Psychrometric Chart Structure and Application

SIEMENS

Contents

1. Thermodynamic fundamentals

1.1 General

A thin layer of air surrounds the earth. Variations in this layer cause changes in what is called the barometric (atmospheric) pressure (measured with a barometer). Air's most critical feature is that life forms need it to breathe. A grown person, for example, requires approximately 0.5 m³ of air to breathe per hour to maintain life processes. In addition, the air fulfills other vital requirements. For example, air absorbs vast amounts of water in the form of vapor from the surfaces of lakes and oceans, transports it long distances and then lets it fall to earth in the form of precipitation.

The physical quantities used to describe the state of the air are referred to as variables. Air conditioning systems also deal with these variables. The most important are air temperature, humidity and pressure.

1.2 Pure dry air

Air is a mixture of gases, vapors and contaminants. Dry, clean air exists only theoretically. Dry, clean air would consist of:

Gaseous material:	Chemical symbol:	Volume: %	Weight: %
Nitrogen	N_2	78.060	75.490
Oxygen	O_2	20.960	23.170
Argon	Ar	0.930	1.290
Carbon dioxide	CO_2	0.030	0.040
Hydrogen	H_2	0.010	0.001
Neon	Ne	0.002	0.001
Helium, Krypton, Xenon	He, Kr, Xe	0.008	0.008

1.3 Humid air

Absolutely dry air does not exist in the free atmosphere – it always contains a certain amount of water vapor. Moist air is thus a mixture of dry air and water vapor.

The water vapor portion plays a very important role in heating, ventilation and air conditioning systems. This is true even though the largest possible amount of water vapor in the air – when considering air states of interest – amounts to only a few grams per kilogram (kg) of dry air. Air with a humidity that is too low or too high is uncomfortable. Further, physical characteristics of raw materials or the end product of industrial operations are often strongly dependent on the humidity level of production and storage areas.

In order to create the desired air state in a room, the air must first be prepared. This means that, depending on the situation, it must be cleaned, heated, cooled, humidified or dehumidified. The air state changes required for this can be calculated with the help of the gas laws. This is not especially difficult, yet can be involved and time consuming. Alternately, you can graphically depict the individual variables of the air state with the help of a psychrometric chart. This diagram considerably simplifies calculations involving air state changes.

1.4 Variables

1.4.1 Temperature

Temperature characterizes the sensible, that is, perceptible, heat state of the air. Temperature can be measured with a thermometer. The units of temperature are °C or, absolute, in Kelvin (K). Temperature differentials are always specified in K.

1.4.2 Absolute humidity

Absolute humidity x is the amount of water in grams (g) per kilogram (kg) of air. The absolute humidity x is specified in tables and diagrams in units of g/kg.

1.4.3 Relative humidity

At a given temperature and pressure, air can only hold a certain maximum amount of water vapor. The higher the temperature and air pressure, the higher the maximum possible water content. This maximum water content for any given state of air is referred to as saturation. If you mark the saturation point at every air temperature in the psychrometric chart and connect the points, the result is referred to as the saturation line.

For the example of an air temperature of 20 °C and an air pressure of 1,013 mbar, the saturation point is reached when the water content is 14.6 g/kg air. If this kilogram of air, however, now only holds 7.3 grams of water vapor or 50 % of the maximum amount it can hold, then we say this air has a relative humidity (r.h.) of φ = 50 %.

The relative humidity φ thus specifies, at a specific temperature, how much water vapor is carried by the air relative to the saturation amount. It is calculated as follows:

$$\varphi = \frac{x}{x_S} \cdot 100\,\%$$

φ = relative humidity
x = water vapor in [g/kg]
x_S = water vapor for saturated air in [g/kg]

1.4.4 Density

Density is mass divided by volume, e.g., the amount of mass in kilograms of a material having a volume $V\rho$ of one cubic meter. The unit of density ρ is thus kg/m³.
At 0 °C and sea level, the density ρ is:

- Dry air: ρ = 1.293 [kg/m³]
- Water vapor: ρ = 0.804 [kg/m³]

1.4.5 Specific heat

The specific heat c of a solid, liquid or gaseous material is the amount of heat required to heat up a mass of 1 kg of the material by 1 K. Specific heat is specified in units of J/kg*K or kJ/kg*K (J = Joules; kJ = kiloJoules). Specific heat increases with increasing temperature of the material and for gases also with increasing pressure. As a result, for gases, we distinguish between c_P, the specific heat at constant pressure and c_V, the specific heat at constant volume.

Tables generally specify the values for c_P at 20 °C and 1,013 mbar air pressure. These values are also suitable for calculations in heating, ventilation and air conditioning systems and hold for:

- Dry air: c_P = 1.01 [kJ/kg.K]
- Water vapor: c_P = 1.86 [kJ/kg.K]

1.4.6 Heat content or enthalpy

One of the essential air calculations is the determination of the quantity of heat required to reach an air state in a room defined by a certain temperature and humidity. In this case, the air, whose state is known, must be modified by suitable processes such as mixing, heating, cooling, humidifying or dehumidifying to be converted to the state desired. Most of these processes also result in changes in the heat content (enthalpy) h of the air being processed. In thermodynamics, we refer to the enthalpy of a material having a mass of 1 kg as specific enthalpy h [kJ/kg]. Absolutely dry air having a temperature of $\vartheta = 0$ °C and a theoretical water content of x = 0 g/kg, has an enthalpy defined as h = 0 kJ/kg.

This air state corresponds to the zero point of the enthalpy scale. Values of enthalpy < 0 kJ/kg are specified with a negative sign (–). Differences in enthalpy Δh between the beginning and end state of an air modification, can be depicted graphically on an psychrometric chart with ease. If we multiply the mass [kg] of the air to be processed with the graphically determined enthalpy difference Δh, the result is the required quantity of heat for this state change.

Example:

What is the quantity of heat required to change the state of 1,000 kg of air from $\vartheta_1 = 0$ °C and $x_1 = 3$ g/kg to $\vartheta_2 = 22$ °C and $x_2 = 7$ g/kg? (air pressure = 1,013 mbar)

Solution:

Two quantities of heat are required to heat the 1,000 kg of air from 0 °C to 22 °C (sensible heat) and to vaporize 1,000 · 4 g/kg = 4 kg water (latent heat). You have to calculate these two types of heat separately. The specific heat of the dry air is, on average, for the range in question, 1.01 kJ/kg.K, and for water 4.19 kJ/kg.K.

The vaporization heat ρ for water is both temperature- and pressure-dependent. It can be set to 2,450 kJ/kg for water vapor partial pressures < 0.1 bar and for temperatures < 45 °C. Thus, for:

- Heating the air:
  $Q_L = M_L \cdot c_P \cdot \Delta\vartheta = 1{,}000 \cdot 1.01 \cdot 22 = 22{,}220$ [kJ]
- Heating the water:
  $Q_W = M_W \cdot c_W \cdot \Delta\vartheta = 4 \cdot 4.19 \cdot 22 \quad = \quad 370$ [kJ]
- Vaporizing the water:
  $Q_V = M_W \cdot \rho = 4 \cdot 2{,}450 \qquad = \quad 9{,}800$ [kJ]

The calculation of this partial quantity of heat yields the required quantity of heat for the state change
$Q_{1,2} = Q_L + Q_W + Q_V$ = 22,220 + 370 + 9,800 = 32,390 [kJ]

The calculation with the graphically derived Δh from the psychrometric chart is more exact than the method of calculation using approximate temperature-dependent values. This is because each point of the psychrometric chart uses the exact variables associated with that point.

1.4.7 Pressure

Pressure is a force acting over a surface. The weight of the air on the surface of the earth is called atmospheric pressure. At sea level, the average value is 1.013 bar or 760 mm Hg.

The pressure unit in the international system of units (SI units) is:
- 1 Newton/m² = 1 N/m² = 1 Pa (Pascal)

The bar, however, is used for practical purposes in HVAC:
- 1 bar = 1,000 mbar (millibar) = 10^5 N/m² = 10^5 Pa

In earlier times, the pressure for air and thermic calculations was often specified by the height of a column of fluid, e.g. mm of a water column (mm WG) or in meters of a water column (m WG).
- 1 bar = 10,130 mm WG or 10.13 m WG
- 1 mm WG = 10 Pa or 1 m WG = 10 kPa

1.4.8 Material flow

Material flow involves:
- Volumetric flow V in [m³/s] or [m³/h]
- Mass flow m in [kg/s] or [kg/h]

A homogeneous liquid or gaseous medium having a volume m or a mass in kg, flowing uniformly through a flow cross section in one second, is referred to as a volumetric flow or a mass flow.
The term flow rate is also frequently used. When specifying flow rates, both SI units are used (m³/s or kg/s).

1.4.9 Conversion tables
Energy, work and quantity of heat

Unit:	J	kJ	kWh	kcal
1 J = 1 Nm = 1 Ws	1	0.001	$\approx 0.28 \cdot 10^{-6}$	$\approx 0.24 \cdot 10^{-3}$
1 kJ = 1 kWs	1,000	1	$\approx 0.28 \cdot 10^{-3}$	≈ 0.24
1 kWh	$3,600 \cdot 10^3$	3,600	1	860
1 kcal	4,186	≈ 4.19	$\approx 1.16 \cdot 10^{-3}$	1

Power and heat flow (thermal output)

Unit:	W	kW	kJ/h	kcal/h
1 W = 1 J/s	1	0.001	0.001	0.860
1 kW	1,000	1	$\approx 0.28 \cdot 10^{-3}$	860
1 kJ/h	≈ 0.28	$\approx 0.28 \cdot 10^{-3}$	1	≈ 0.24
1 kcal/h	1,163	$1.163 \cdot 10^{-3}$	≈ 4.19	1

Pressure

Unit:	Pa	bar	mm WS	mm Hg (Torr)
1 Pa = 1 N/m²	1	10^{-5}	≈ 0.1	$\approx 7.5 \cdot 10^{-3}$
1 bar	10^5	1	$1.02 \cdot 10^4$	≈ 750
1 mm WG	9.81	$\approx 10^{-4}$	1	$7.36 \cdot 10^{-2}$
1 mm Hg = 1 Torr	133.3	$\approx 1.3 \cdot 10^{-3}$	13.6	1

2. Structure of the psychrometric chart

2.1 General Early on, attempts were made to simplify calculations dealing with air state changes by using graphs. There are different types of diagrams used for psychrometric calculations. The Mollier diagram is commonly used in Europe, while the Carrier diagram is used in the U.S. Both have the same principle format. Only the axes directions are different. Temperature is on the y (vertical) axis in the Mollier diagram, while water content is on the x (horizontal) axis. The reverse is true for the Carrier diagram.

Psychrometric charts are either h-x or t-x diagrams, where t is the temperature in °C, h is the enthalpy in kJ/kg and x is the absolute water content of the air in g/kg. Using the "psychrometric chart for humid air", you can graphically represent and easily calculate air states and state changes associated with air conditioning systems. The following parameters are meaningful when describing and calculating state changes of humid air:

- ϑ = air temperature [°C]
- φ = relative humidity [% r.h.]
- x = water content [g/kg*]
- h = enthalpy [kJ/kg*]

(* relative to 1 kg dry air)

The characteristics and behaviour of humid air depend on the barometric pressure. Thus, any given psychrometric chart can only be drawn for a single specific barometric pressure. The chart used in this chapter is based on a barometric pressure of 980 mbar or 735 Torr.
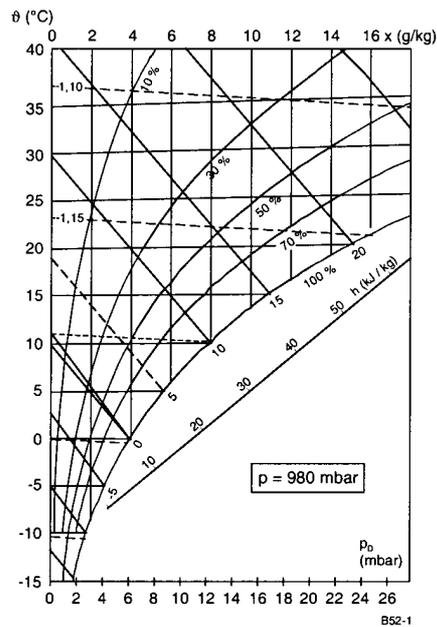

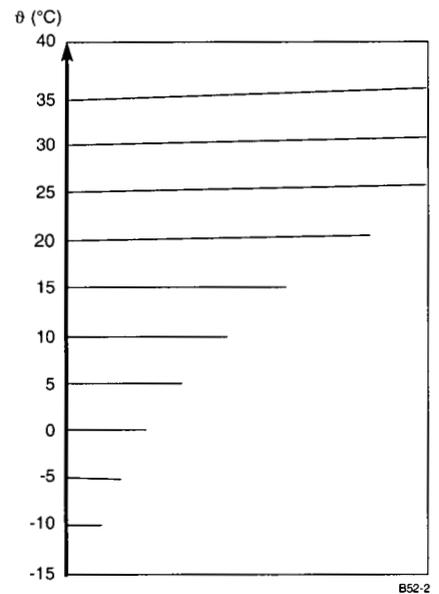Fig. 2-1   Psychrometric chart according to Mollier

Fig. 2-2   Temperature scale including iso-therms

2.2 The structure of the Mollier diagram

2.2.1 Temperature scale The temperature scale serves as the standard variable for the psychrometric Mollier diagram. It is on the vertical axis over the range of interest. For air conditioning systems, the range is approximately −15 °C to +40 °C.

The horizontal parameter lines extending from left to right are isotherms, that is, lines of constant air temperature. The isotherms for 0 °C run parallel to the horizontal axis; the isotherms at higher temperatures have increasingly larger slopes.

2.2.2 Absolute humidity scale The second important variable, the water content x (or absolute humidity of the air), is on the horizontal axis. Vertically running parameter lines are designated as lines of constant water content. If water content x is known in addition to temperature t, then the state point of the air can be unambiguously determined in the psychrometric chart. Because of this, the psychrometric chart is often also referred to as the t-x diagram. The units for absolute humidity or water content x are: grams of water per kilogram dry air [g/kg].
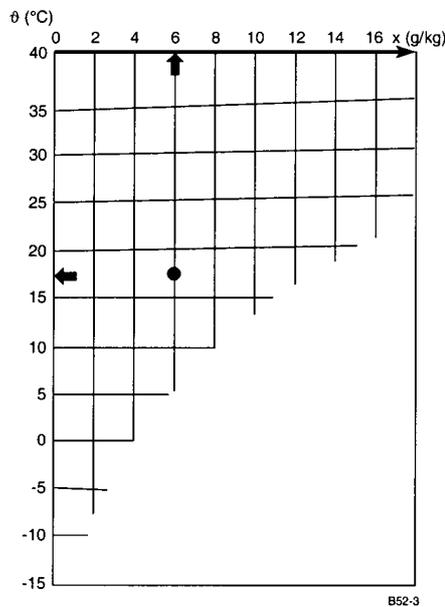
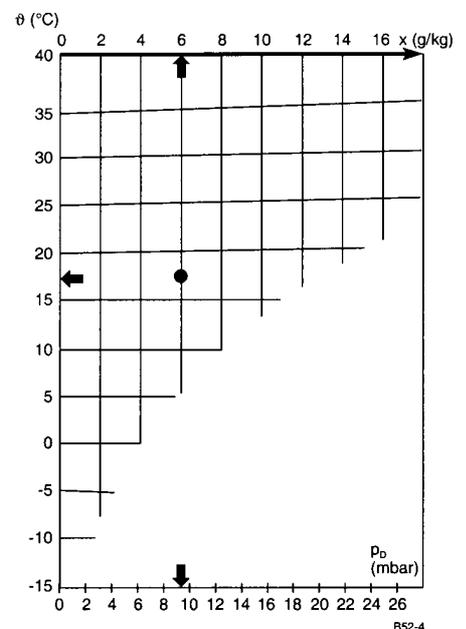Fig. 2-3   Water content x and air temperature   Fig. 2-4   Vapor pressures p_D

2.2.3 Vapor pressure scale Overheated water vapor, gaseous and mixed with air, has a certain vapor pressure p_D. This vapor pressure is part of the total air pressure and is thus referred to as the partial pressure of the water vapor. This partial pressure is dependent on the ratio of water vapor to dry air in the mixture. The larger the portion of water vapor, the larger its partial pressure p_D. We can thus represent the partial vapor pressure p_D in mbar as horizontals parallel to the water content x. It then becomes easy to determine from the diagram which partial pressure p_D corresponds to a specified water content x in g/kg.
(Example x = 6 g/kg ➢ p_D ≈ 9.5 mbar).

2.2.4 Saturation pressure, saturation line

Water vapor partial pressure in air, i.e., water vapor concentration, can be increased until the saturation pressure p_S is reached. At this point, the air can no longer accept additional water vapor. Attempting to increase the partial pressure above the saturation pressure by adding more water vapor causes condensation and fog or fine droplets become visible.

The saturation pressure p_S is dependent on air temperature and pressure. Since the air pressure can be considered constant for each psychrometric chart, we can ignore it. Thus, we can either calculate the saturation pressure p_S for each temperature between 20 °C and 100 °C, or take the values from tables and enter them in the psychrometric chart.

Connecting saturation pressures at various temperatures with one another in a psychrometric chart creates a saturation line. The air is 100 % saturated along these limit lines; it cannot accept any more water vapor without condensation taking place. As an example (Fig. 2-5), air at 20 °C (at 980 mbar) can hold a maximum of 15 g/kg water vapor.
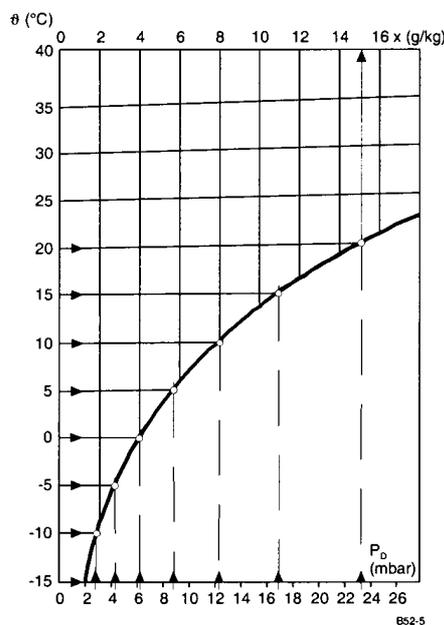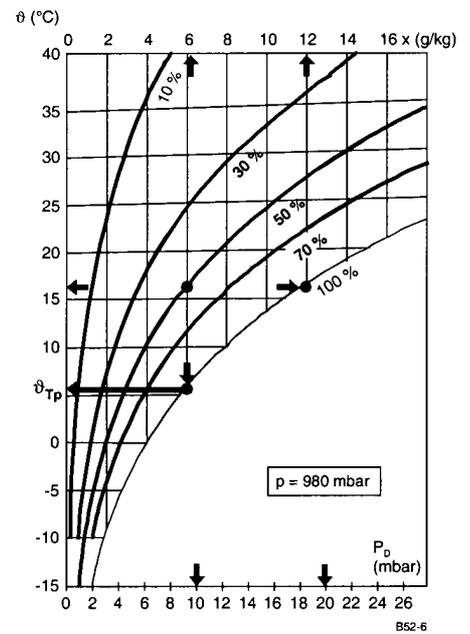

Fig. 2-5   Saturation pressure p_S

Fig. 2-6   Dew point temperature

2.2.5 Dew point temperature (saturation temperature)

Inspection of the diagram (Fig. 2-6) shows that the saturation line cannot be reached simply by increasing the water vapor content x. If we cool air for example, having an absolute water content of x = 6 g/kg, from 20 °C to +5 °C, the saturation line is reached at a temperature of approximately +6 °C. Further cooling to 5 °C leads to condensation. We thus refer to the intersection of a vertical x line with the saturation line as the dew point. The associated temperature is referred to as the dew point or saturation temperature.

Water vapor condenses on surfaces and bodies, whose temperature is below the dew point, that is, droplets form. If you wish to dehumidify (remove water from) a water vapor-air mixture, then you have to cool it sufficiently to exceed the dew point temperature. The lower you go below the dew point temperature, the greater the dehumidifying effect.

2.2.6 Lines having a constant relative humidity

The air is 100 % saturated with water vapor along the saturation line (dew point line), that is, the relative humidity is 100 % there. If, for example, the air holds only half this amount of water vapor, that is 50 % of the saturated water vapor quantity, then we refer to the level of saturation as φ = 50 % relative humidity (r.h.). If we now place a point in the psychrometric chart at every temperature where the saturated water vapor quantity is 50 %, and then connect all the points, the result is a line having a constant relative humidity (r.h.) φ of 50 %.

Example

ϑ = 17.5 °C	x = 12.4 g/kg	φ = 100 % r.h.
ϑ = 17.5 °C	x = 6.2 g/kg	φ = 50 % r.h.
ϑ = 7.5 °C	x = 6.4 g/kg	φ = 100 % r.h.
ϑ = 7.5 °C	x = 3.2 g/kg	φ = 50 % r.h.

In the same manner, by adding and connecting the points having 90 % or 70 % of the saturated water-vapor amount, we can construct a line having a constant relative humidity of φ = 90 % or 70 % r.h. This is the method of constructing all lines having a constant relative humidity between 5 and 100 % r.h.

Instead of the relationship between water content and saturated water vapor quantity, we can use the relationship between partial pressure and saturation pressure of the water vapor to construct lines having constant relative humidity.
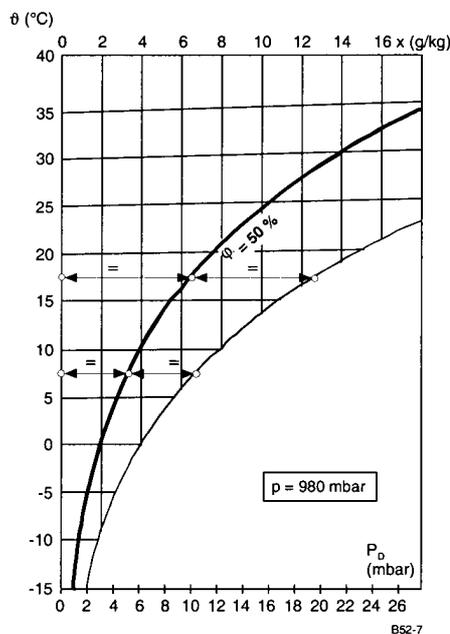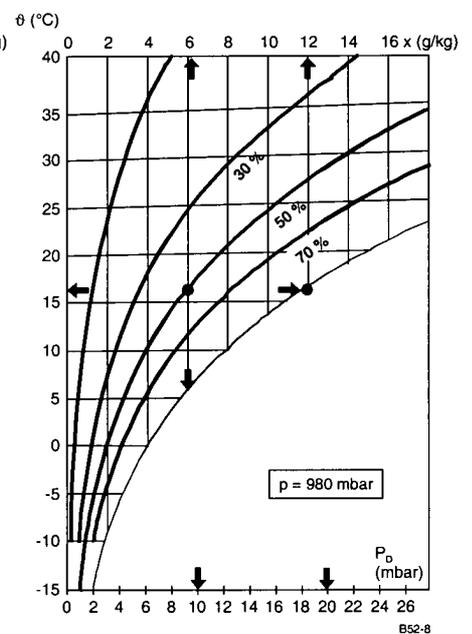

Fig. 2-7   Relative humidity φ

Fig. 2-8   Determining seven variables for a given point

2.2.7 Temperature, humidity, pressure

Using the information concerning the structure of the psychrometric chart covered in section 2.2.6, you can now define the following seven variables for a given point:

- Temperature ϑ = 17 °C
- Dew point temperature ϑ_S = 7 °C
- Absolute humidity   x = 6.2 g/kg
- Water vapor content at saturation x_S = 12.4 g/kg
- Relative humidity φ = 50 % r.h.
- Partial pressure p_D = 9.5 mbar
- Saturation pressure p_S = 19.0 mbar

2.2.8 Lines having constant enthalpy

The enthalpy h (heat content) of humid air has two components: dry air enthalpy and water vapor enthalpy.
The specific enthalpy of water vapor is considerably larger than that of dry air. Thus, water vapor holds a large portion of the enthalpy of humid air. Absolutely dry air having a temperature of $\vartheta = 0\ °C$ and a theoretical water content of $x = 0\ g/kg$ has an enthalpy defined as $h = 0\ kJ/kg$, i.e. this air state has been selected as the zero point of the enthalpy scale. Starting from this point, the enthalpy of each point of the diagram can be calculated. We do this by adding the energy required to heat the air to that required to heat the water.

If water is sprayed into air, or if you bring air into contact with wet surfaces, water vaporizes and in doing so, withdraws the heat of vaporization exclusively from the resulting mixture. Because **this procedure exchanges essentially no heat with the surroundings,** we consider the enthalpy of the air-water mixture to remain unchanged. We then say that this state change took place at constant enthalpy.

There is, however, a displacement between the decreasing sensible (perceptible) and increasing latent portion of the air enthalpy. This displacement causes the mixture to cool. The slope of the lines of constant enthalpy (isenthalpics or adiabatics) on the diagram is the ratio of the sensible to the latent heat.

Provided that the various specific heats of dry and humid air are taken into account in the construction of the isotherms (lines of constant temperature), the **isenthalpic** lines (lines of constant enthalpy) will be **parallel.** The **enthalpy scale** is shown below the saturation lines in the psychrometric chart and runs at right angles to the isentropics (Fig. 2-9).

You can now read off from this scale the enthalpy of the air state defined in paragraph 2.2.7 ($\vartheta = 17\ °C$ / $x = 6.2\ g/kg$): ➢ **h ≈ 33 kJ/kg.**
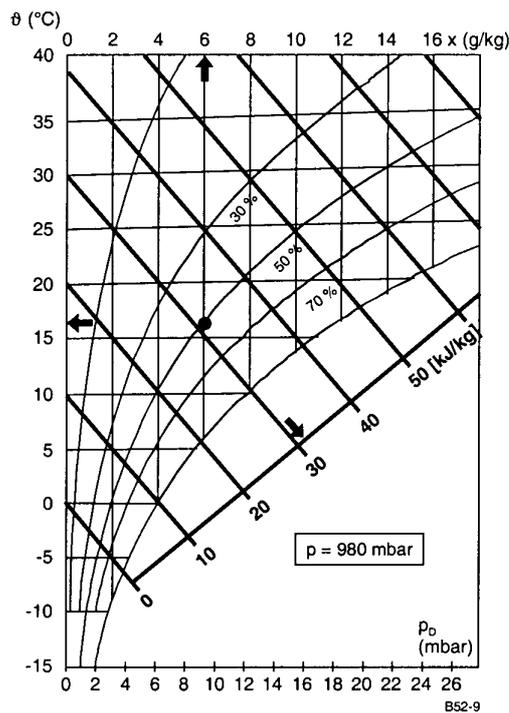

Fig. 2-9   Enthalpy in a psychrometric chart

2.2.9 Wet bulb temperature

Another term used in the thermodynamics of humid air is the so-called wet bulb temperature.

When vaporizing water, the humidity of air can increase until saturation occurs. If the temperature of the water being vaporized has the same temperature as the air, then only latent heat is required for the vaporization. This is taken from the sensible heat, a process referred to as vaporization cooling. In this situation, there is also a state change at constant enthalpy until the saturation pressure (intersection with the saturation line) is reached.

The temperature of this intersection of the isentropics with the saturation line is referred to in air conditioning systems as the wet-bulb temperature or cooling limit.

If we now wish to add lines of constant wet bulb temperature (adiabatics) to the psychrometric chart, we notice that, logically, they must have the same slope as the isentropics. When making exact calculations, we must be careful when calculating the enthalpy, that the starting point is the enthalpy of the water component at 0 °C. For adiabatics, on the other hand, we assume that the water temperature at the start of the state change is the same as the air temperature. This causes a slight change in the slope of the adiabatics relative to the isentropics.
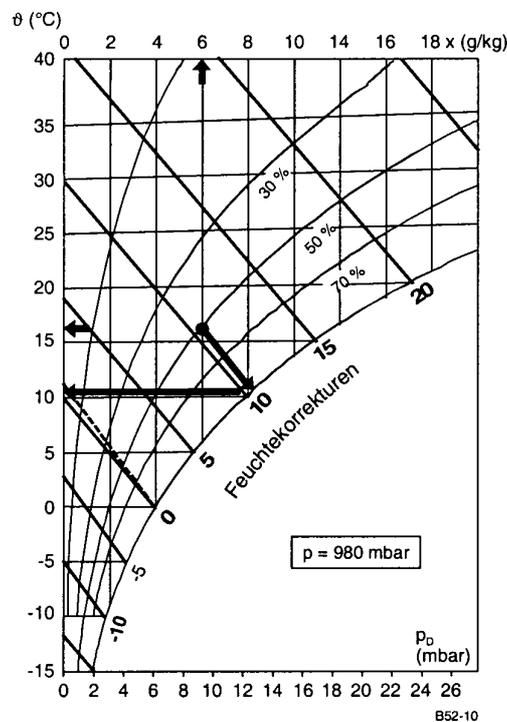

Fig. 2-10 Wet bulb temperature

Wet bulb temperature is measured using a wet-and-dry bulb hydrometer.

The wet-and-dry bulb hydrometer has two thermometers. The sensor of one thermometer is located inside an absorbent cloth which is thoroughly saturated with distilled water before each measurement. During the measurement, the "wet" thermometer has the air being measured intensively moved through it in order to ensure an effective vaporization process. This is accomplished either by using a small built-in fan (aspiration wet-and-dry bulb hydrometer) or by manually rotating the thermometer about the axis of a handle (sling wet-and-dry bulb hydrometer).

The measurement must last long enough (approx. 1 to 2 minutes) to ensure that the wetted sensor has reached the wet bulb temperature. You can measure practically all air states using a wet-and-dry bulb hydrometer and define them using a psychrometric chart.

If you wish to determine the wet bulb temperature for an arbitrary air state point in the psychrometric chart, draw a line from the state point parallel to the isentropics to the saturation line. The temperature of the intersection point of this line with the saturation line is the wet bulb temperature of this air state point (Fig. 2-10).

For the air state defined in section 2.2.7 (ϑ = 17 °C / x = 6.2 g/kg), the result is a wet bulb temperature of approx. 11 °C. When the air is saturated, the wet bulb temperature is the same as the dry air temperature.

2.2.10 Density The density in kg/m³ of humid air is dependent on three different criteria:

1. On air pressure: The psychrometric chart is always drawn for a certain barometric pressure. You must thus take care when making air conditioning calculations that you use a diagram drawn up for the appropriate elevation above sea level. If none is available, then you have to make the relevant conversions (see chapter 4 "Calculation of the altitude correction" for the necessary calculations).
2. On temperature: The higher the temperature of the air, the more it expands and the lower its density.
3. On water vapor content: Water vapor is specifically lighter than air. Thus, the density of the mixture drops with increasing water vapor content. The lines of constant density must thus slope to the right.

The density can now be determined by interpolation for the air state defined in section 2.2.7 of (ϑ = 17 °C / x = 6.2 g/kg): The state point lies approximately in the middle between ρ = 1.15 and 1.20 kg/m³,
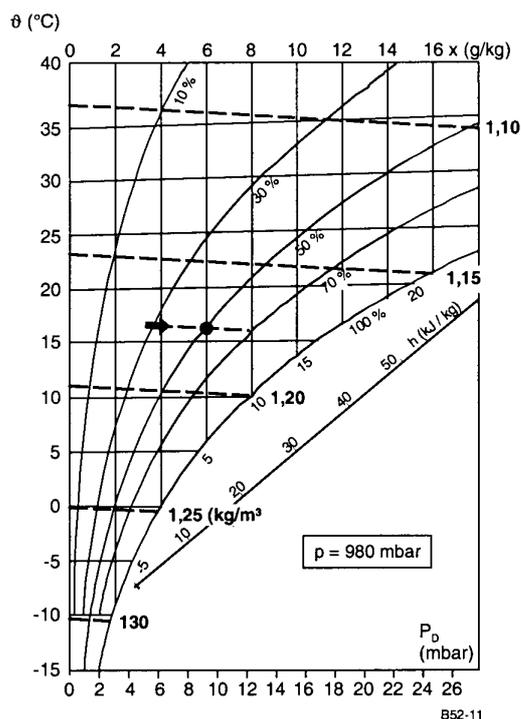➤ average ρ = 1.175 kg/m³ (Fig. 2-11).

Fig. 2-11   Air density

This closes the "Structure of the psychrometric chart" chapter. Using the psychrometric chart, you can define the air state using ten variables:

	Air state	Symbol	Value	Unit
1	Wet bulb temperature	ϑ_L	17	°C
2	Absolute humidity	x	6.2	g/kg
3	Partial vapor pressure	p_D	1.0	kPa
4	Saturation pressure	p_s	2.0	kPa
5	Saturation temperature (dew point temperature)	ϑ_s	7	°C
6	Relative humidity	φ	50	% r.h.
7	Enthalpy	h	33.0	kJ/kg
8	Wet bulb temperature	ϑ_F	11.5	°C
9	Density	ρ	1.175	kg/m³
10	Absolute humidity at saturation pressure	x	12.4	g/kg

These ten variables are shown in the following psychrometric chart (Fig. 2-12):
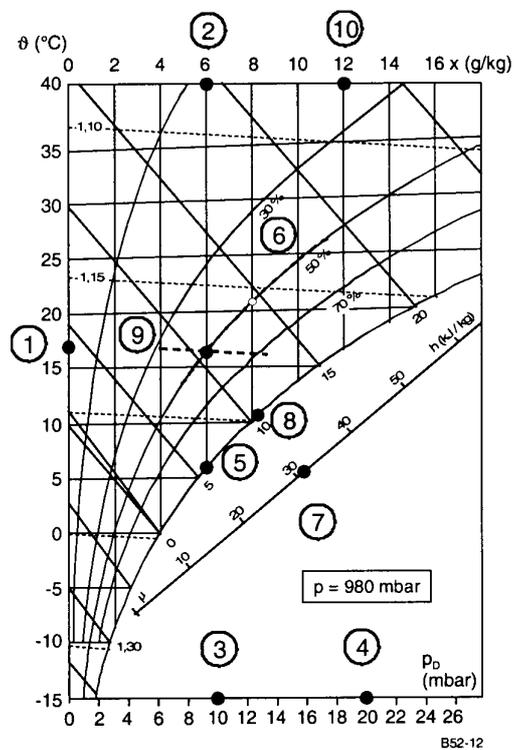

Fig. 2-12   Defining the air state using ten variables

3. Application of the psychrometric chart

3.1 General The following state changes are involved when conditioning air in heating, ventilation and air conditioning systems:

- Mixing two volumes of air having different states
- Heating air
- Cooling air
- Humidifying air
- Drying air (dehumidifying)

The application and calculation examples in this chapter are based on a Mollier psychrometric chart having sea level pressure conditions of 1,013 mbar (= 760 mm Hg).

3.2 Mixing two quantities of air If two air flows, $\dot{m}_1$ and $\dot{m}_2$ [kg/h], having the different states Z_1. and Z_2. are mixed, a third state Z_M results whose variables can be determined from a psychrometric chart. The mixing point divides the line connecting Z_1 and Z_2 into two sections L_1 and L_2, which are inversely proportional to the two original states (L = length in mm). The mixing point always lies closer to the air state of the larger of the two mixed quantities. Two equations describe the relationship:

$\dot{m}_1 / \dot{m}_2 = L_2 / L_1$ or
$\dot{m}_1 / \dot{m}_2 = (L_{1,2} - L_1) / L_1$

Substituting and solving for L_1:

$$L_1 = \frac{L_{1,2} \cdot \dot{m}_2}{\dot{m}_1 + \dot{m}_2}$$

In Fig. 3-1, in the mixing chamber of a ventilation system, $\dot{m}_1 = 1500$ kg/h outside air at state Z_1 ($\vartheta_1 = 5$ °C and $\varphi_1 = 50$ % r.h.), where r.h. = relative humidity, is mixed with $\dot{m}_2 = 2500$ kg/h circulating air of state Z_2 ($\vartheta_2 = 20$ °C and $\varphi_2 = 60$ % r.h.). In order to be able to calculate the mixing point Z_M, points Z_1 and Z_2 are marked on the diagram and connected with a line. In this example, the distance between the two points $L_{1,2}$ (depending on the format of the psychrometric chart) is measured to be 46 mm for example.

Consequently, the length of L_1 is:

$$L_1 = \frac{L_{1,2} \cdot \dot{m}_2}{\dot{m}_1 + \dot{m}_2} = \frac{46 \cdot 2,500}{1,500 + 2,500} \approx 28.8 \text{ [mm]}$$
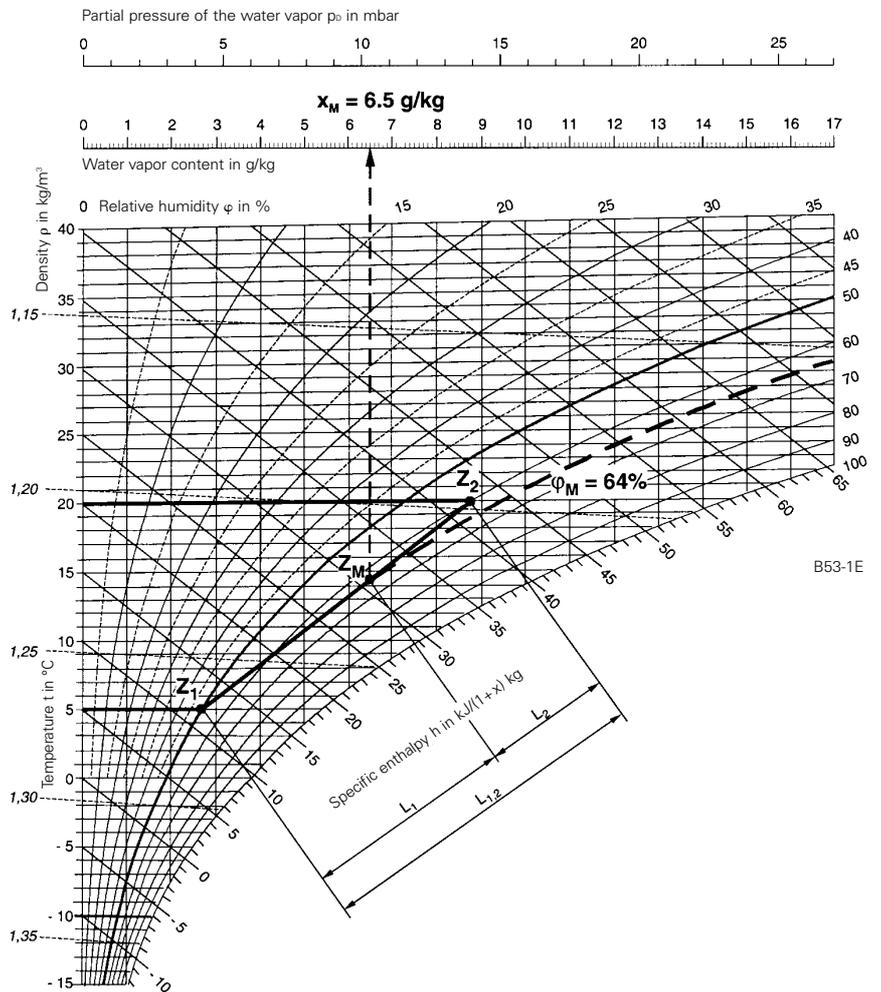
Fig. 3-1   Mixing of air flows Z_1 and Z_2

The length L_1 extended from point Z_1 locates the position of mixing point Z_M with $\vartheta_M = 14.4\ °C$, $X_M = 6.5\ g/kg$, $\varphi_M = 64\ \%$ r.h.

The state of the mixing point can be calculated, e.g., the mixing temperature:

$$\vartheta_M = \frac{\vartheta_1 \cdot \dot{m}_1 + \vartheta_2 \cdot \dot{m}_2}{\dot{m}_1 + \dot{m}_2}\ [°C]$$

or the absolute humidity (the vapor content) at the mixing point:

$$x_M = \frac{x_1 \cdot \dot{m}_1 + x_2 \cdot \dot{m}_2}{\dot{m}_1 + \dot{m}_2}\ [g/kg]$$

The mixing process described here takes place in the region of unsaturated air.

If, however, cold outside air of winter is mixed with the warmer and relatively humid circulating air, then the mixing point could fall below the saturation line, that is, in the mist area (occurs, e.g., in inside swimming pools).

The formation of mist in a mixing chamber of an air handling system is generally not problematic. This is because the air is heated by a heater battery before use, which vaporizes the suspended water droplets, shifting the air state to the region of unsaturated air.

The graphical determination of the mixing point in the mist region follows the same method used for Fig. 3-1. Note, however, that the lines of constant temperature (isotherms) bend downward to the right (mist isotherms). They thus run nearly parallel to lines of constant heat content (isentropics or adiabatics, refer to Fig. 3-2).
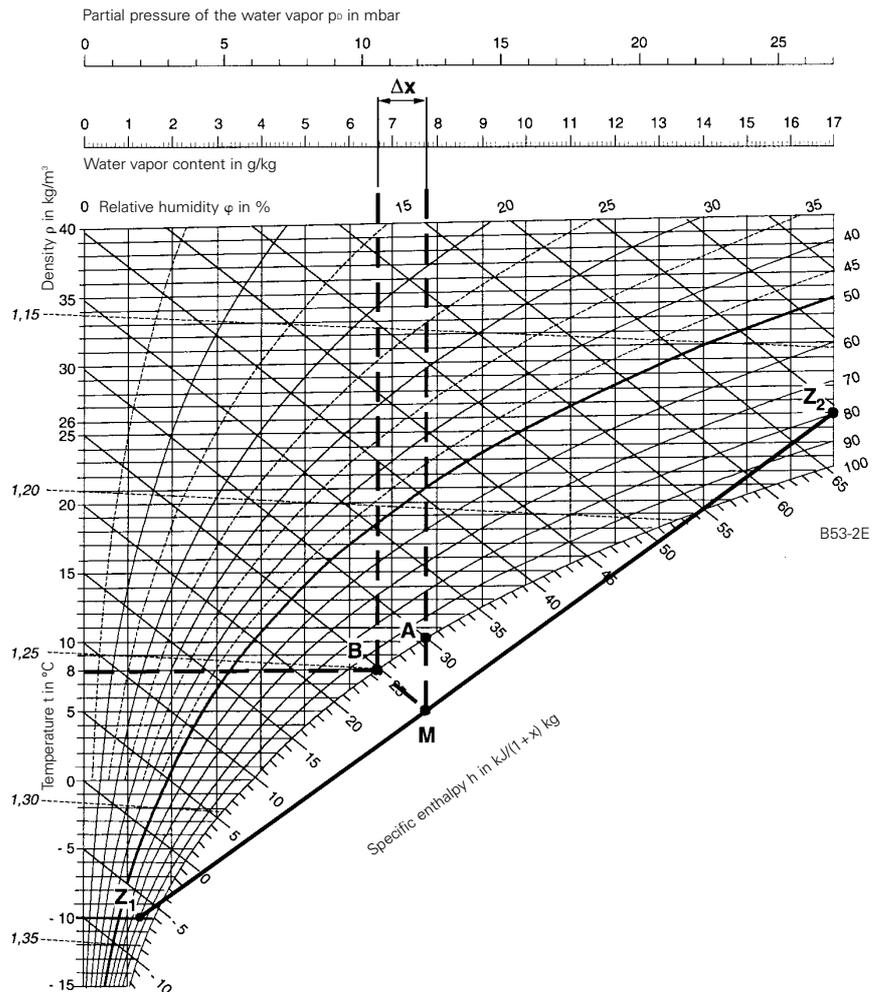

Fig. 3-2   Air mixture with mixing point in the mist region

The air is oversaturated at point M, that is, it contains Δx too much water relative to its temperature. This water is held in the air in the form of fine droplets, that is, as mist in the air. If we heat the air starting from state M, then the temperature of the droplets first increases to the saturation line (point A). If we continue to add heat, then the air state changes to the unsaturated region, that is, no more surplus humidity exists and the mist disappears.

If, however, we leave the temperature of the air unchanged and separate out the surplus water (e.g. by absorption), then the air state moves in the direction of the mist isotherms and reaches the saturation point at B.

3.3 Warming the air

Warming the air is the simplest case of a state change, because in this case neither air nor water is added or removed (x = constant). This process thus follows along a vertical line on the psychrometric chart. The relative humidity, however, is reduced. In order to reach the desired air temperature, heat in the amount of Δh [kJ/kg] has to be added to the air, where:

$$\Delta h = h_2 - h_1 \ [kJ/kg]$$

Fig. 3-3 shows a heating process where air, having a mass flow rate of $\dot{m}_L = 1$ kg/s (= 3600 kg/h) at $\vartheta_1 = +5$ °C and $\varphi_1 = 60$ % r.h., is heated to $\vartheta_2 = 25$ °C. The required thermal output of the air heater is $\dot{Q}_{AH} = \dot{m}_A \cdot \Delta h$. This yields:

$$\dot{Q}_{AH} = \dot{m}_A \cdot (h_2 - h_1) = 1 \cdot (33.3 - 13) \left[\frac{kg \ kJ}{s \ kg} \right] = 20.3 \ \frac{kJ}{s} = 20.3 \ [kW]$$

The calculated thermal output of $Q_{AH} = 20.3$ kW can be supplied using an air heater battery driven by e.g., hot water or steam. If hot water is used, we assume, at full load, that the water is cooled by $\Delta \vartheta = 20$ K (e.g., flow temperature $\vartheta_V = 70$ °C, return temperature $\vartheta_R = 50$ °C). The water flow $\dot{m}_W$ can be calculated from:

$$\dot{m}_W = \frac{\dot{Q}_{AH}}{c_W \cdot \Delta \vartheta_W} \left[\frac{kJ \ kg \ K}{s \ kJ \ K} \right] = \frac{20{,}3}{4.19 \cdot 20} = 0.24 \ [kg/s]$$

(c_W is the average specific heat of water and here amounts to 4.19 kJ/kg K)

If condensed saturated steam is used, then we remove the heat of vaporization r (the quantity of heat required to bring 1 kg of water from its vaporization temperature to the vapor state), and possibility also the heat of condensation. If we take into account the heat of vaporization, then the amount of steam required is:

$$\text{Dampfmenge } \dot{m}_S = \frac{\text{Thermal output } \dot{Q} \left[\dfrac{kJ}{s} \right]}{\text{Heat vaporisation } r \left[\dfrac{kJ}{kg} \right]} = \left[\frac{kg}{s} \right]$$

$$\dot{m}_S = \frac{\dot{Q}}{r} = \frac{21.0}{2258} = 0.009 \left[\frac{kg}{s} \right]$$

Partial pressure of the water vapor p_D in mbar

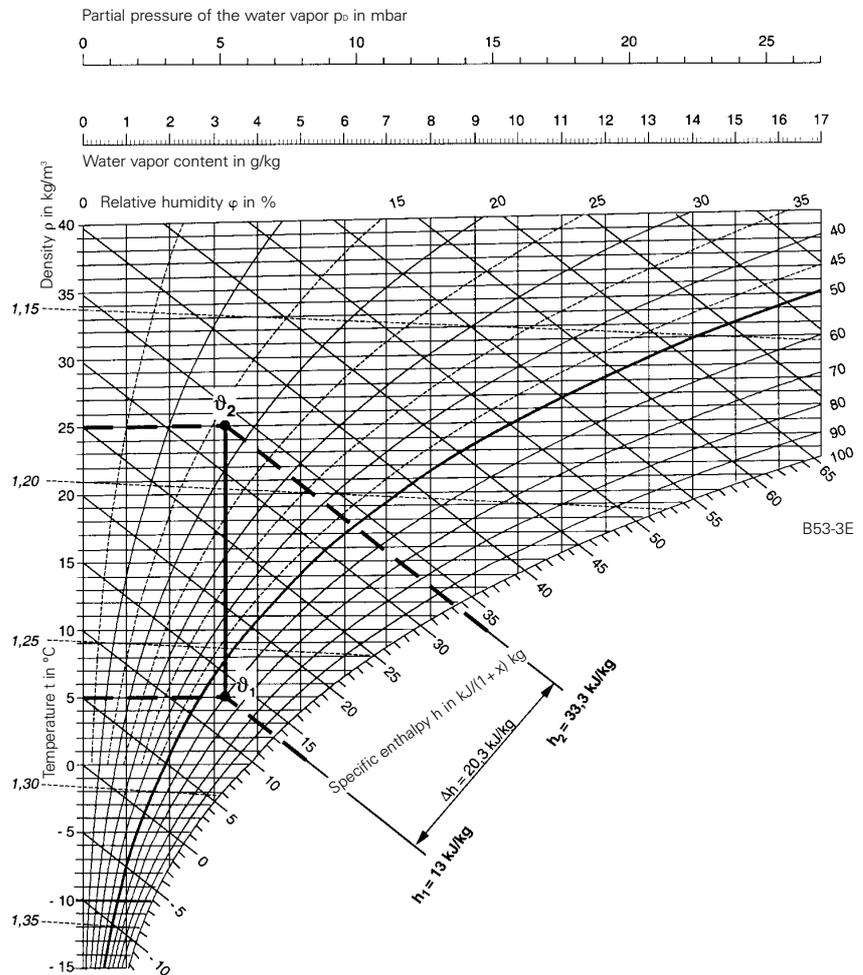Water vapor content in g/kg

Fig. 3-3   Heating the air

3.4 Cooling the air

There are two possibilities for cooling the air:

- Bringing the air in contact with a colder surface (surface cooling)
- Spraying water into the air flow (evaporative cooling)

3.4.1 Surface cooling

There are two possibilities for cooling air using surface cooling:

- The cooling surface temperature lies above the dew point temperature of the air to be cooled (dry cooling surface, $\vartheta_{CO} > \vartheta_{dp}$)
- The cooling surface temperature lies below the dew point temperature of the air to be cooled (wet cooling surface, $\vartheta_{CO} < \vartheta_{dp}$)

Cooling without water condensation

When the temperature of the cooling surface ϑ_{CO} is higher than the dew point temperature ϑ_{TP} of the air to be cooled, no water condenses on the cooling surface. The absolute water vapor content x of the air remains unchanged, but the relative humidity nevertheless increases (Fig. 3-4). The state change in the psychrometric chart thus runs down parallel to the lines of constant water vapor content x. Depending on the cooling output, the air is cooled down to a greater or lesser extent. The temperature of the cooling surface can thus not be completely reached because only a portion of the air comes in direct thermal contact with the cooling ribs (refer to section *"Bypass effect"*).

In order to cool 1 kg air from temperature ϑ_1 to temperature ϑ_2, heat in the amount of Δh must be removed where:

Δh = $h_1 - h_2$ [kJ/kg]

The average cooling surface temperature ϑ_{CO} depends on the construction of the cooler and generally lies approx. 1 to 2 K above the average value between the flow and return water temperature of the cooling water:

$$\vartheta_{CO} = \frac{\vartheta_{FT} + \vartheta_{RT}}{2} + 1\ldots2 \ [K]$$

In Fig. 3-4, the mass $\dot{m}_A = 1$ kg/s (= 3600 kg/h) of air is cooled to $\vartheta_2 = 20$ °C from $\vartheta_1 = 29$ °C and $\varphi_1 = 40$ % r.h. using a surface cooler having an average cooling surface temperature of $\vartheta_{CO} = 18$ °C. The cooling water in turn heats from $\vartheta_{FT} = 15$ °C to $\vartheta_{RT} = 19$ °C. A quantity of heat of:

$\Delta h = h_1 - h_2 = 54.7 - 45.2 = 9.5$ kJ/kg must be removed from one kg of air.

The cooling output needed for this is:

$$\dot{Q}_{CO} = \dot{m}_A \cdot \Delta h = 1 \cdot 9.5 \left[\frac{kg\ kJ}{s\ kg} \right] = 9.5 \ kJ/s = 9.5 \ [kW]$$

The volume of water is calculated in the same manner as for heating air except that there is a much smaller temperature differential $\Delta\vartheta$ available while cooling:

$$\dot{m}_W = \frac{\dot{Q}_{CO}}{c_W \cdot \Delta\vartheta_W} \left[\frac{kJ\ kg\ K}{s\ kJ\ K} \right] = \frac{9.5}{4.19 \cdot (19 - 15)} = 0.57 \ [kg/s]$$
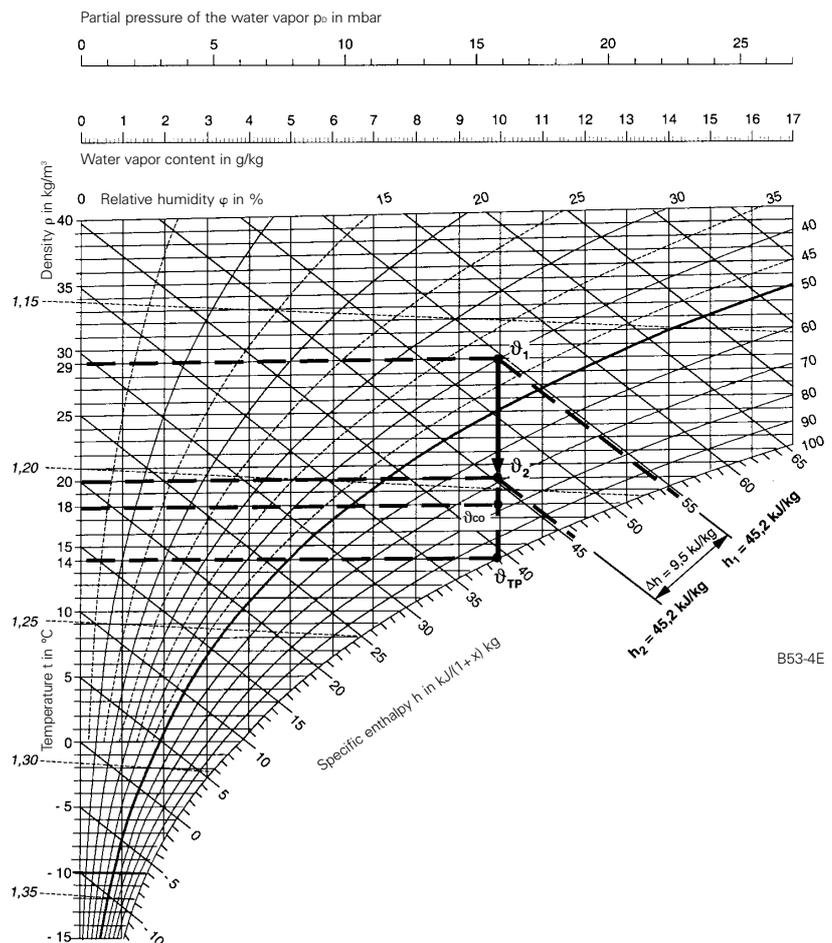

Fig. 3-4 Air cooling using a dry cooling surface

Cooling with water condensation   If the temperature of the cooling surface ϑ_{CO} is under the dew point ϑ_{TP} of the air to be cooled, then part of this air will be cooled so far that water vapor will condense on the cooling surface. The air state after the cooler can thus be imagined as a mixture of three components: cooled dehumidified air, only cooled air and essentially uncooled air (refer to sectio *"Bypass effect"*).

The state change in the psychrometric chart is represented by a straight line for simple calculations which proceed from the starting state of the air ϑ_1 to the intersection of the average cooling surface temperature ϑ_{CO} with the saturation line. Depending on the cooling output, an air state ϑ_2 can occur. This state lies between the starting temperature of the air ϑ_1, and the average cooling surface temperature ϑ_{CO}. The cooling surface temperature cannot be completely reached, however (because of the bypass effect). Even for small cooling outputs, the air is not only cooled, but water condenses out also. The absolute humidity then decreases, whereby the relative humidity increases.

In Fig. 3-5, the mass $\dot{m}_A = 1$ kg/s (= 3600 kg/h) air is cooled to $\vartheta_2 = 20$ °C from $\vartheta_1 = 29$ °C and $\varphi_1 = 40$ % r.h., using a surface cooler having an average cooling surface temperature of $\vartheta_{CO} = 10$ °C. At the same time, the cooling water is heated from $\vartheta_{FT} = 6$ °C to $\vartheta_{RT} = 12$ °C. The quantity of heat removed from one kg air must thus be:

$$\Delta h = h_1 - h_2 = 54.7 - 42.2 = 12.5 \ [\text{kJ/kg}].$$

The cooling output required for this is:

$$\dot{Q}_{CO} = \dot{m}_A \cdot \Delta h = 1 \cdot 12.5 \left[\frac{\text{kg kJ}}{\text{s kg}} \right] = 12.5 \ \text{kJ/s} = 12.5 \ [\text{kW}]$$

The volume of water is calculated in the same manner as for heating the air.

The available temperature differential $\Delta\vartheta$ is, however, much smaller.

$$\dot{m}_W = \frac{\dot{Q}_{CO}}{c_W \cdot \Delta\vartheta_W} \left[\frac{\text{kJ kg K}}{\text{s kJ K}} \right] = \frac{12.5}{4.19 \cdot (12 - 6)} = 0.5 \ [\text{kg/s}]$$
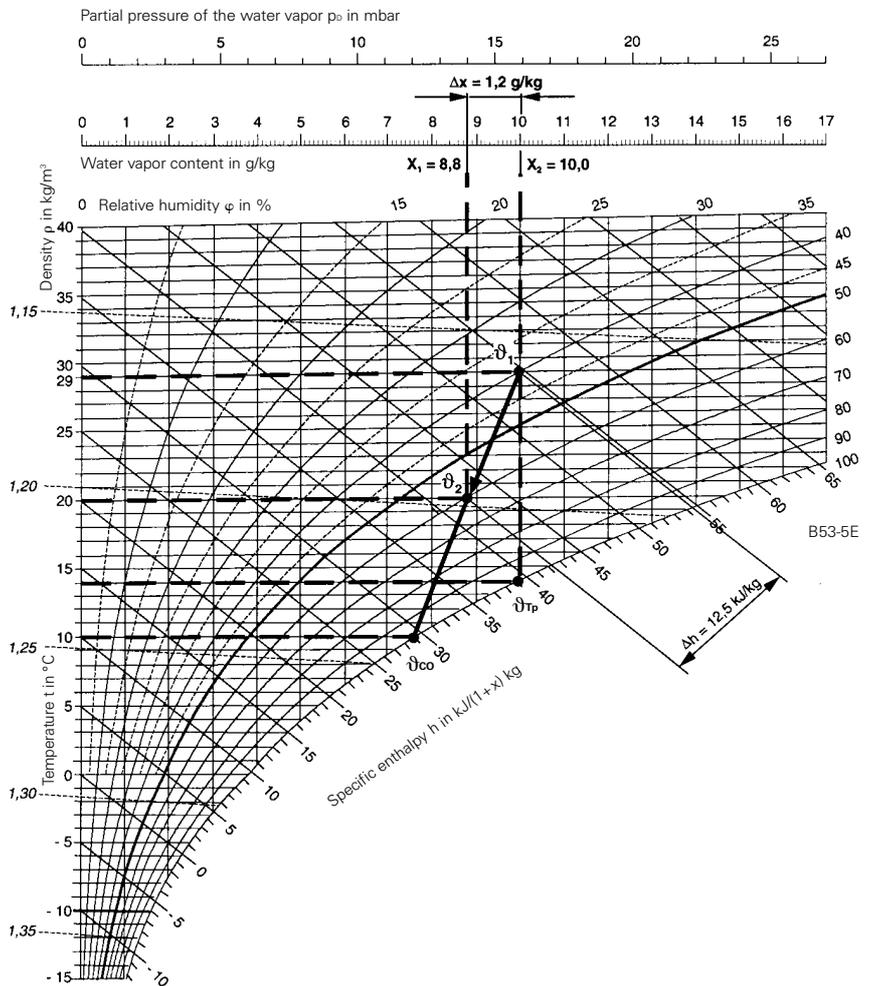
Fig. 3-5   Cooling with water condensation

The volume of condensed water per kg of air is:

$$\Delta x = x_1 - x_2 = 10.0 - 8.8 = 1.2 \; [g/kg]$$

As the examples (Fig. 3-4 and 3-5) indicate, more cooling energy is needed for the same temperature change, when cooling with water condensation than for dry cooling. With additional cooling output, the heat of vaporization of the extracted water is conducted away, thus:

$$\Delta \dot{Q}_{CO} = r \cdot \Delta x = 2{,}450 \cdot 0.0012 \left[\frac{kg \; kJ}{kg \; kg}\right] \approx 3 \left[\frac{kJ}{kg}\right]$$

When cooling using water condensation, the state change does not run along a straight line as shown in Fig. 3-5. It rather runs along a more or less curved line from ϑ_1 to ϑ_{KO} (Fig. 3-6). The bend in the curve is, among other things, co-determined by the hydraulic circuit employed.
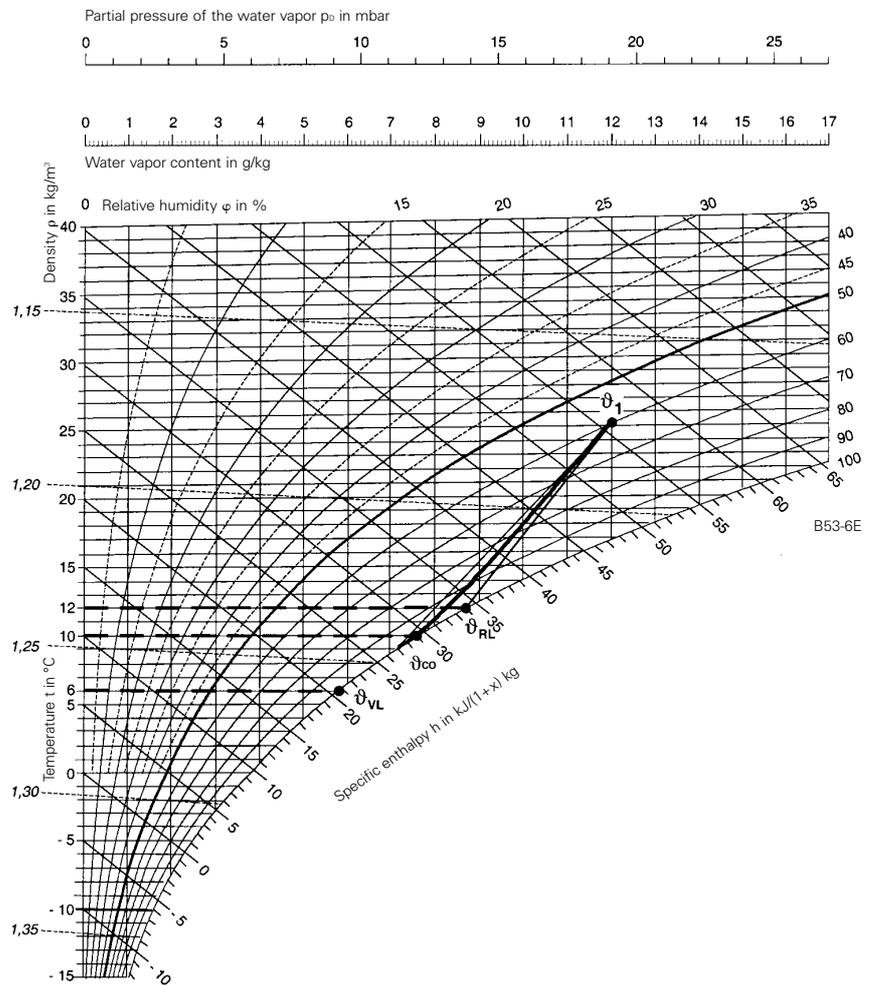
Fig. 3-6 Actual cooling process for a wet cooling surface

Bypass effect: By bypass effect, we refer to the condition that, in the cooler, the only part of the water in the air that condenses is the part that comes in intimate thermal contact with the cooling surfaces. The other part of the air, the "bypass air", leaves the cooler essentially unchanged. The air leaving the cooler is thus a mixture of unsaturated heated air and saturated colder air. This means that the state change in the cooler moves along a curve bent downwards.

However, the smaller the spacing between the lamella in the cooler and the more numerous the number of cooler tubes, the more air contacts the cooler and the straighter the curve. When calculating the cooling output needed for a modern cooler, you can ignore the bypass effect.

3.4.2 Cooling output In air conditioning systems, state changes (e.g., cooling and humidifying) are not normally one-sided. There is an additional parameter that provides the relationship of sensible (perceptible) heat portion (temperature change) to the total change in enthalpy (temperature and water content). This parameter is called the

Sensible Heat Factor or SHF

If, for example, air having an initial state of t_1 = 25 °C and x_1 = 9 g/kg is cooled and dehumidified to t_2 = 15 °C and x_2 = 7.8 g/kg, then the required cooling output can be divided into a sensible cooling output (cooling from 25 °C to 15 °C) and a latent cooling output (dehumidifying from 9 g/kg to 7.8 g/kg). We calculate the sensible heat factor for this example as follows:

$$SHF = \frac{Q_{sens}}{Q_{tot}} = \frac{\Delta h_{sens}}{\Delta h_{tot}} = \frac{48.5 - 38.5}{48.5 - 35.0} = 0.74 \approx 0.75 \ (\Delta h_{tot} = \Delta h_{sens} + \Delta h_{lat})$$

If the psychrometric chart (Fig. 3-7) has a scale of Q_{sens} / Q_{total} with an associated fixed point (ϑ = 21 °C, x = 8 g/kg), the process of the state change can be determined graphically by a parallel displacement (SHF ≈ 0.75). Or, in the case where the SHF is known, the direction of the state change can be determined and projected by parallel displacement to any arbitrary point on the diagram.
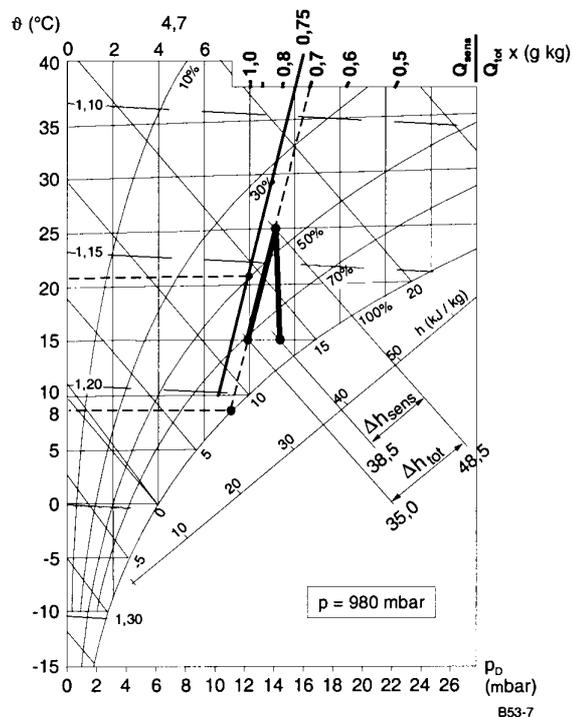

Fig. 3-7   Cooling process with the help of the Sensible Heat Factor (SHF)

3.4.3 Humid cooling   See "Humidification with variable water temperature" (section 3.5.1).

3.5 Air humidification   Air that is too dry can be humidified with water or vapor. If water or vapor is added to air having the state of x_1 and h_1, then the absolute humidity x does not change, rather the enthalpy h is increased by the enthalpy of the added water or vapor.
Δx is the weight and Δh is the enthalpy of the added water or vapor. The state of the humidified air x_2, h_2 is then:

$x_2 = x_1 + \Delta x$ and $h_2 = h_1 + \Delta h$

3.5.1 Humidifying with water (excess water)

Water is sprayed in the jet chamber of the central air conditioning unit. The air flowing past absorbs a part of this water in the form of vapor. Most of the water, however, falls back into the collector. This water can be continually aspirated from the collector and pumped back to the jets. The system thus operates with recycled water most of the time. A level switch ensures that the small volume of water vaporized and absorbed by the air is replaced (Fig. 3-8a).
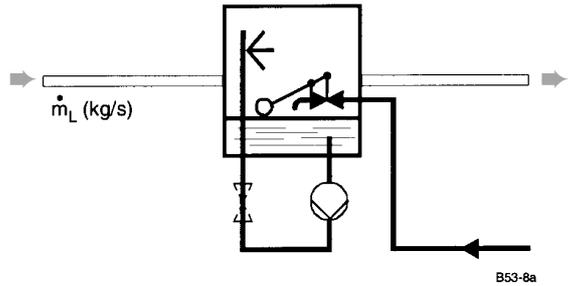

Fig. 3-8a   Humidifying with recycled water

The water from the collector can also be heated or cooled in a heat exchanger before it is returned and sprayed. We refer to this process as humidification at a variable water temperature (Fig. 3-8b).
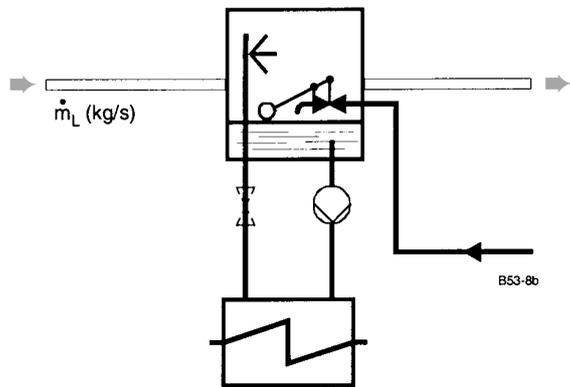

Fig. 3-8b   Humidifying at a variable water temperature

Humidifying using a circulating volume of water

For the consideration of the circulating water humidification process, we take the initial state of the air to be $\vartheta_A = 23\ °C$, $\varphi = 25\ \%$ r.h., and the temperature of the water in the collector to be $\vartheta_W = 17\ °C$ (Fig. 3-9). The air flow into which the water is sprayed absorbs water vapor up to the point of saturation. No external heat is added to vaporize the water. The heat of vaporization is partially taken from the water itself; at the same time, sensible heat passes from the air to the water. Cooling the water, however, takes place only to the extent that just enough sensible heat passes from the air to the water as the water can convert to heat of vaporization.

An equilibrium state thus exists where the saturated air and the circulating water have the same temperature. This equilibrium temperature changes along a line of constant enthalpy until the air reaches a saturated state. This temperature is referred to as the cooling limit because the water can only be cooled to this limiting temperature. The cooling limit is the intersection of the isentropics (adiabatics) with the saturation line and is determined by the starting state of the air, that is, from its temperature and relative humidity. The air and water temperature at the cooling limit is also referred to as wet bulb temperature ϑ_F (see section 2.2.9).
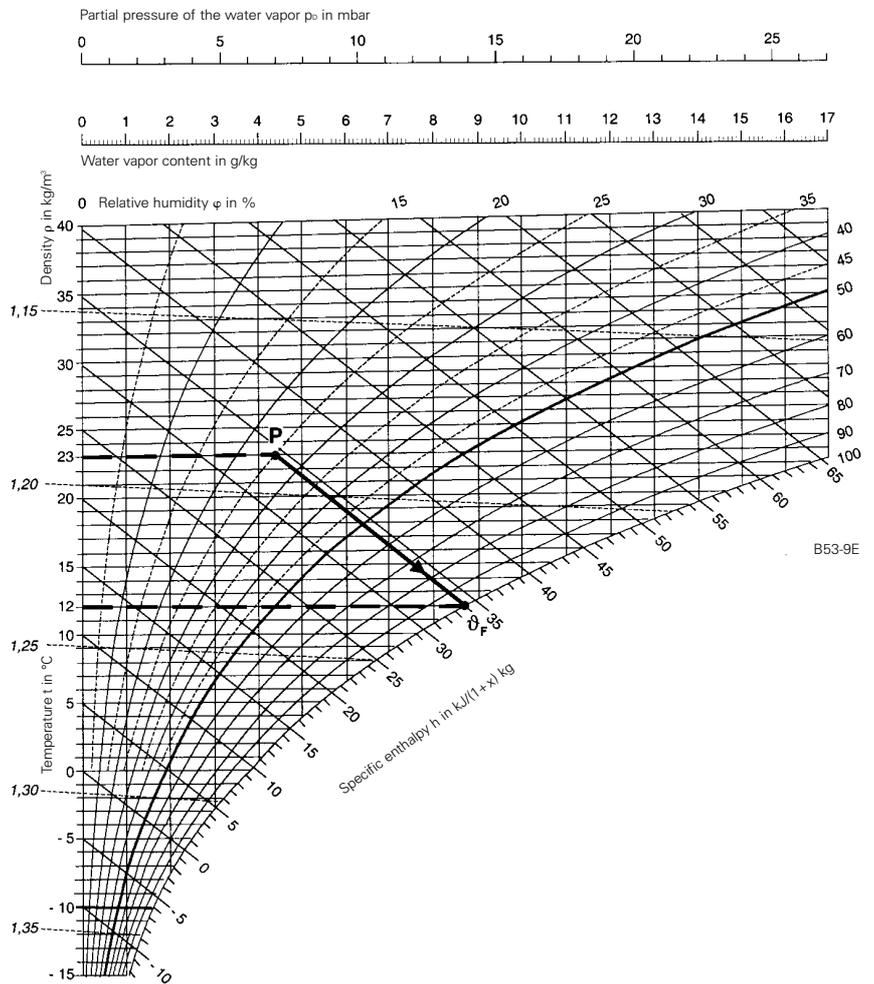
Fig. 3-9   Humidifying with circulation water

The water temperature can initially lie under the wet bulb temperature of the air. In this case, more sensible heat passes from the air to the water. This occurs because the air has to give up its heat to the water, which needs latent heat to vaporize. As a result, the water gradually heats until the wet bulb temperature is reached.
After some time with this type of humidification, the circulation water reaches the wet bulb temperature. This is true regardless of whether it had a higher or lower temperature at the start.

The state change in the psychrometric chart

The state change in the psychrometric chart proceeds from starting state P of the air along an h line of constant enthalpy. It moves toward the wet bulb temperature ϑ_F, which lies on the intersection of the h line with the saturation line. The saturation line is not completely reached, because the efficiency of conventional humidifiers can only reach 95 % under the best of circumstances.
The state change just described does not agree completely with actual practice. This is because the enthalpy of air being humidified, in and around the area of the water being absorbed, decreases or increases (depending on the temperature and volume of the water added). In the important regions of air conditioning operation this error, however, is negligible.

If air of state P_1 with $\vartheta_1 = 27\ °C$, $\varphi_1 = 30\ \%$ r.h. is to be adiabatically humidified to state P_2 with $\varphi_2 = 80\ \%$ r.h., then we first have to calculate the humidification efficiency (Fig. 3-10). Depending on the efficiency, we have to choose an appropriate humidifier having an adequate number of atomizing jets. For this state change, we obtain the following values for the water vapor content of the air from the psychrometric chart:

- At the humidifier entry: $x_1 = 6.7$ [g/kg]
- At the humidifier discharge: $x_2 = 10.2$ [g/kg]
- At the saturation state: $x_3 = 11.1$ [g/kg]

The required humidification efficiency η_B can then be calculated as follows:

$$\eta_B = \frac{x_2 - x_1}{x_3 - x_1} \cdot 100\ \% = \frac{10.2 - 6.7}{11.1 - 6.7} \cdot 100\ \% = \frac{3.5}{4.4} \cdot 100\ \% = 79.5\ \%$$

The data sheets of a humidifier manufacturer provide the required information, that, for example, a humidifier having two jet connections provides a humidification efficiency of approx. 80 % for a water / air ratio of 0.7. The humidification efficiency is among other things dependent on the type and number of atomizing jets, the length of the humidification path, the water pressure and the air speed. In order to reach the required humidification efficiency, the pump has to spray through the humidifier 0.7 times as many kg of water as air. Thus, 700 g water is sprayed per kg air, of which 3.5 g ($x_2 - x_1$) are vaporized.
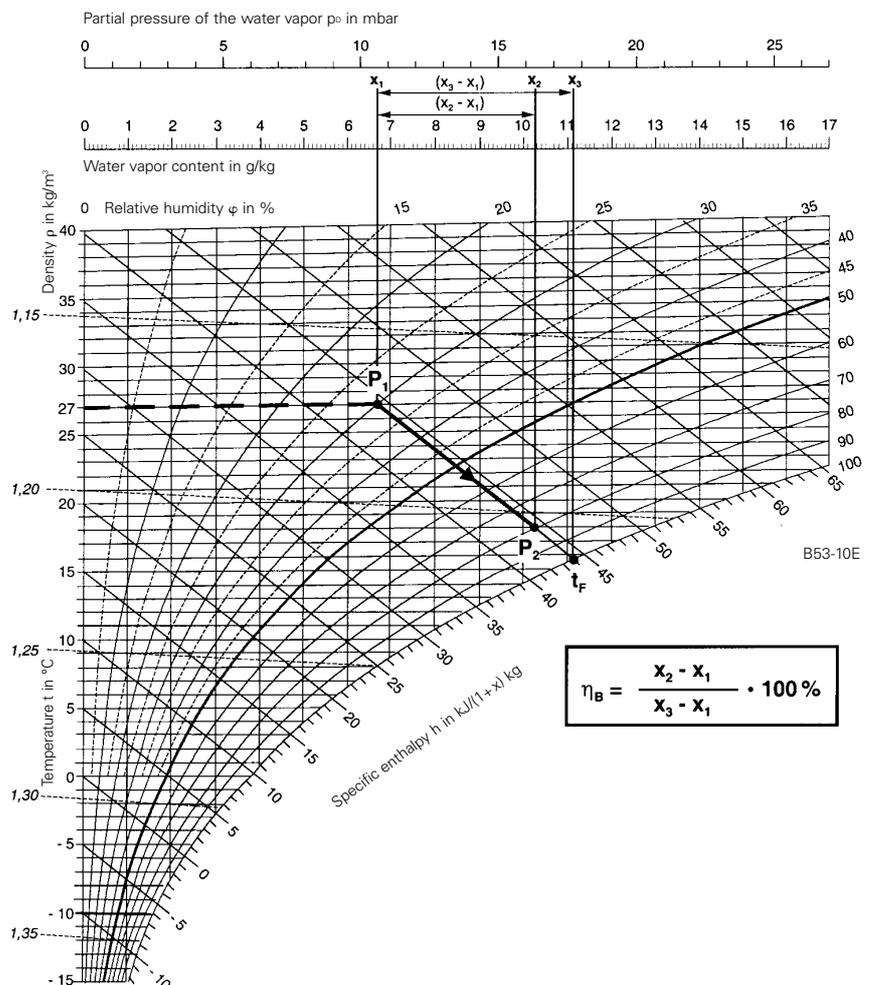

Fig. 3-10 Determination of the humidification efficiency

Humidification with variable water temperature

A relatively large volume of water is sprayed (1 to 1.5 kg water/kg air) into the air stream when using this type of humidification. The water droplets are brought to the desired temperature in the heat transfer unit. The droplets, in this case, are considered to be cooling or heating surfaces (see Fig. 3-8b). Using these prerequisites, the state change in the psychrometric chart takes place on a straight line extending from the starting state of the air to the intersection of the water temperature and the saturation line. The saturation line is not quite reached.

In practice, the water – when flowing through the air washer – is more or less heated (or cooled) depending on the water / air ratio and water inlet temperature. The state change of the air thus starts at the intersection of the water inlet temperature with the saturation line. It ends at the intersection of the water outlet temperature with the saturation line. The result is a line gently curving in the direction of the water outlet temperature.

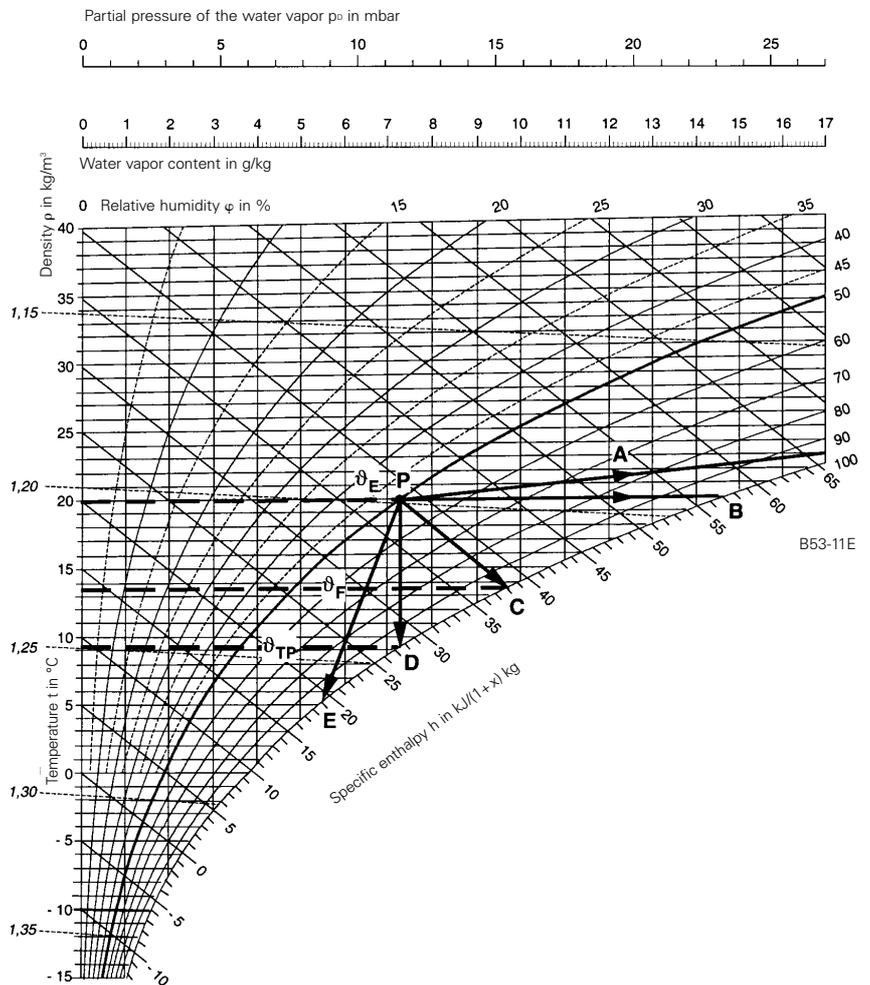Fig. 3-11 illustrates the possible air state changes for a given air state P and different spray water temperatures.

Fig.3-11   Humidifying with a variable water temperature

- If the water outlet temperature is higher than the air inlet temperature ϑ_E, the temperature, the enthalpy and the water content of the air at state P increase (air state lies above ϑ_E, e.g., on the P-A line)
- If the water outlet temperature is the same as the air inlet temperature ϑ_E, then, because of the temperature equality, latent but no sensible heat is transferred. The air temperature thus remains unchanged, but increases its water vapor content and thus also its enthalpy (air state change from P in the direction of B)

- For water inlet temperatures that lie between the wet bulb temperature ϑ_F and the air temperature ϑ_E, the air at state P is cooled while the water content and the enthalpy increase (air state changes proceed within the range of the lines P-B to P-C)
- If the water outlet temperature is the same as the wet bulb temperature ϑ_F of the air at state P, then the air is cooled, the enthalpy remains the same, but the water content increases (air state change from P in the direction of C)
- For water outlet temperatures that lie between the wet bulb temperature ϑ_F and the dew point temperature ϑ_{TP} of the air at state P, the temperature and enthalpy decrease, while the water content increases a small amount (air state change proceeds in the region of the lines P-C to P-D)
- If the water outlet temperature lies under the dew point temperature ϑ_{TP} of the air at state P, the air temperature, enthalpy and water content all decrease (air state change proceeds in the region of the lines P-D to P-E)

The state of the air leaving the spray humidifier cannot only be affected by changing the water inlet temperature, but also by the quantity of water sprayed. If less water is sprayed than is necessary to reach the possible end state of the air, then the increase in temperature (or cooling) of the water is larger. An intermediate state exists for the air: the smaller the water / air ratio, the less the air deviates from its starting condition. The temperature and the relative humidity of the discharge air are also influenced by lowering the water/air ratio (by restricting the volume of water) and by changing the water entry temperature (e.g., by mixing cold and warm water).

3.5.2 Humidifying with steam

This process involves blowing saturated steam into the air duct. The steam is supplied by a local steam air humidifier (e.g. Condair) or by a central steam generation plant (outside supplier). Adding steam causes the water vapor content and the enthalpy of the air to increase. Here, Δx is the weight and Δh is the enthalpy of the added steam.

The enthalpy h_D of saturated steam is very high because it holds heat of vaporization amounting to 2,676 kJ/kg for steam at 100 °C (see tables at the end of this section).

The enthalpy increase during the steam humidification amounts to:

$$\Delta h = h_D \cdot \Delta x$$

If 1 kg of air of e.g. $\vartheta_E = 20$ °C, $x_1 = 5$ g/kg and $h_1 = 32.5$ kJ/ kg is humidified with 6 g of steam at 100 °C (Fig. 3-12), the amount of heat added is:

$$\Delta h = \Delta x \cdot h_S = 0.006 \cdot 2676 \left[\frac{kg\ kJ}{kg\ kg} \right] \approx 16.1\ [kJ/kg]$$

The state of the humidified air consists of:

$$h_2 = h_1 + \Delta h = 32.5 + 16.1 = 48.6\ [kJ/kg]$$
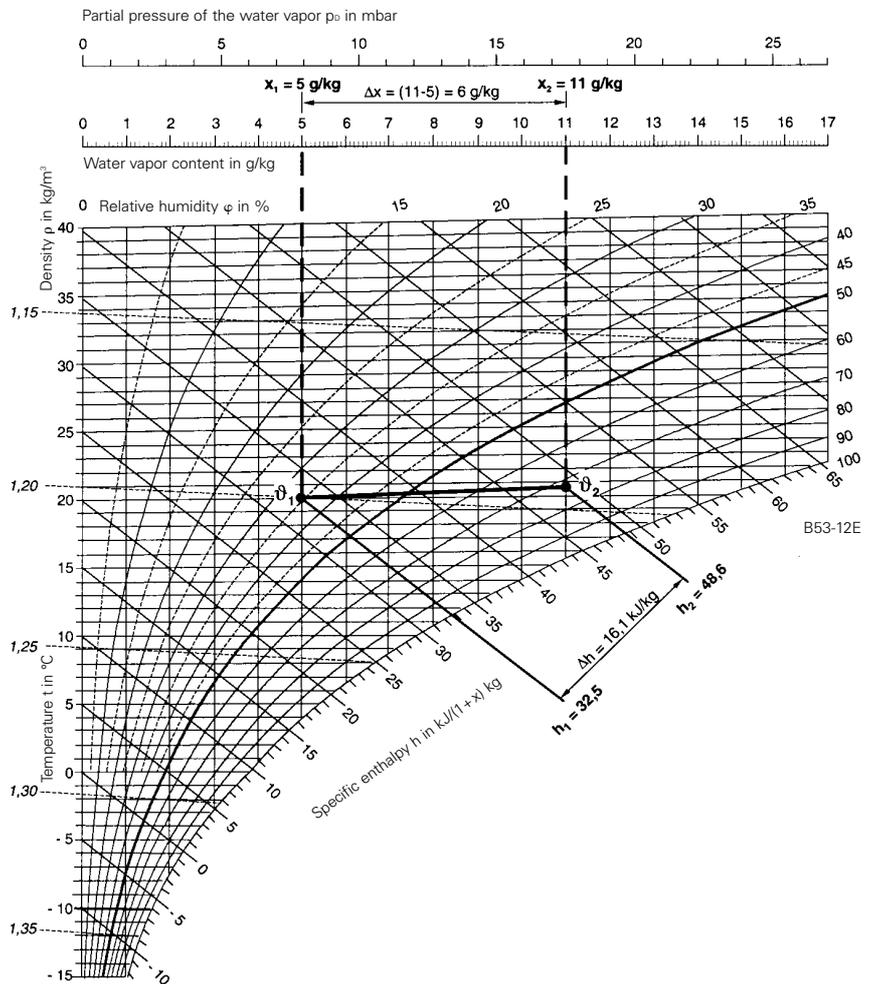$$x_2 = x_1 + \Delta x = 5 + 6 = 11\ [g/kg]$$

Fig. 3-12   Steam humidification using saturated steam (method of calculation)

The direction of the state change in the psychrometric chart is determined when point h_1, x_1 is connected with the new point h_2, x_2.
As shown in the example for steam humidification, the direction of the state change in the psychrometric chart is solely dependent on the enthalpy of the added steam. Because of this, most of the diagrams have a "direction scale" $\Delta h/\Delta x$ on which the direction of the state change can be directly determined and shifted in parallel (Fig. 3-13).
Since $\Delta h = h_S \cdot \Delta x$ and, consequently, $\Delta h/\Delta x = h_S$, the enthalpy h_S of the added steam can be directly indicated on the direction scale (see tables at the end of this section).
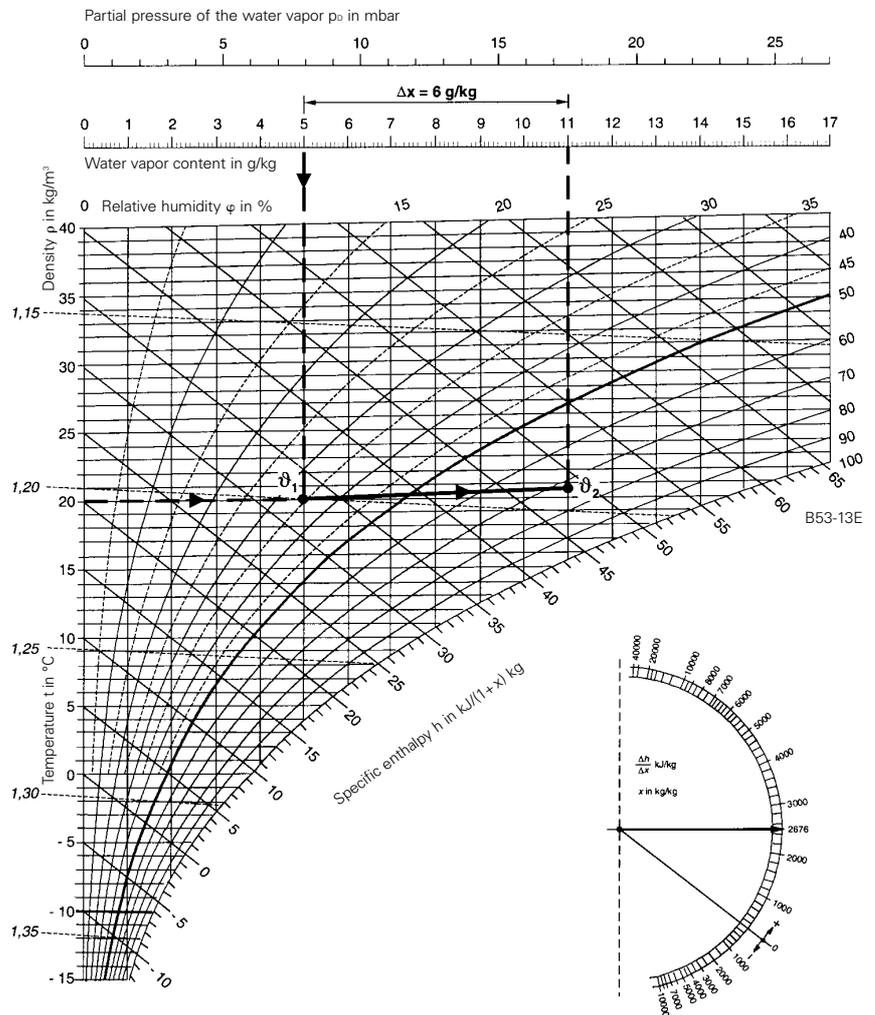The direction of the state change is thus determined as follows:

Fig. 3-13 Steam humidification (state change using the direction scale)

1. Draw a vertical line down from the "water vapor content" scale at x_1 and $x_2 = x_1 + \Delta x$.
2. Draw a straight line from the pivot point of the direction scale to the associated enthalpy h_D (e.g. 2,676 kJ/kg) on the direction scale.
3. Slide this straight line parallel to itself to the starting state ϑ_1 and possibly extend it to vertical x_2. The intersection with this vertical gives the new air state at ϑ_2.

If low pressure saturated steam is used for humidification, the state change essentially follows the isotherms of the air being humidified, that is, there is practically no temperature change, rather only a latent increase of the enthalpy and of the water vapor content of the air.

Variables of water and steam at saturation as a function of temperature

Temperature	Absolute pressure	Enthalpy		Heat of vaporization
[°C]	[bar]	[kJ/kg]		[kJ/kg]
ϑ	p	h_{water}	h_{steam}	r
100	1.013	419.1	2676	2257
105	1.208	440.2	2684	2244
110	1.433	461.3	2691	2230
115	1.691	482.5	2699	2216
120	1.985	503.7	2706	2202
125	2.321	525.0	2713	2188
130	2.701	546.3	2720	2174
135	3.131	567.7	2727	2159
140	3.614	589.1	2733	2144

Variables of water and steam at saturation as a function of pressure

Absolute pressure	Temperature	Enthalpy		Heat of vaporization
[bar]	[°C]	[kJ/kg]		[kJ/kg]
p	ϑ	h_{water}	h_{steam}	r
1.0	99.6	417.5	2675	2258
1.5	111.4	467.1	2693	2226
2.0	120.2	504.7	2706	2202
2.5	127.4	535.3	2716	2181
3.0	133.5	561.4	2725	2163
3.5	138.9	584.3	2732	2147
4.0	143.6	604.7	2738	2133
4.5	147.9	623.2	2743	2120
5.0	151.8	640.1	2748	2107

3.6 Air drying

During air drying, the water content of the air (absolute humidity) is reduced. This can happen in different ways:

- Cooling the air using water condensation (subcooling method)
- Absorption of the water by absorption material (absorption method)
- Mixing in dry air

3.6.1 Subcooling method

The too humid air is brought into contact with cooling surfaces, whose temperature lies below the dew point of the air. As a result, part of the water vapor condenses out on the cooling surface which reduces the water vapor content of the air. The state change in the psychrometric chart (Fig. 3-14) is represented by a straight line, which runs from the starting state P of the air to the intersection of the average cooling surface temperature ϑ_{CO} with the saturation line.
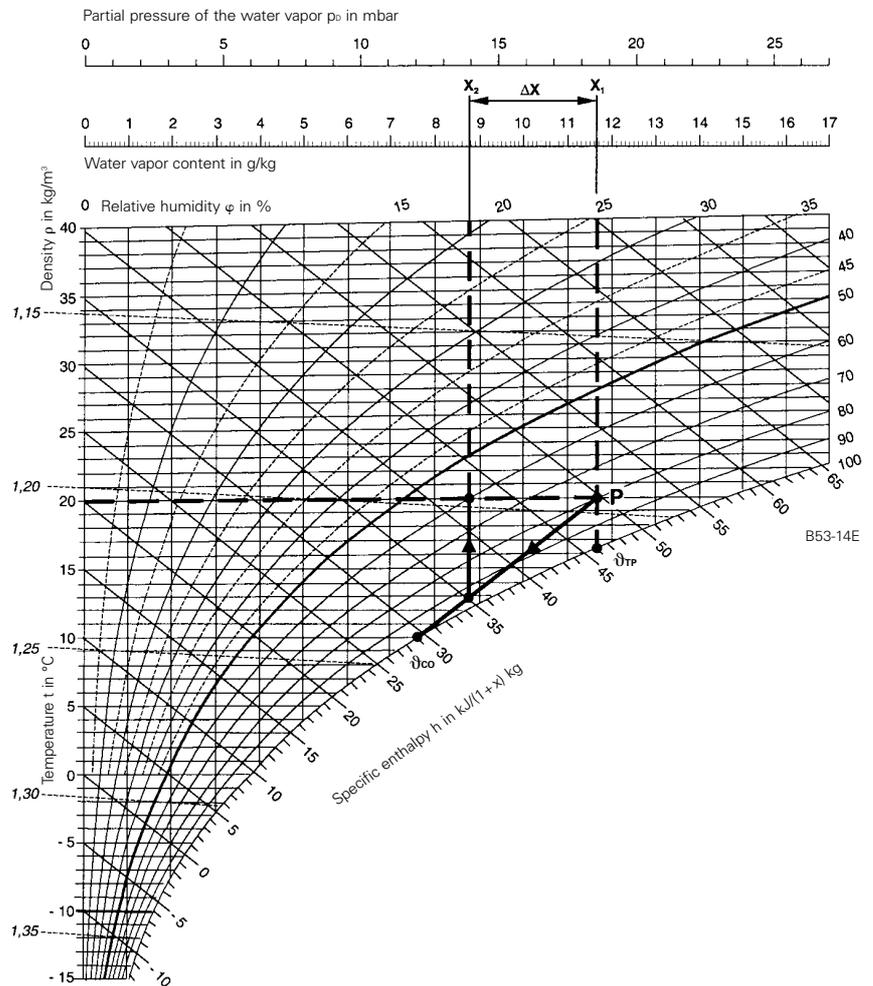
Fig.3-14 Drying air using the subcooling method

The amount of water condensing out depends on the cooling capacity. The water vapor content decreases by Δx, while the relative humidity of the air increases. Air cooling is always associated with this type of dehumidification, which is why the air must be reheated in most cases. Heating, of course, lowers the relative humidity.

3.6.2 Absaorption method

This method involves bringing the air into contact with hygroscopic material, that is, material which can absorb water vapor from the air. The most often used hygroscopic material is silica gel. The water vapor adheres by absorption and condenses on the exceptionally large surface area of the silica gel (1 gram of silica gel has the extraordinary surface area of 300 to 500 m^2). The released latent heat of vaporization raises the air temperature, while the absolute and relative humidity are reduced. When the silica gel becomes saturated, it can be regenerated by heating it to approximately 150 to 200 °C, using, for example, hot air.

Since no heat is transported into or out of the system for this absorption process, the enthalpy of the air remains constant. Fig. 3-14 shows these (adiabatic) state changes in the psychrometric chart.
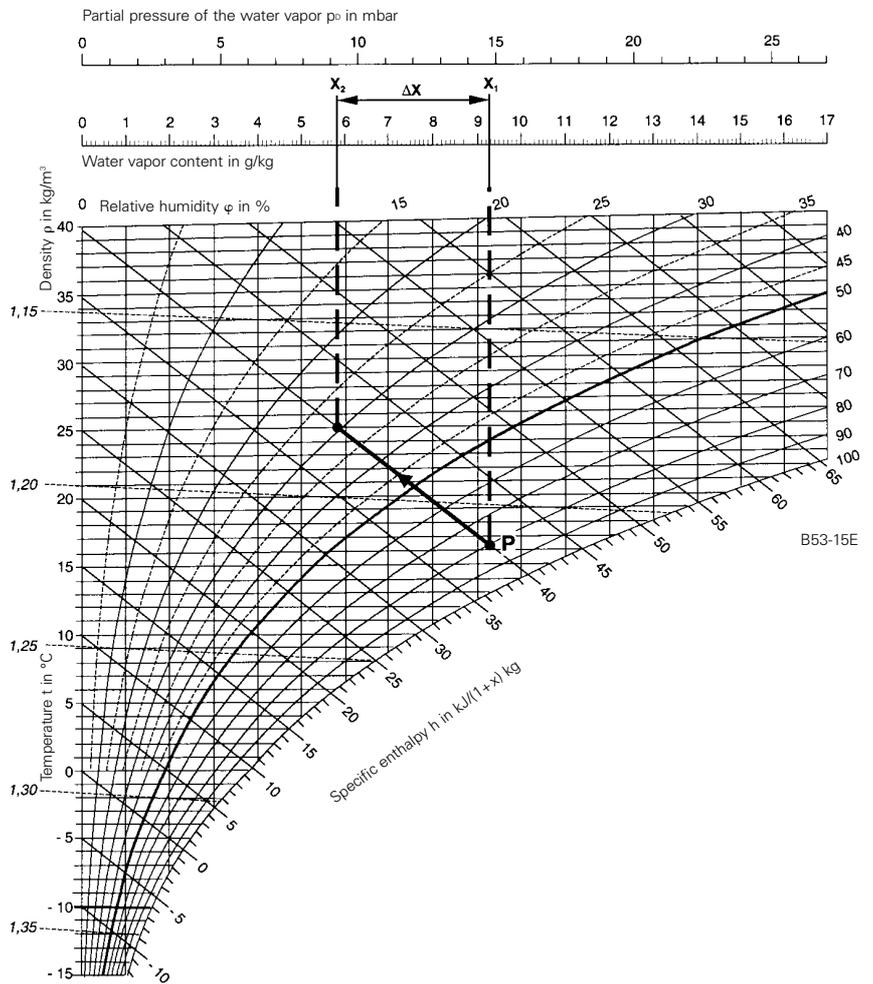
Fig. 3-15 Drying air using the absorption method

3.6.3 Drying air by mixing in dry air

This method is shown in Fig. 3-16. Here (mostly cold) air L_2 is mixed with humid air L_1 ($\varphi_1 = 70$ %, $x_1 = 14$ g/kg). The vapor content of L_2 is considerably below that of L_1. The state of the air after mixing (M_1) is determined by the mixing ratio (see section 3.2).

If the temperature of the mixed air M_1 is lower than that of the humid air L_1, then it has to be reheated to the initial temperature. Following this reheating, air state M_2 results having the same temperature as the humid air L_1. However, the vapor content is reduced ($x_2 = 10$ g/kg) and, as a result, the mixture has a lower relative humidity of $\varphi_2 = 50$ % (e.g. in swimming pool applications).
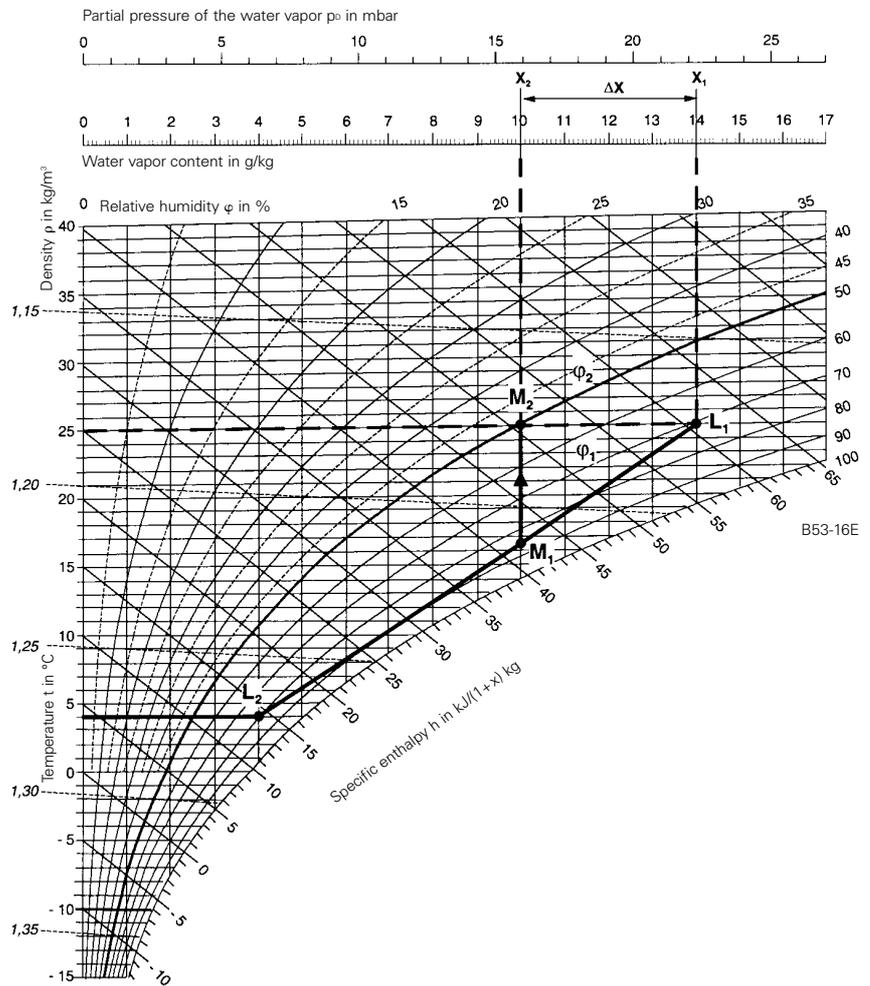
Fig. 3-16   Air drying by mixing and reheating

3.7 Converting air volumes to air masses

From chapter 1 "Thermodynamic fundamentals" we know that the air density is dependent on the other variables of pressure, temperature and water vapor content.

Any given psychrometric chart is only valid for a certain air pressure (converting to other air pressures is covered in chapter 4 "Calculation of the altitude correction"). As a result, the determination of density using the psychrometric chart can be restricted to the two variables of temperature and water vapor content.

In the example of Fig. 3-17, for air state P_1, having the temperature $\vartheta_1 = 15$ °C and the relative humidity $\varphi_1 = 65$ %, the air density is $\rho_1 = 1.22$ kg/m³. For a volumetric air flow rate of $V_1 = 1000$ m³/h, the air mass flow rate is:

$$\dot{m}_1 = \rho_1 \cdot \dot{V}_1 = 1.22 \cdot 1{,}000 \left[\frac{\text{kg m}^3}{\text{m}^3 \text{ h}} \right] = 1{,}220 \left[\frac{\text{kg}}{\text{h}} \right]$$

If we now heat the air (without changing the absolute humidity x), the air mass m remains constant. In this example, the air is heated at constant absolute humidity x to state P_2 having temperature $\vartheta_2 = 25$ °C, relative humidity $\varphi_2 = 35$ % and air density $\rho_2 = 1.18$ kg/m³. The volumetric air flow rate $V_1 = 1000$ m³/h then changes to:

$$\dot{V}_2 = \frac{\dot{m}}{\rho_2} = \frac{1{,}220}{1.18} \left[\frac{\text{kg m}^3}{\text{h kg}} \right] = 1{,}034 \left[\frac{\text{m}^3}{\text{h}} \right]$$

The volumetric air flow rate changes from 1,000 to 1,034 m³/h as a result of the temperature increase, while the air mass flow rate (at constant absolute humidity x) remains unchanged.
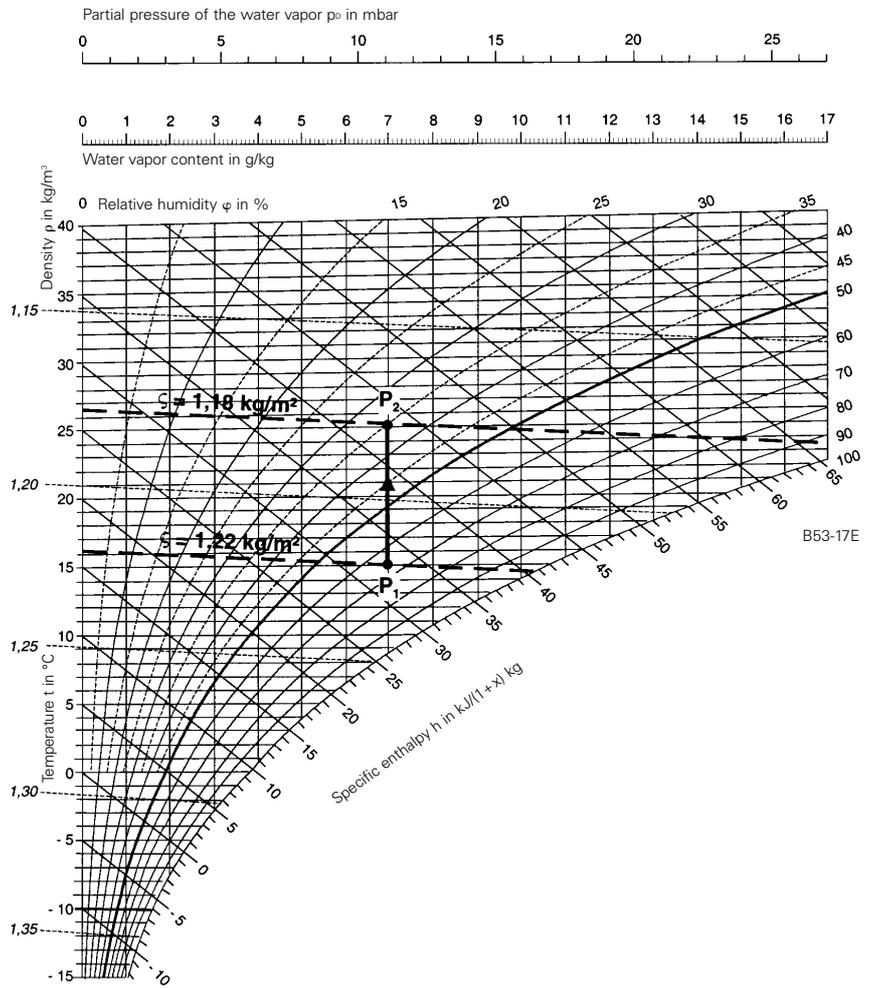
Fig. 3-17   Change in air density upon a temperature increase

4. Calculation of the altitude correction

4.1 Influence of altitude on air pressure

The barometric (atmospheric) pressure has to be taken into account for all calculations involving the psychrometric chart. The barometric pressure is mainly dependent on the elevation above sea level. Thus, each psychrometric chart has a label specifying for which elevazion above sea level or for which barometric pressure it is valid.
Units used for elevation include: meters above sea level, millibar [mbar], kilopascals [kPa], millimeters of mercury [mm Hg] or [Torr].
The psychrometric chart used in engineering texts and schools are for sea level use corresponding to 101.3 kPa = 1,013 mbar = 760 Torr.
In this chapter, you will learn how to convert or even redraw a psychrometric chart to work with another elevations above sea level.

4.2 Calculating the correction factors for φ and ρ

Most psychrometric charts include a table or diagram specifying correction factors for various elevations above sea level. From the diagram you can determine the average barometric value or air pressure p for a specific elevation H.
Then, using the formula:

$$k = \frac{p_1}{p_2}$$

you can calculate the correction factor k for the relative humidity φ and density ρ. Here, p_1 is the reference pressure of your psychrometric chart and p_2 is the average barometric pressure for the elevation above sea level under consideration. An example:
At 1,000 meters above sea level, the average barometric pressure p_2 = 89.9 kPa. The psychrometric chart, however, is referenced to p_1 = 101.3 kPa.
The correction factor k is then:

$$k = \frac{p_1}{p_2} = \frac{89.9}{101.3} \left[\frac{kPa}{kPa} \right] = 0.887$$

Using this correction factor, you can now determine the corrected values for the relative humidity lines. The values φ = 10 %, 20 %, 30 % relative humidity (r.h.), etc., are multiplied by this correction factor. The individual results yield the new values for the existing lines of constant relative humidity.
Thus, using the above correction factor of k = 0.887 yields:

- Humidity line φ =   10 % r.h. becomes a humidity line where
  φ =   8.87 % r.h.
- Humidity line φ =   20 % r.h. becomes a humidity line where
  φ =   17.74 % r.h.
- Humidity line φ = 100 % r.h. becomes a humidity line where
  φ = 88.7   % r.h.

The correction factor is always less than one for psychrometric charts having a sea level reference for locations above sea level. The

especially important saturation line having φ = 100 % r.h. will not be found using this technique. It has to be constructed with the help of a special nomograph (Fig. 4-1).
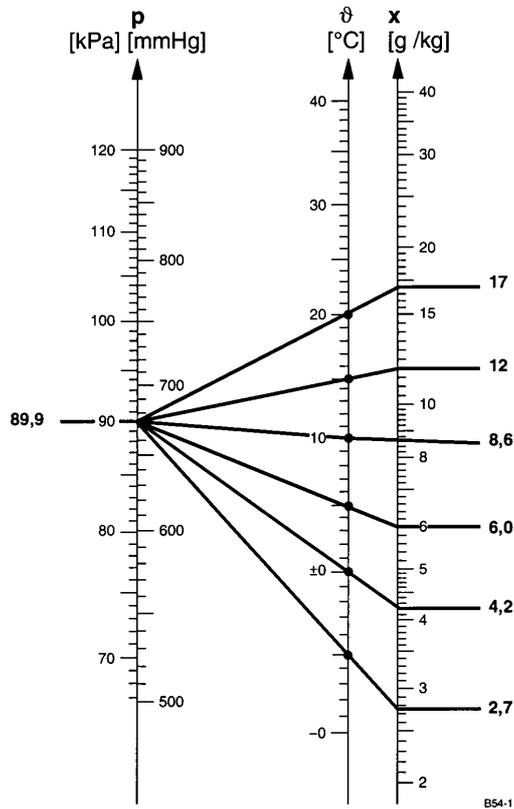
Fig. 4-1 Determining the absolute humidity x to construct a saturation line for a specified air pressure p (shown drawn for p = 89.9 kPa, or H = 1,000 meters above sea level).
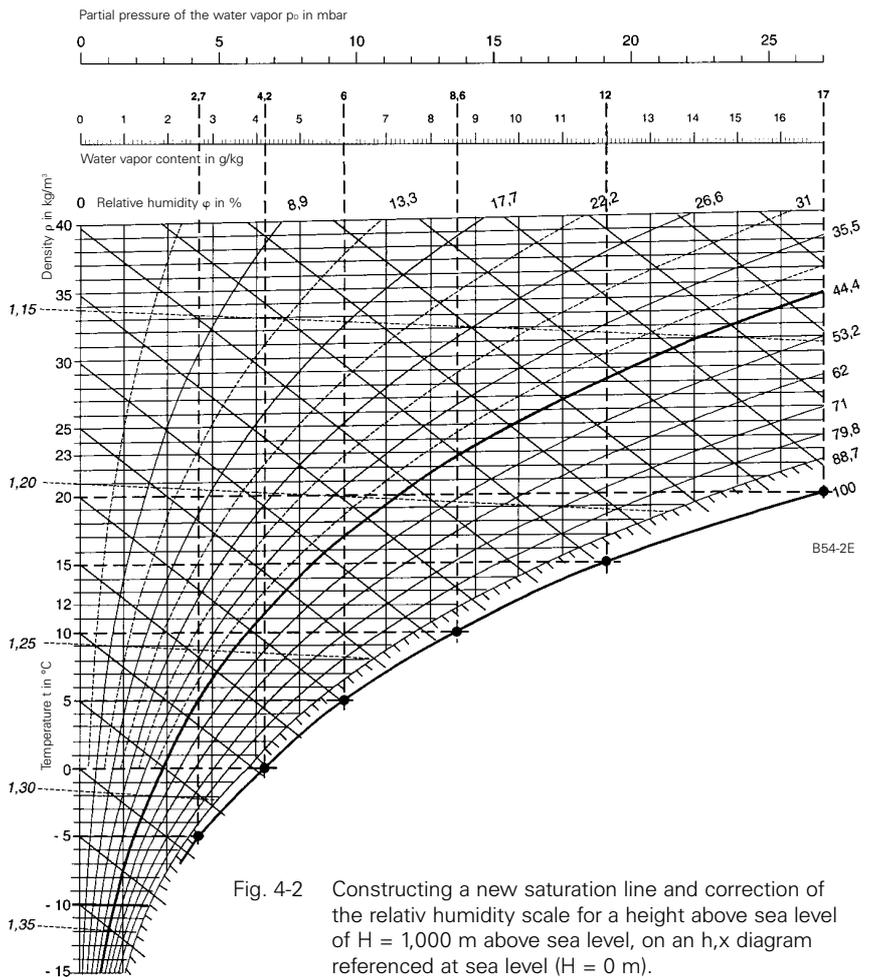

Fig. 4-2 Constructing a new saturation line and correction of the relativ humidity scale for a height above sea level of H = 1,000 m above sea level, on an h,x diagram referenced at sea level (H = 0 m).

4.3 Constructing the saturation line

Constructing a saturation line for e.g. 1,000 meters above sea level in a psychrometric chart referenced at sea level involves the following steps:

1. Determine the average barometric pressure for 1,000 meters above sea level from the table (89.9 kPa) and then mark this value on the p scale of the nomograph (Fig. 4-1).
2. Starting from this marked value of 89.9 kPa, draw a straight line through the temperature scale at $\vartheta = -5$ °C to the x scale.
3. Then, from the same value of 89.9 kPa, draw a straight line through the temperature scale at $\vartheta = 0$ °C to the x scale.
4. Continue in the same manner (repeat steps 2 and 3) in steps of 5 K to determine the value of x in each case.
5. Transfer the x values (g/kg) to the psychrometric chart on the x axis, Fig. 4-2.
   NOTE: The x scale of the nomograph is logarithmic.
6. In the psychrometric chart, extend the temperature lines (–5 °C, 0 °C, etc.) beyond the saturation line into the mist region.
7. Starting from the transferred x values in the psychrometric chart, draw verticals to the intersection of the extended temperature lines.
8. The connecting line to this intersection gives the new saturation line having $\varphi = 100$ % r.h.

You can construct all the altitude lines of the saturation line using this method. To construct the relative humidity lines for values < 100 %, use the procedure explained in chapter 2 "Structure of the psychrometric chart", section 2.2.6 "Lines having a constant relative humidity".

Principle:

In psychrometric charts referenced to sea level, the saturation lines for elevations above sea level always lie in the original mist region, that is, outside the already drawn relative humidity lines.
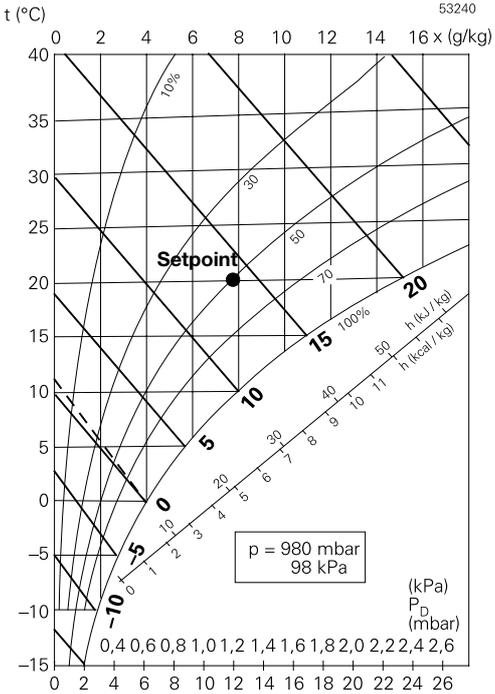
4.4 Changing the air state variables

The influence of altitude on the individual air state variables can be determined from the modified psychrometric chart. If some variables are held constant, changes in the other variables can be determined. For example:
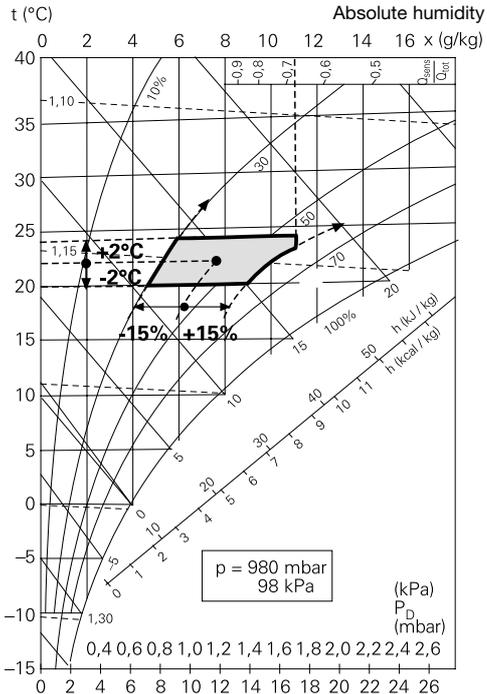
- Air in a saturated state of $\varphi = 100$ % r.h. at 1,000 meters above sea level holds approx. 1.2 g/kg more water vapor than at sea level. This means that for air, where ϑ and φ are constant, the water vapor content x, that is the absolute humidity, increases with altitude
- The enthalpy of saturated air at 1,000 meter is approximately 3 kJ/kg higher than at sea level. The result is that the enthalpy h of air (where ϑ and φ are constant) increases with altitude
- Air having $\varphi = 100$ % r.h. at sea level has a relative humidity of $\varphi = 88.7$ % at 1,000 meters. The difference $\Delta\varphi$ amounts to 11.3 %. The relative humidity of air where ϑ and φ are constant thus decreases with altitude

Setpoint / comfort range in connection with h-x-compensated contro

Setpoint Today, it is still common practice to predefine the room air state based on a fixed setpoint of 20 °C and 50 % r.h., for example. It is then the task of the control system to reach and maintain that setpoint by making use of the various pieces of equipment of the air conditioning plant.

Comfort range New innovative methods provide accurate comfort control by using the so-called comfort range from the psychrometric chart as the setpoint.
There are two major reasons for using an entire field from the psychrometric chart as the setpoint.

1. Comfort: optimum working or living conditions do not only exist at a specific setpoint of room temperature and room humidity but also within a certain band width of the setpoint.
   A temperature setpoint of 22 °C and a relative humidity setpoint of 45 % with variations of ±2 °C and ±15 % r.h. respectively are typical levels used for office spaces. Also, at high temperatures, maximum limitation of absolute humidity should be provided to avoid "muggy" conditions. Typically, this limit value lies at about 10 g/kg.
2. Energy savings: the larger the area of the comfort range, the better the energy savings opportunities.

Based on the comfort range, the following can be said about the air handling functions:
The optimum room air state that should be strived for lies at the border line or inside the comfort range while observing a favourable sequence of operations from an energy savings point of view. Depending on the pre-conditioned air and the supply air temperature setpoint, there is a demand for heating, cooling, humidifying, dehumidifying or for combinations of them. In the cases where heating-humidifying, cooling-humdifying or dehumidifying-reheating is required, the relevant setpoint is identical to the corner of the comfort range, which is best from the energy savings point of view.
In the case of cooling only, the setpoint is obtained by projecting the actual value to the upper border line of the comfort range.
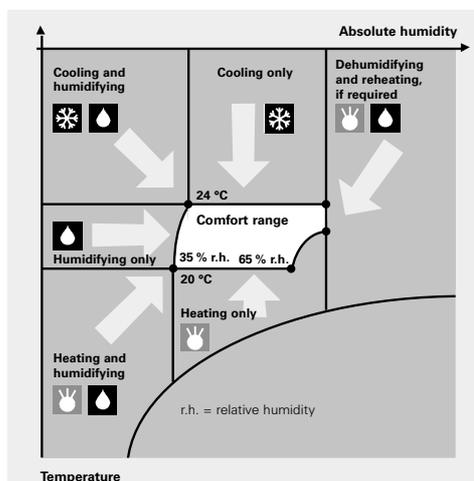In the case of heating only, a projection is made to the lower border line.
For the area in which the relative humidity represents the limit, a special case exists. Here, the temperature setpoint is shifted along the line of relative humidity. This results in more heating but, on the other hand, no cooling is required in that area. This offers cost savings since heating is considerably less expensive than cooling.
In the case of humidification only, the actual value of humidity is projected to the line of relative humidity. This leads to higher humidification costs but, on the other hand, no cooling is required in that area. This also offers cost savings since humidification is less expensive than cooling.

The method of h-x-compensated control makes use of the interrelationship between setpoint and comfort range in order to ensure an optimum sequence of operations from an energy consumption point of view.
A detailed description of h-x-compensated control with a new innovative approach is given in a separate paper.

Reference

The contents of this brochure is an abstract of the training module
B05HV_en "Psychrometric chart: Structure and application"

Siemens Switzerland Ltd
HVP
Training
Gubelstrasse 22
CH-6301 Zug

More technical brochures	ASN-No.	Titel
	0-91899-de	Das h, x-Diagramm
	0-91899-en	The psychrometric chart
	0-91900-de	Gebäudeautomation – Begriffe, Abkürzungen und Definitionen
	0-91900-en	Building automation
	0-91910-de	Messtechnik
	0-91910-en	Measuring technology
	0-91911-de	Regeln und Steuern von Heizungsanlagen
	0-91911-en	Control of heating plants
	0-91912-de	Regeln und Steuern von Lüftungs-/Klimaanlagen
	0-91912-en	Control of ventilation and air conditioning plants
	0-91913-de	Regeltechnik
	0-91913-en	Control technology
	0-91914-de	Kältetechnik
	0-91914-en	Refrigeration technology
	0-91915-de	Wärmerückgewinnung im Kältekreislauf
	0-91915-en	Heat recovery in the refrigeration
	0-91916-de	Einführung in die HLK- und Gebäudetechnik
	0-91916-en	Introduction to building technology
	0-91917-de	Hydraulik in der Gebäudetechnik
	0-91917-en	Hydraulics in building systems
	0-91918-de	Stetige Leistungsregelung im Kältekreislauf
	0-91918-en	Modulating capacity control in the refrigeration cycle

www.siemens.com/buildingtechnologies