.00
¥ night time operation : 18.00 — 6.00
The daily operation time is split up in periods falling into the specified day time operation and periods falling into the night time operation.
Each air handling system will have a mix of daytime and night time operation, depending on the type and utilization of the building. Therefore it is necessary to calculate separately the humidification energy demands for daytime operation ($Q_{hum,d}$) and night-time operation ($Q_{hum,n}$) and then to sum those figures, taking into account the corresponding operation times of the system.

<u>*Annual demand for thermal energy for humidification during daytime*</u>
To calculate the energy consumption on a yearly basis the following correlation factors for humidification during the day time period (6.00 — 18.00) are introduced:

$$¥ K_{hum\ sa,d} = \sum_{i=1}^{i=4380} x_{sa,i} \tag{8.59}$$

$$¥ K_{hum\ oa,d} = \sum_{i=1}^{i=4380} x_{oa,i} \tag{8.60}$$

$$¥ K_{hum\ ea,d} = \sum_{i=1}^{i=4380} x_{ea,i} \tag{8.61}$$

where

$K_{hum,sa,d}$	=	correlation factor for humidification, supply air, daytime in kg/kg·h/a
$K_{hum,oa,d}$	=	correlation factor for humidification, outdoor air, daytime in kg/kg·h/a
$K_{hum,ea,d}$	=	correlation factor for humidification, difference between extract air and outdoor air, daytime in kg/kg·h/a
$x_{sa,i}$	=	moisture content of supply air at the hour i in ¡C
$x_{oa,i}$	=	moisture content of outdoor air at the hour i in ¡C
$x_{ea,i}$	=	moisture content of extract air at the hour i in ¡C

The applied calculation rules to establish these cumulated figures are given in Annex B.
With these correlation factors, the annual demand of thermal energy for humidification during day time operation can be calculated as:

$$Q_{hum,d} = \eta_{VAV} \cdot q_v \cdot \rho \cdot 2500 \cdot \left[\left(K_{hum\ sa,d} - K_{hum\ oa,d} \right) - \frac{\eta_{x,s}}{100} \left(K_{hum\ ea,d} - K_{hum\ oa,d} \right) \right] \frac{t_{op,d}}{4380} \tag{8.62}$$

where

$Q_{hum,d}$	=	annual thermal energy consumption for humidification during daytime in kWh/a
η_{VAV}	=	reduction factor for variable air volume system (VAV)
q_v	=	air flow rate in m^3/s
ρ	=	density of the air, established at 1,2 kg/m^3
2500	=	evaporation heat of water in kJ/kg
$K_{hum\ sa,d}$	=	correlation factor supply air during day time in kg/kg·h/a (see Annex A)

$K_{hum\ oa,d}$ = correlation factor outdoor air during day time in kg/kg·h/a (see Annex A)

$K_{hum\ ea,d}$ = correlation factor extract air during day time in kg/kg·h/a (see Annex A)

$\eta_{x,s}$ = seasonal moisture recovery efficiency in %

$t_{op,d}$ = annual operation time of the air handling system during day time in h/a

4380 = day time hours per year for continuous operation in h/a

Annual demand for thermal energy for humidification during night time

To calculate the energy consumption on a yearly basis the following correlation factors for humidification during the night time period (18.00 — 6.00) are introduced:

$$K_{hum\ sa,n} = \sum_{i=1}^{i=4380} x_{sa,i} \tag{8.63}$$

$$K_{hum\ oa,n} = \sum_{i=1}^{i=4380} x_{oa,i} \tag{8.64}$$

$$K_{hum\ ea,n} = \sum_{i=1}^{i=4380} x_{ea,i} \tag{8.65}$$

where

$K_{hum,sa,n}$ = correlation factor for humidification, supply air, night-time in kg/kg·h/a

$K_{hum,oa,n}$ = correlation factor for humidification, outdoor air, night-time in kg/kg·h/a

$K_{hum,ea,n}$ = correlation factor for humidification, difference between extract air and outdoor air, night-time in kg/kg·h/a

$x_{sa,i}$ = moisture content of supply air at the hour i in ¡C

$x_{oa,i}$ = moisture content of outdoor air at the hour i in ¡C

$x_{ea,i}$ = moisture content of extract air at the hour i in ¡C

The applied calculation rules to establish these cumulated figures are given in Annex B.
With these correlation factors, the annual demand of thermal energy for humidification during night time operation can be calculated as:

$$Q_{hum,n} = \eta_{VAV} \cdot q_V \cdot \rho \cdot 2500 \cdot \left[\left(K_{hum\,sa,n} - K_{hum\,oa,n} \right) - \frac{\eta_{x,s}}{100} \left(K_{hum\ ea,n} - K_{hum\ oa,n} \right) \right] \frac{t_{op,n}}{4380} \tag{8.66}$$

where

$Q_{hum,n}$ = annual thermal energy consumption for humidification during night-time in kWh/a

η_{VAV} = reduction factor for variable air volume system (VAV)

q_v = air flow rate in m^3/s

ρ = density of the air, established at 1,2 kg/m^3

2500 = evaporation heat of water in kJ/kg

$K_{hum\ sa,n}$ = correlation factor supply air during night-time in kg/kg·h/a (see Annex A)

$K_{hum\ oa,n}$ = correlation factor outdoor air during night-time in kg/kg·h/a (see Annex A)

$K_{hum\ ea,n}$ = correlation factor extract air during night-time in kg/kg·h/a (see Annex A)

$\eta_{x,s}$ = seasonal moisture recovery efficiency in %

$t_{op,n}$ = annual operation time of the air handling system during night-time in h/a
4380 = night-time hours per year for continuous operation in h/a

Total annual demand for thermal energy for humidification
The total annual demand for humidification is calculated by adding the annual demand during daytime and during night time:

$$Q_{hum} = Q_{hum,d} + Q_{hum,n} \quad (8.67)$$

where

Q_{hum} = total annual demand for thermal energy for humidification in kWh/a

$Q_{hum,d}$ = annual thermal energy consumption for humidification during daytime in kWh/a

$Q_{hum,n}$ = annual thermal energy consumption for humidification during night-time in kWh/a

8.7 CALCULATION OF THE ANNUAL CONSUMPTION OF ELECTRICAL ENERGY

The devices, which consume a considerable amount of electrical energy in AHU are:
¥ electrical motors for the fans including speed control system (P_F)
¥ pumps for heat exchangers and humidifiers (P_p)
¥ motors for rotary heat exchangers (P_{RHE})
¥ water treatment system ($P_{w,treatment}$)

The chapters 4 to 7 provide detailed information on each single energy consumer.

The calculation of annual electrical energy and total costs can be done by means of the program Eurovent-AHU, see Annex C