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Psychrometric chart – structure and applications

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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 breath. A grown person, for example, requires approximately 0.5 m^3 air to breathe per hour to maintain life processes. In addition, the air fulfils other vital requirements. For example, air absorbs vast amounts of water in form of vapor from the surfaces of lakes and oceans, transports it large 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

Table 1-1 Composition of air

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.

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 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 de-humidified. 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 state changes.

1.4 Variables

1.4.1 Temperature θ

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 differences $\Delta\theta$ (often also ΔT) are always specified in K.

1.4.2 Absolute humidity x

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 φ

At a given temperature and pressure, dry 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 possible 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 1013 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 φ = 50 % RH.

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 ρ

Density is mass divided by volume, e.g., the amount of mass in kilograms of a material having a volume V of one cubic meter. The units of density ρ are thus kg/m^3.

At 0 °C and sea level the density ρ is:

- Dry air: ρ = 1.293 kg/m^3
- Water vapor: ρ = 0.804 kg/m^3

1.4.5 Specific heat c

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 1013 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 Thermal capacity or enthalpy h

One of essential air calculations is the determination of the quantity of heat required to reach an air state in a room defined by a 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 de-humidifying to be converted to the state desired. Most of these processes also result in changes in the thermal capacity (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 in kJ/kg. Absolutely dry air having a temperature θ = 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 a 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 air from θ_1 = 0 °C and x_1 = 3 g/kg to θ_2 = 22 °C and x_2 = 7 g/kg? (air pressure = 1013 mbar)

Solution:

Two quantities of heat are required to heat the 1000 kg air from 0 °C to 22 °C (sensible heat) and to vaporize 1000 kg · 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 r for water is both temperature and pressure dependent. It can be set to 2,450 kJ/kg with the water vapor partial pressure < 0.1 bar and temperatures < 45 °C.
Thus for:

- Heating the 1000 kg air from 0 °C to 22 °C:
  $Q_{Air} = m_{Air} \cdot c_P \cdot \Delta T$ = 1,000 kg · 1.01 kJ/(kg·K)· 22 K = 22,220 kJ

- Heating the 4 kg water from 0 °C to 22 °C:
  $Q_{Water} = m_{Water} \cdot c_W \cdot \Delta T$ = 4 kg · 4.19 kJ/(kg·K)· 22 K = 370 kJ

- Vaporizing the 4 kg water:
  $Q_{Vapor} = m_{Water} \cdot r$ = 4 kg · 2450 kJ/kg = 9800 kJ

The calculation of this partial quantity of heat yields the required quantity of heat for the state change

$Q_{1 \rightarrow 2} = Q_{Air} + Q_{Water} + Q_{Vapor}$ = (22,220 + 370 + 9800) kJ = 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 p

Pressure is a force acting on a surface. The weight of the air on the surface of the earth is called atmospheric pressure. At sea level the average value is 1013 mbar or 760 mmHg.

The pressure unit in the international system of units (SI units) is:

- 1 Newton/m^2 = 1 N/m^2 = 1 Pa (Pascal)

The unit bar, however, is used for practical purposes in the HVAC-technology:

- 1 bar = 1000 mbar (millibar) = 10^5 N/m^2 = 10^5 Pa (100,000 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 (mmWC) or in meters of a water column (mWC).

- 1 bar = 10,130 mmWC or 10.13 mWC
- 1 mmWC = 10 Pa or 1 mWC = 10 kPa

1.4.8 Medium flow q_v or q_m

Medium flow involves:

- Volume flow q_v in m3/s or m3/h
- Mass flow q_m in kg/s or kg/h

A homogeneous liquid or gaseous medium having a volume in m^3 or a mass in kg, flowing uniformly through a flow cross section in one second, is referred to as a volume flow or a mass flow.

The term flow rate is also frequently used. When specifying flow rates both SI units are used m^3/s or kg/s.

2 Structure of the psychrometric chart

2.1 General

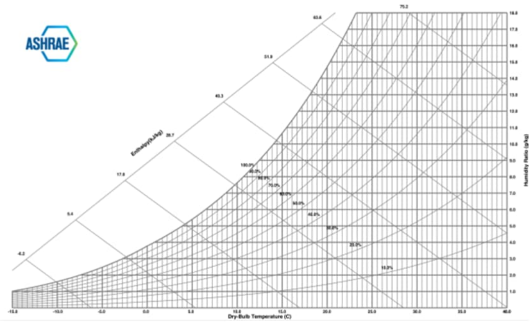
Early on, attempts were made to simplify calculations dealing with air state changes by using graphs. There are different types of diagrams used for psychrometric calculations. The h,x diagram (formerly called the “Mollier” diagram) is commonly used in Europe, while the psychrometric chart (often also called the “Carrier” diagram) is used in the USA and Asia. Both have the same principle format. Only the axes directions are different. Temperature is on the y (vertical) axis in the h,x diagram, while water content is on the x (horizontal) axis. The reverse is true for the psychrometric chart.

Source: Carrier

Dr. Willis Haviland Carrier (1876-1950) was an American engineer and innovator/inventor. He is considered as the “father” of modern air-conditioning. In 1911 he developed the Carrier-Diagram with the name “psychrometric chart”, which is also often referred to as the “t,x-diagram”. It is used to illustrate the air state regarding temperature and humidity. It is mainly used in the USA as well as in the Asian region.

Source: TU Dresden

Prof. Richard Mollier was a German engineer (1863-1935 a.o. TU Dresden). In 1923 he presented the h,x-diagram at a congress of thermodynamics experts in Los Angeles. He based his ideas on those of Dr. Willis Carrier. The diagram was named after Mollier as a credit to his accomplishments until the SI-Units were introduced. It was consequently renamed to “h,x-diagram”. It is primarily utilized in European countries.



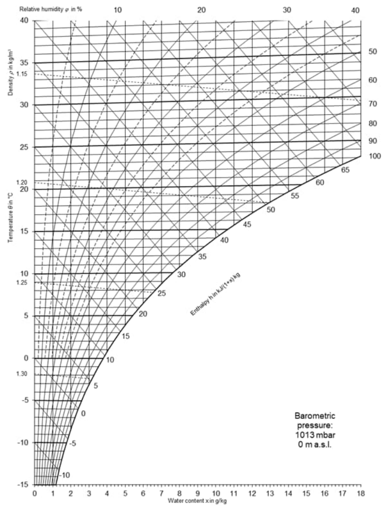


Fig. 2-1 Psychrometric chart (Source: ASHRAE) h,x-diagram (Source: Siemens) in comparison

Psychrometric diagrams are either h,x or t,x diagrams, where θ is temperature in °C, h is enthalpy in kJ/kg, x is the absolute water content of the air in g/kg and φ is the relative humidity in % RH. 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 in °C
- h = enthalpy in kJ/kg (relative to 1 kg dry air)
- x = water content in g/kg (relative to 1 kg dry air)
- φ = relative humidity as % RH

The characteristics and behavior of humid air depend on the barometric air pressure. Thus, any given psychrometric chart can only be drawn for a single specific barometric pressure. The diagram used in this chapter is based on a barometric air pressure of 1013 mbar (= 0 m a.s.l.).

2.2 The structure in detail

2.2.1 Temperature scale and isotherms

The temperature scale serves as the standard variable for the psychrometric h,x 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 towards the right side due to the heat of the increasing water content.

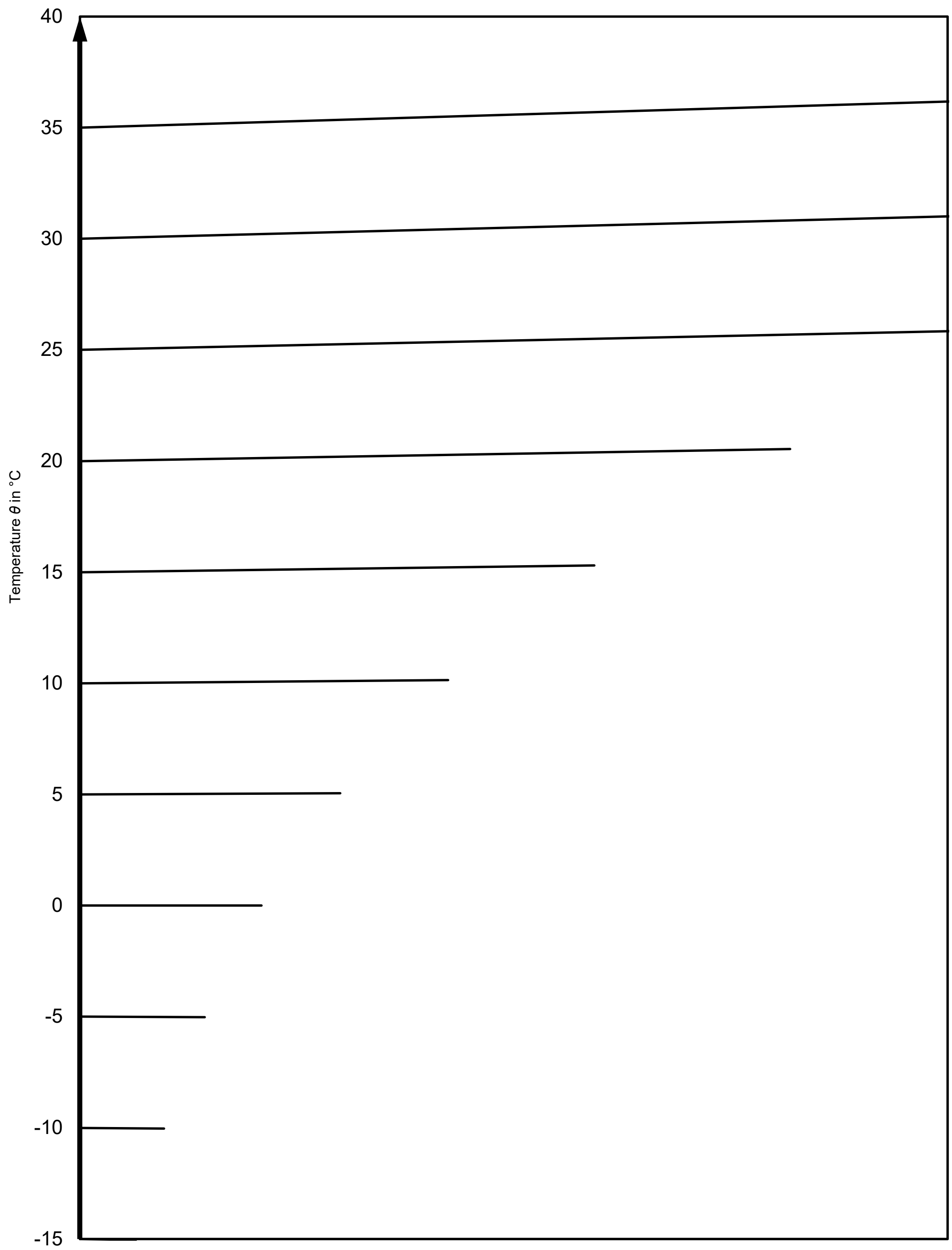


Fig. 2-2 Temperature scale including isotherms

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 θ, 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).

For the examples in chapter 2.2 a temperature of 20 °C and an absolute humidity of 6 g/kg will be used.

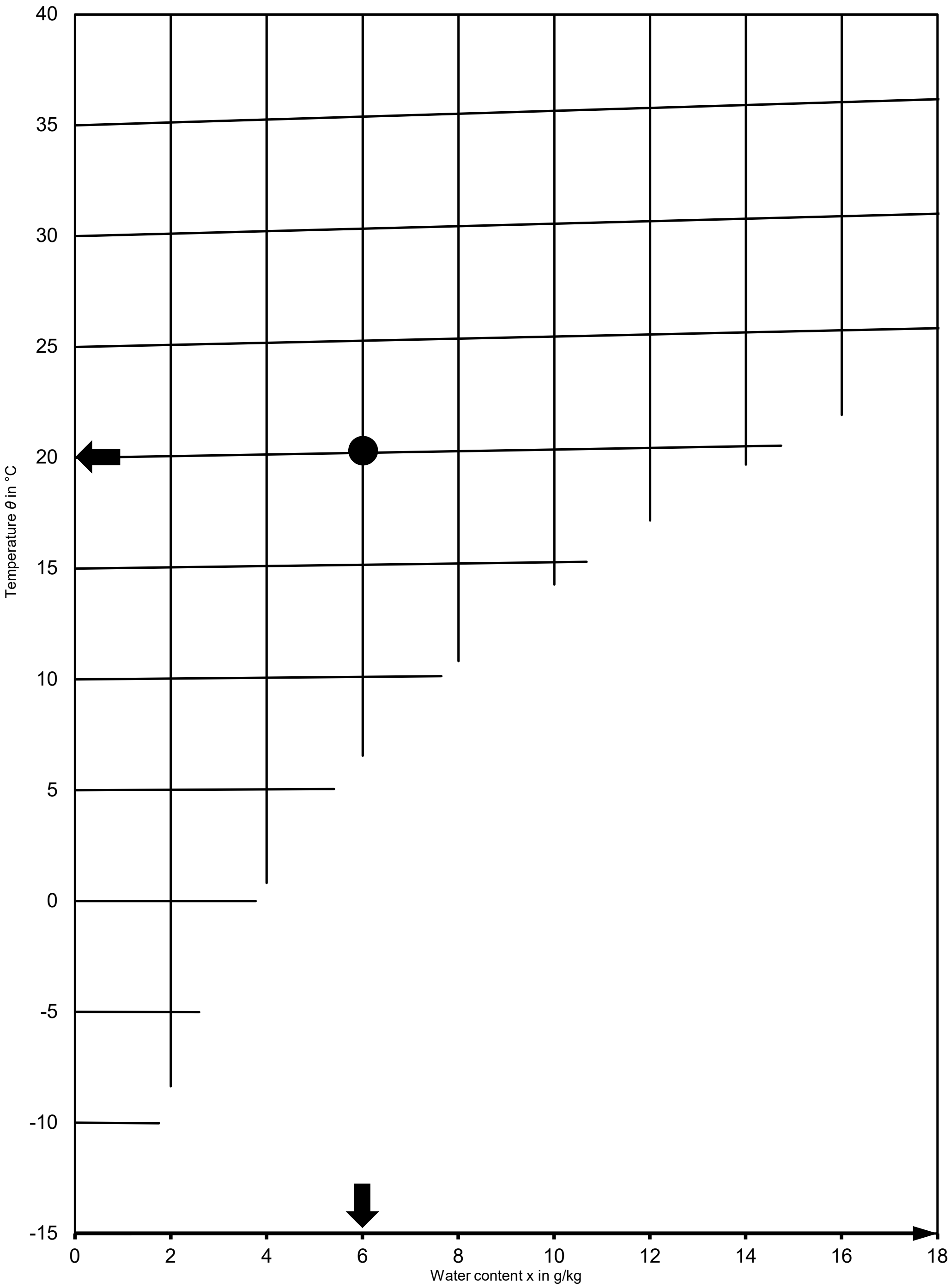


Fig. 2-3 Water content x and dry air temperature θ

2.2.3 Steam pressure scale

Overheated water vapor, gaseous and mixed with air, has a certain steam pressure p_D. This steam 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 steam 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 $\Rightarrow$ $p_D \approx 9.5$ mbar).

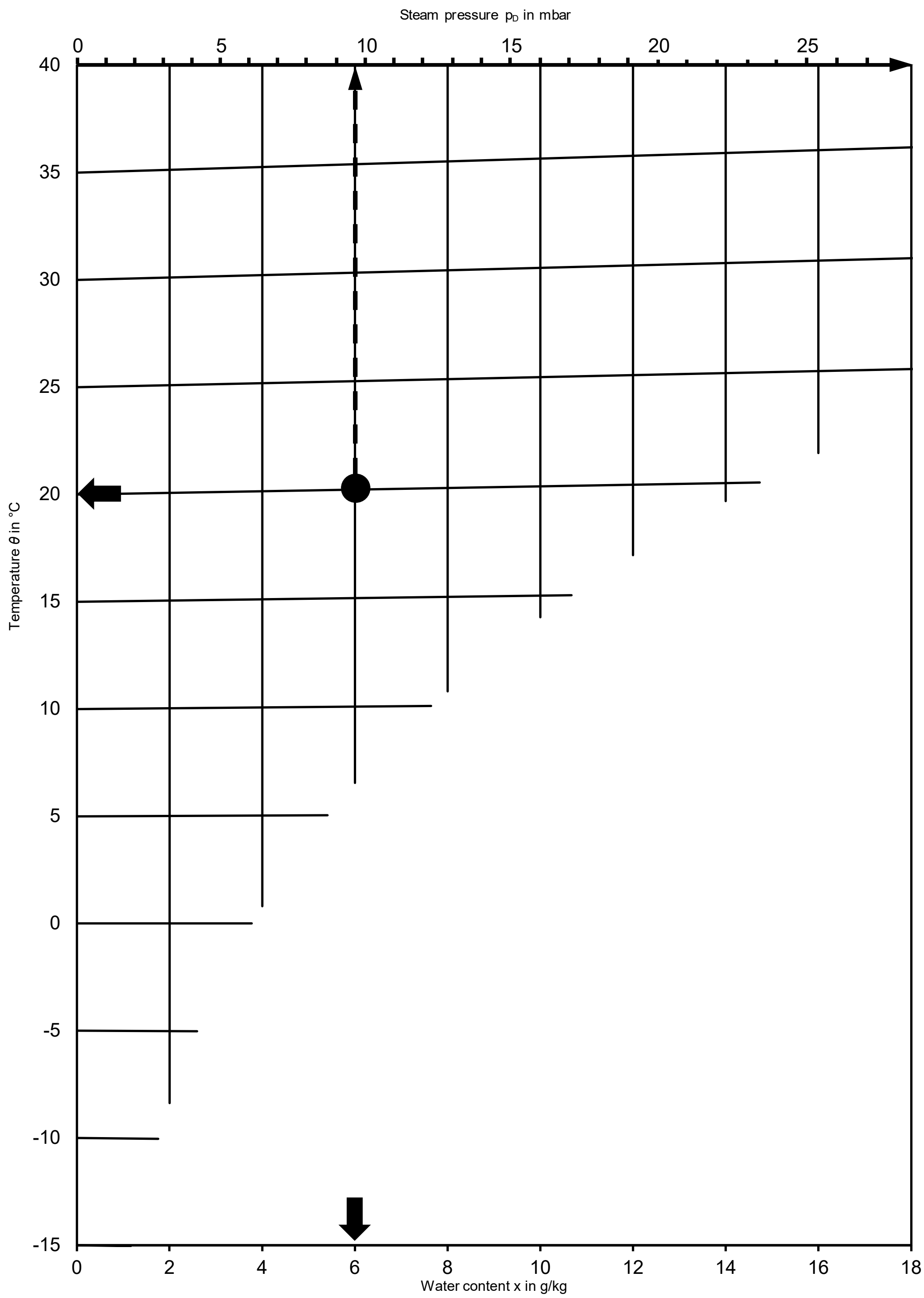


Fig. 2-4 Steam pressure p_D

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 a fog of fine droplets becomes visible. The saturation pressure p_S is dependent on air temperature and pressure (constant in each diagram). Thus, we can enter the saturation pressure p_S for each temperature up to 100 °C. As an example (Fig. 2-5), air at 20 °C (at 1013 mbar) can hold a maximum of 14.7 g/kg water vapor. Connecting saturation pressures at various temperatures with one another in a psychrometric chart creates the saturation line.

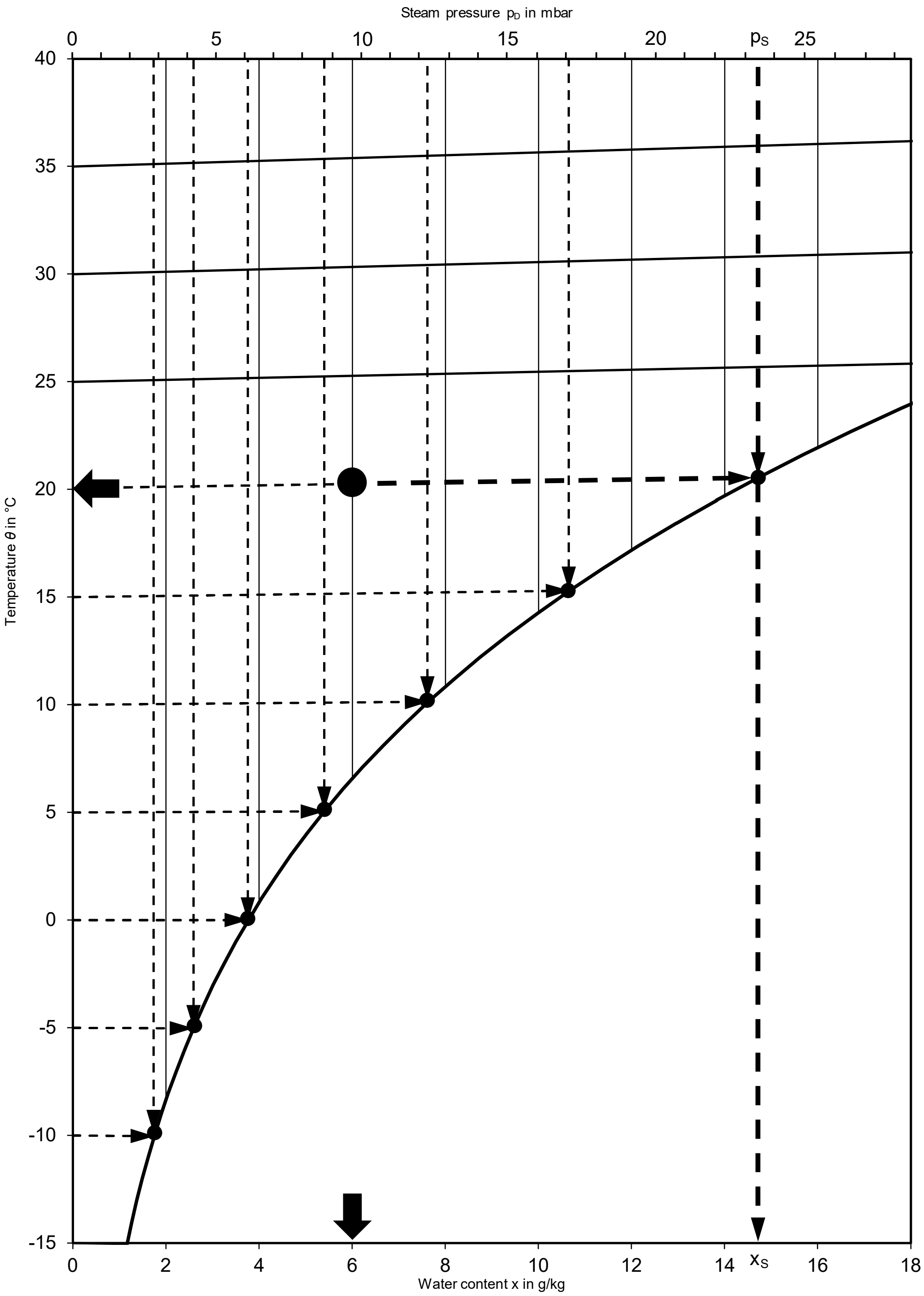


Fig. 2-5 Saturation pressure p_S and absolute humidity x_S

2.2.5 Dew point temperature (saturation temperature) θ_{DP}

Looking at the diagram (Fig. 2-6) it shows that the saturation line cannot be reached simply by increasing the water vapor content x. If we cool air e.g., having an absolute water content of x = 6 g/kg, from 20 °C to + 5 °C, then the saturation line is reached at a temperature of approximately + 6.5 °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, and therefore droplets form. If you wish to dehumidify a water vapor-air mixture, then you have to cool it sufficiently to fall below the dew point temperature. The lower you go below the dew point temperature, the greater the dehumidification effect.

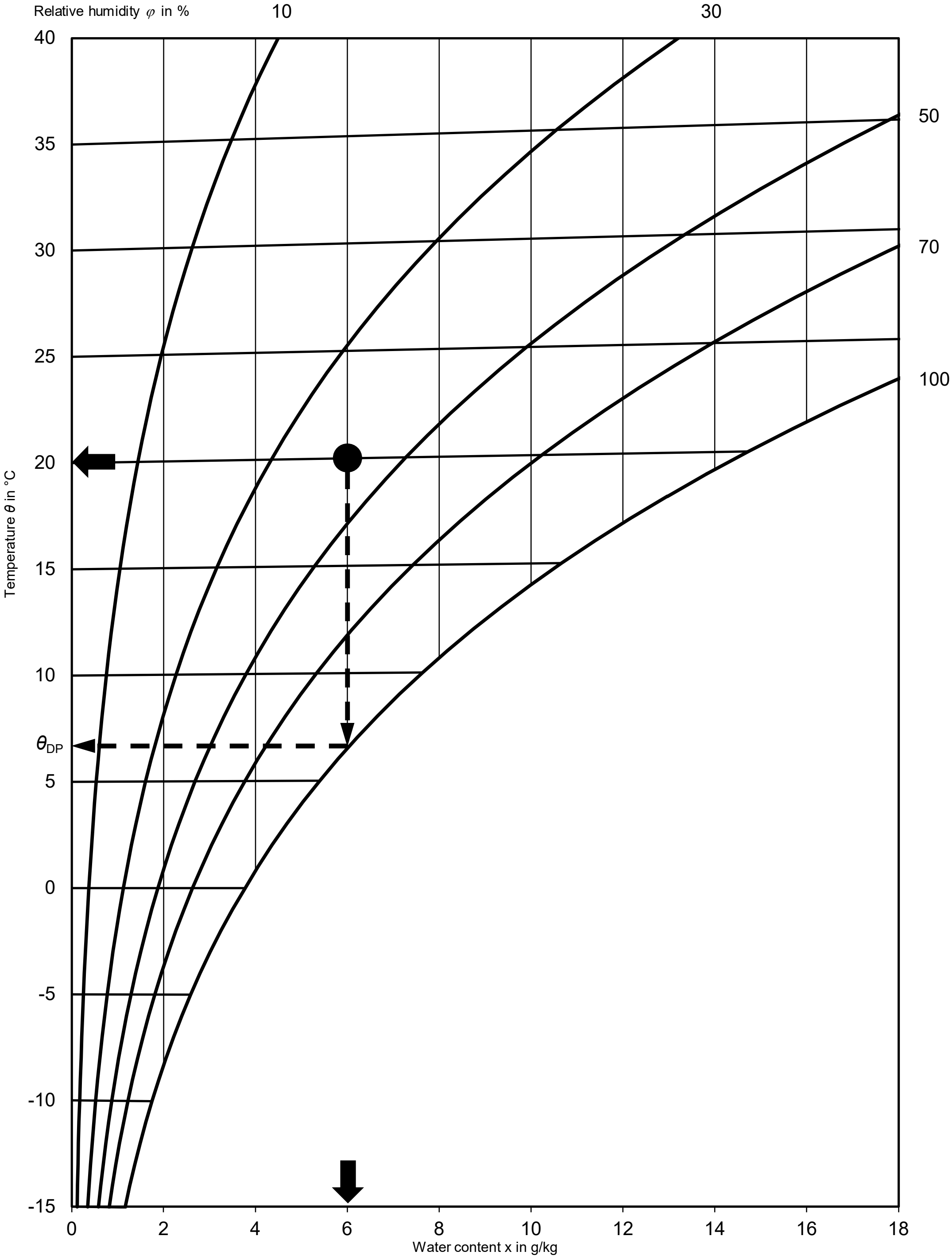


Fig. 2-6 Dew point temperature θ_{DP}

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 (RH). If we now place a point in the psychrometric chart (Fig. 2-7) 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 φ of 50 %.

Example: θ =	17.5 °C	x = 12.4 g/kg	φ = 100 % RH
θ =	17.5 °C	x = 6.2 g/kg	φ = 50 % RH
θ =	7.5 °C	x = 6.4 g/kg	φ = 100 % RH
θ =	7.5 °C	x = 3.2 g/kg	φ = 50 % RH

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 % RH or 70 % RH. This is the method of constructing all lines having a constant relative humidity between 5 and 100 % RH.

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.

2.2.7 Lines having constant enthalpy

The enthalpy h (thermal capacity) 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 θ = 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 and 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 (isenthalpic or adiabatic lines) 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 (Fig. 2-8).

You can now read off from this scale the enthalpy of the air state defined in chapter 2.2 with a temperature of θ = 20 °C and an absolute humidity of x = 6 g/kg: h $\approx$ 35 kJ/kg.

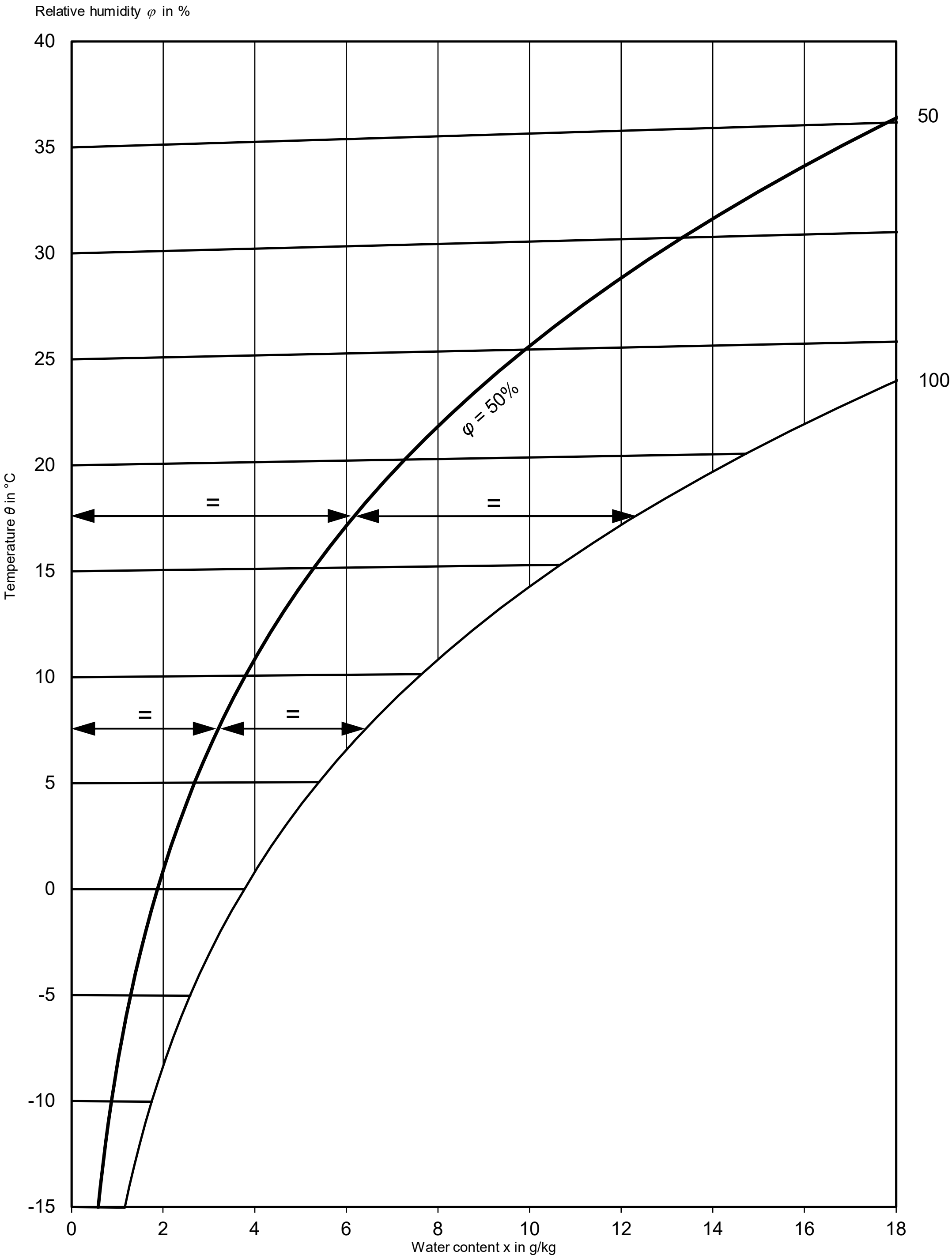


Fig. 2-7 Relative humidity φ

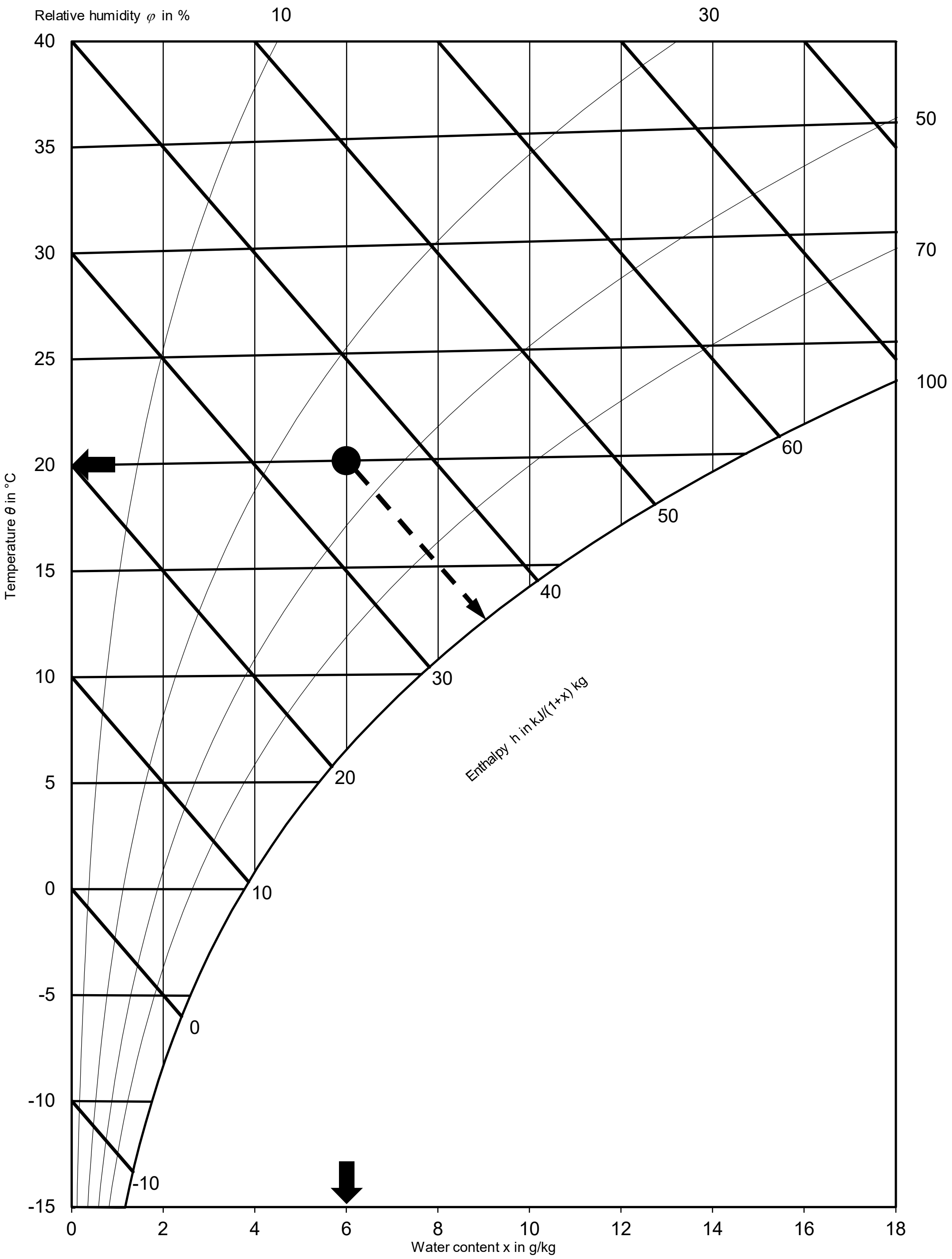


Fig. 2-8 Enthalpy h

2.2.8 Wet-bulb temperature θ_{WB}

Another term used in the thermodynamics of humid air is "wet-bulb temperature" θ_{WB}.
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 isenthalpic lines 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 (adiabatic lines) to the psychrometric chart, we notice that logically they must have the same slope as the isenthalpic lines. 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 adiabatic lines, 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 adiabatic lines relative to the isenthalpic lines.

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 placed 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 around 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 min.) 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 isenthalpic lines 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. 2-9).

For the air state defined in chapter 2.2 with a temperature of θ = 20 °C and an absolute humidity of x = 6 g/kg the result is a wet-bulb temperature of approx. 13 °C. When the air is saturated, the wet-bulb temperature is the same as the dry air temperature.

2.2.9 Density ρ

The density ρ in kg/m^3 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 height 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 downwards to the right.

The density can now be determined for the air state defined in chapter 2.2 with a temperature of θ = 20 °C and an absolute humidity of x = 6 g/kg. The state point lies slightly below the value of ρ = 1.20 kg/m^3 in the diagram (Fig. 2-10).

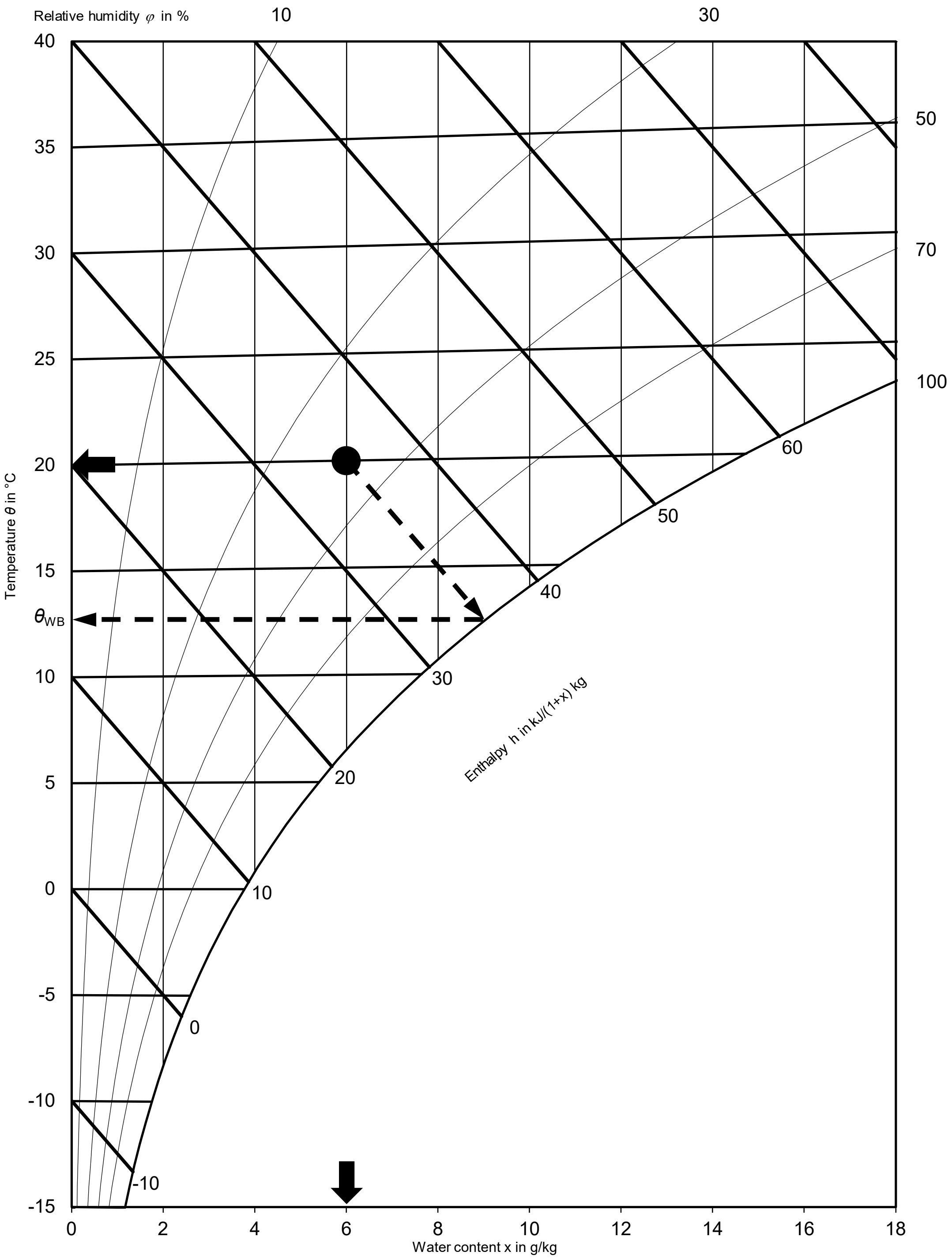


Fig. 2-9 Wet-bulb temperature θ_{WB}

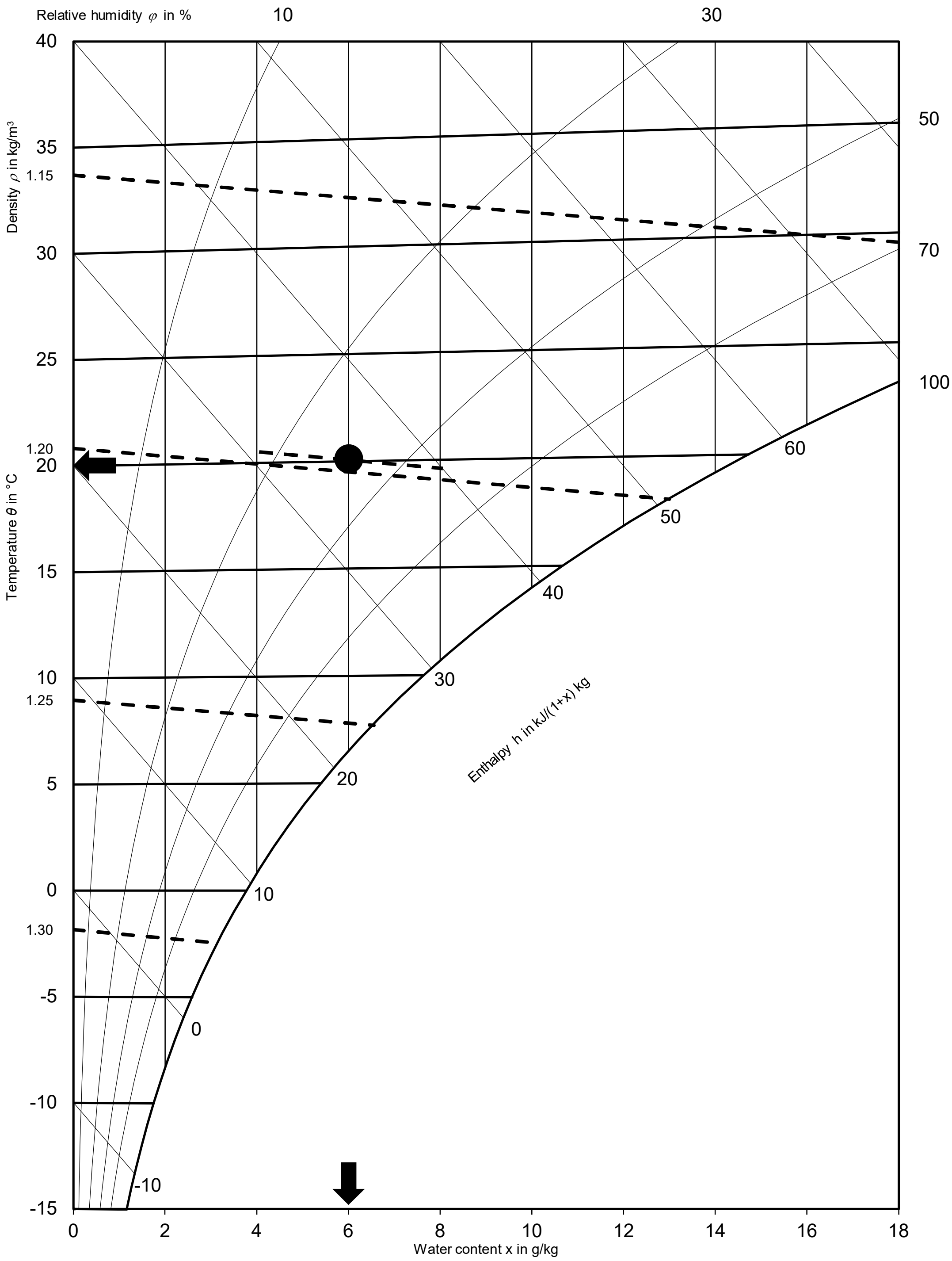


Fig. 2-10 Air density ρ

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:**	**Units:**
1	Air temperature (dry bulb)	θ_A	20	°C
2	Absolute humidity	x	6	g/kg
3	Partial vapor pressure	p_D	9.6	mbar
4	Saturation pressure	p_S	23.3	mbar
5	Absolute humidity at saturation pressure	x_S	14.7	g/kg
6	Dew point temperature	θ_{DP}	6.5	°C
7	Relative humidity	φ	41.4	% RH
8	Enthalpy	h	35.3	kJ/kg
9	Wet bulb temperature	θ_{WB}	13	°C
10	Density	ρ	1.2	kg/m^3

Table 2-1 Ten state variables of an air state

These ten variables are shown in the following psychrometric chart (Fig. 2-11):

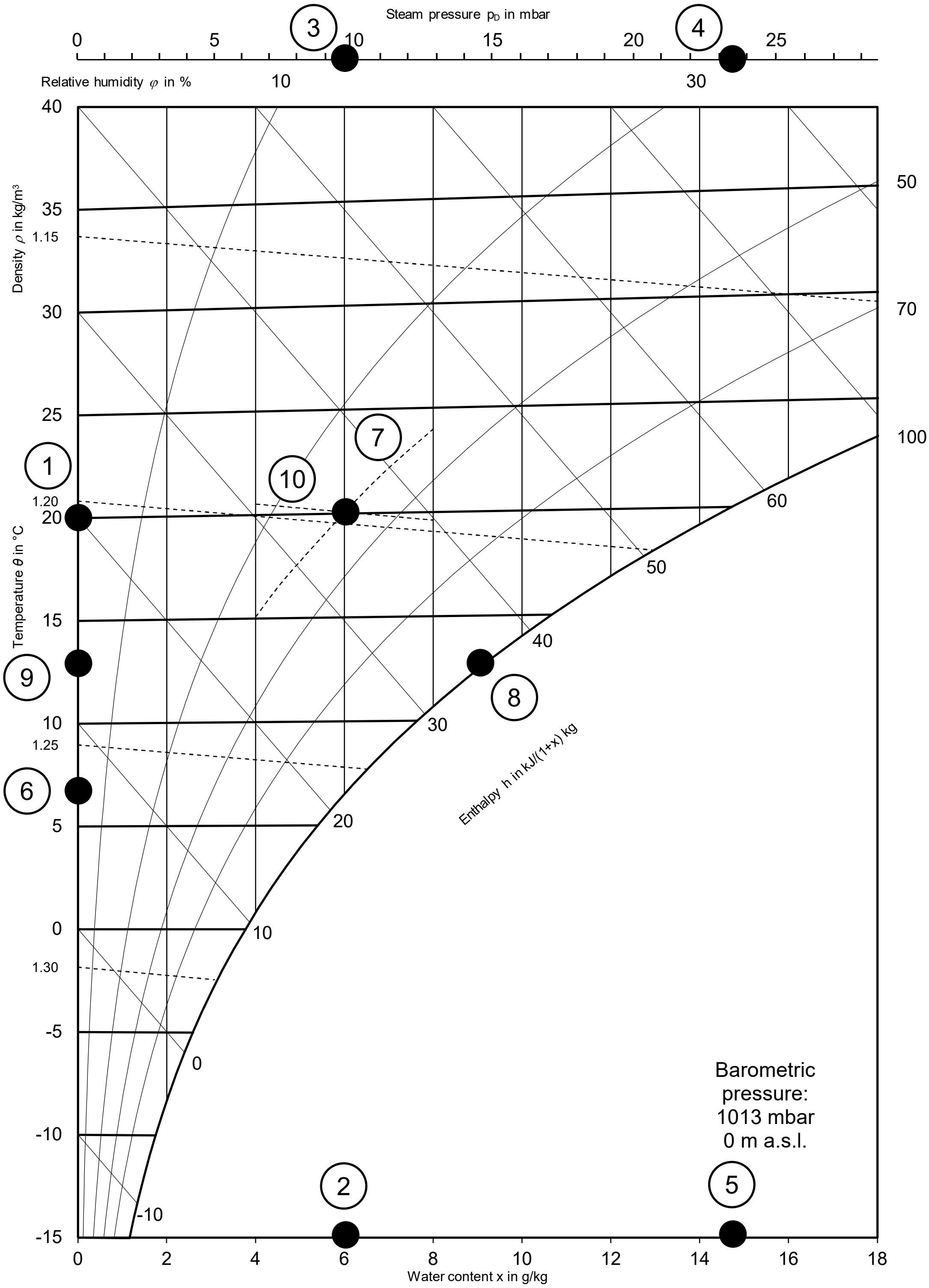


Fig. 2-11 Ten state variables of an air state in the psychrometric chart

3 Application of the psychrometric chart

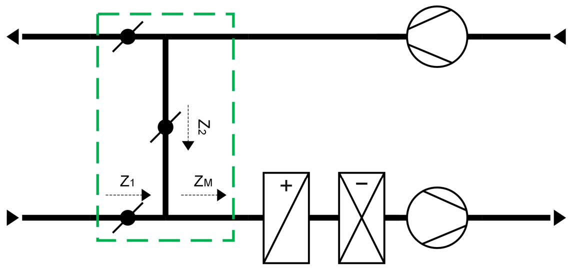
3.1 General

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 1013 mbar (= 760 mmHg).

3.2 Mixing two quantities of air



If two air flows, q_{m1} and q_{m2} [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 (Fig. 3-1). 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:

$$\frac{q_{m1}}{q_{m2}} = \frac{L_2}{L_1} \text{ or } \frac{q_{m1}}{q_{m2}} = \frac{L_{1,2} - L_1}{L_1}$$

Substituting and solving for L_1:

$$L_1 = \frac{L_{1,2} \cdot q_{m2}}{q_{m1} + q_{m2}}$$

In

Fig. 3-1, in the mixing chamber of a ventilation system, q_{m1} = 1,500 kg/h outside air at state Z_1 (θ_1 = 5 °C and φ_1 = 50 % RH) is mixed with q_{m2} = 2,500 kg/h recirculating air of state Z_2 (θ_2 = 20 °C and φ_2 = 60 % RH). 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 straight 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 64 mm.

Consequently, the length of L_1 is:

$$L_1 = \frac{L_{1,2} \cdot q_{m2}}{q_{m1} + q_{m2}} = \frac{64 \text{ mm} \cdot 2500 \frac{kg}{h}}{(1500 + 2500) \frac{kg}{h}} = 40 \text{ mm}$$

The length L_1 extended from point Z_1 locates the position of mixing point Z_M with

θ_M = 14.4 °C, x_M = 6.5 g/kg, φ_M = 64 % RH.

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

$$\theta_M = \frac{\theta_1 \cdot q_{m1} + \theta_2 \cdot q_{m2}}{q_{m1} + q_{m2}} = \frac{5\ °C \cdot 1500\ \frac{kg}{h} + 20\ °C \cdot 2500\ \frac{kg}{h}}{(1500 + 2500)\ \frac{kg}{h}} = 14.4\ °C$$

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

$$x_M = \frac{x_1 \cdot q_{m1} + x_2 \cdot q_{m2}}{q_{m1} + q_{m2}} = \frac{2.7\ \frac{g}{kg} \cdot 1500\ \frac{kg}{h} + 8.8\ \frac{g}{kg} \cdot 2500\ \frac{kg}{h}}{(1500 + 2500)\ \frac{kg}{h}} = 6.5\ \frac{g}{kg}$$

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 recirculating air, then the mixing point could fall below the saturation line, that is, in the mist area (occurs, e.g., for inside swimming pools).

The formation of mist in a mixing chamber of an air handling unit is generally not problematic. This is because the air is heated in a heater 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 (Fig. 3-2) 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 (isenthalpic or adiabatic lines).

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 Δx (e.g., by absorption), then the air state moves in the direction of the mist isotherms and reaches the saturation point at point B.

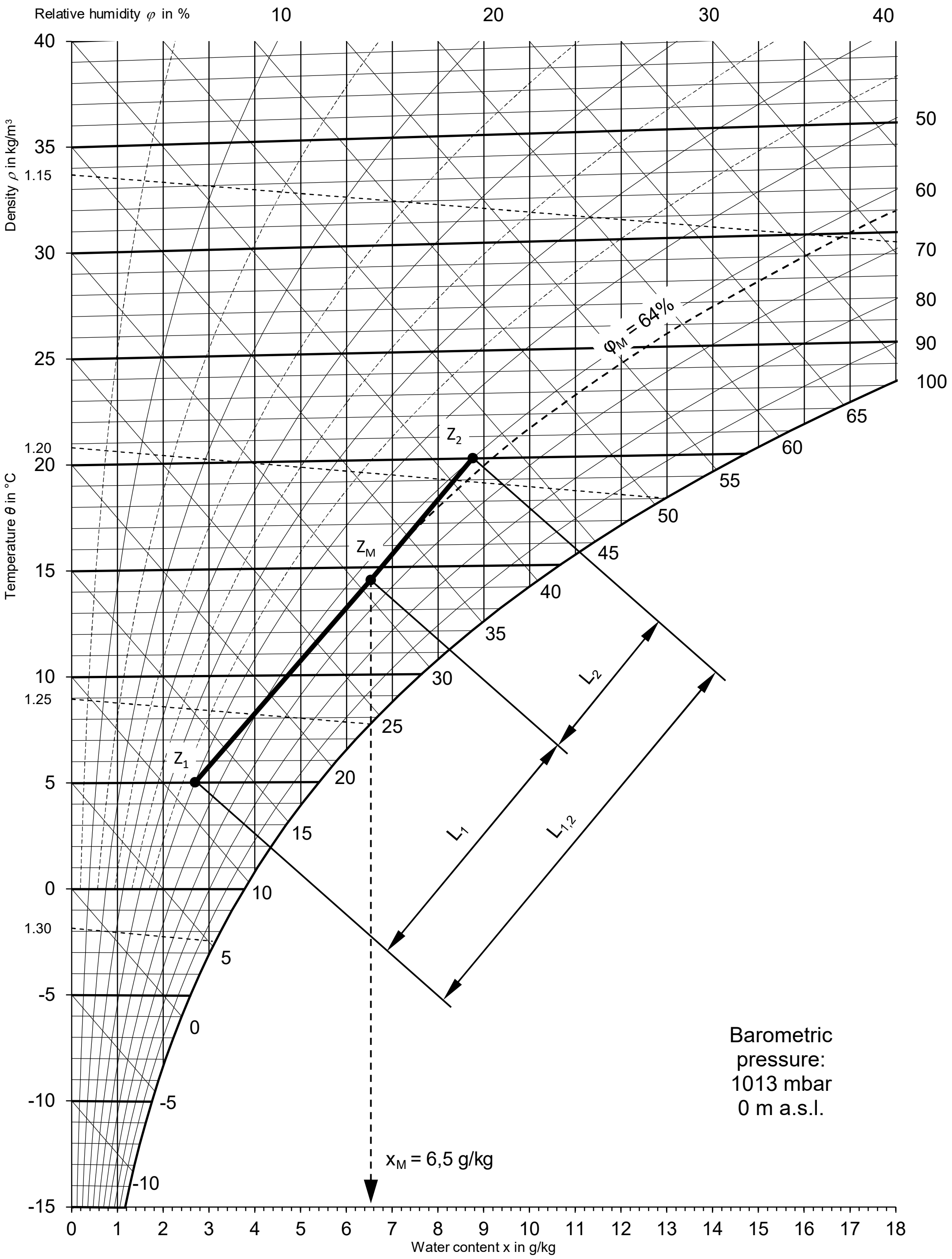


Fig. 3-1 Mixing of air flows Z_1 and Z_2

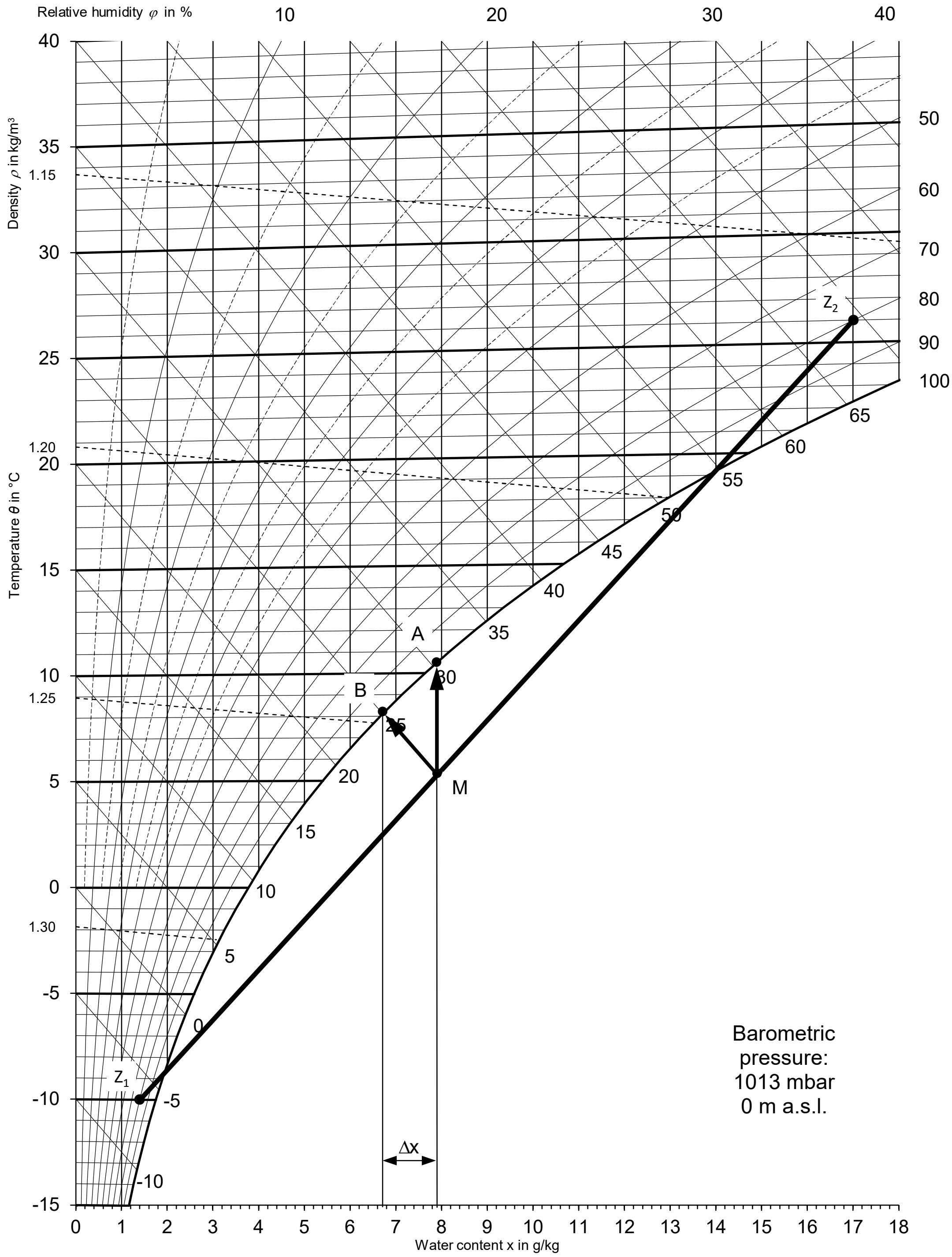
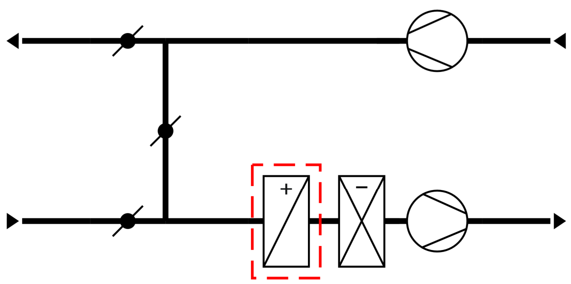


Fig. 3-2 Air mixture with mixing point in mist region

3.3 Heating the air



Heating 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 in kJ/kg has to be added to the air, where:

$$\Delta h = (h_2 - h_1)\ \frac{kJ}{kg}$$

Fig. 3-3 shows a heating process where air, having a mass flow rate q_{mA} = 1 kg/s (= 3600 kg/h), at θ_1 = +5 °C and φ_1 = 60 % RH, is heated to θ_2 = 25 °C.

The required thermal output of the air heating coil is

$\Phi_{AH} = q_{mA} \cdot \Delta h$ with h_1 = 13 kJ/kg and h_2 = 33.3 kJ/kg.

This yields:

$$\Phi_{AH} = q_{mA} \cdot (h_2 - h_1) = 1\ \frac{kg}{s} \cdot (33.3 - 13)\ \frac{kJ}{kg} = 20.3\ \frac{kJ}{s} = 20.3\ kW$$

The calculated thermal output of Φ_{AH} = 20.3 kW can be supplied to the air heating coil with e.g., heated water or steam.

If heated water is used, we assume, at full load, that the water is cooled by $\Delta\theta_W$ = 20 K (e.g., supply temperature θ_{ST} = 60 °C, return temperature θ_{RT} = 40 °C). The water flow q_{mW} can be calculated from:

$$q_{mw} = \frac{\Phi_{AH}}{c_W \cdot \Delta\,\theta_W} = \frac{20.3\ \frac{kJ}{s}}{4.19\ \frac{kJ}{kg \cdot K} \cdot 20\ K} = 0.24\ \frac{kg}{s}$$

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 only take into account the heat of vaporization, then the amount of steam required is:

$$q_{ms} = \frac{\Phi_{AH}}{r} = \frac{21.0\ \frac{kJ}{s}}{2258\ \frac{kJ}{kg}} = 0.009\ \frac{kg}{s}$$

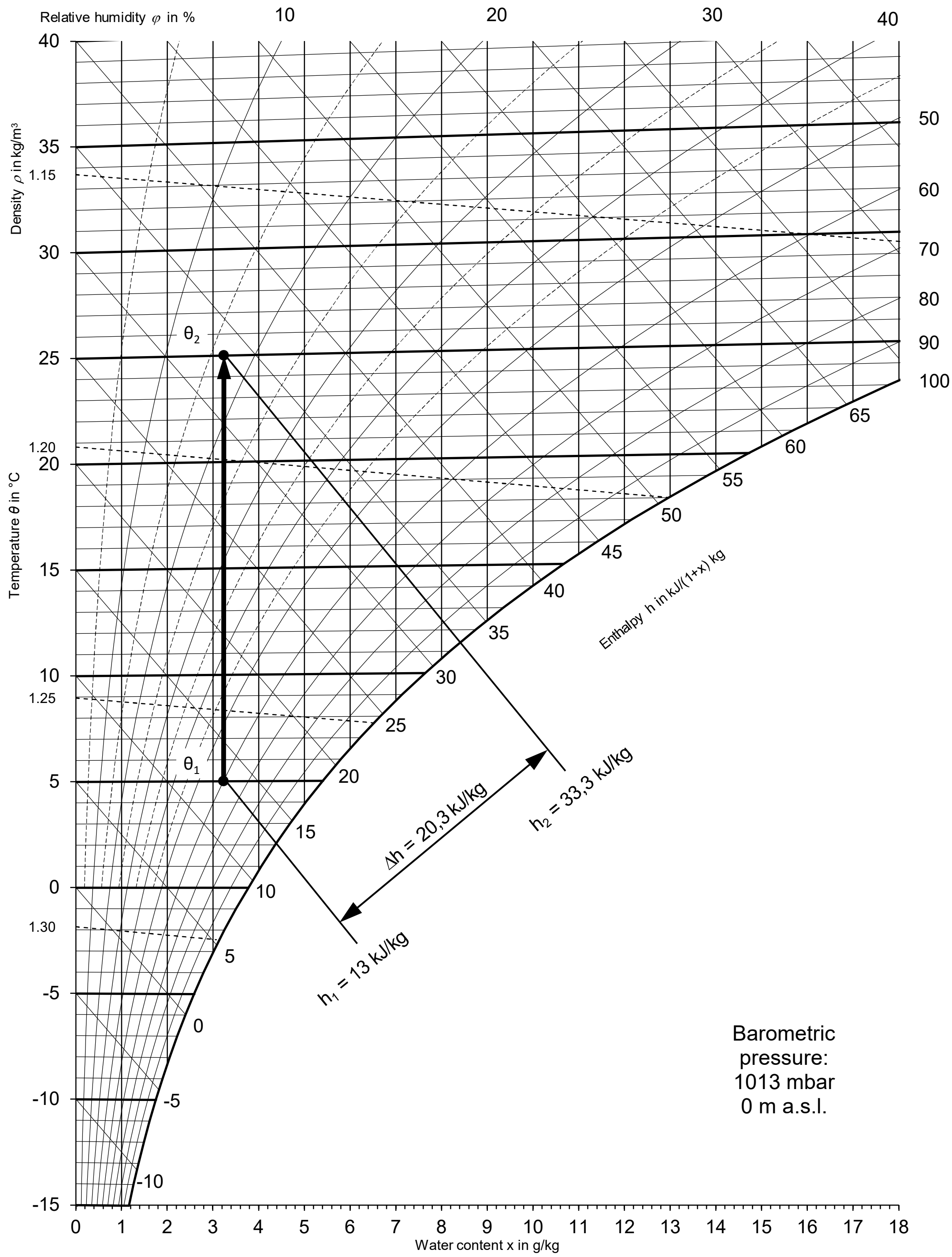
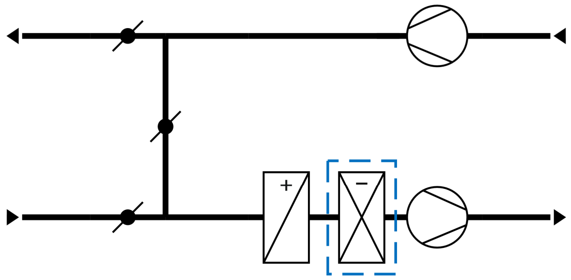


Fig. 3-3 Heating the air

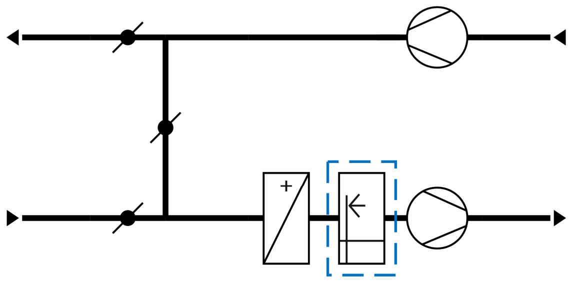
3.4 Cooling the air

There are two possibilities for cooling the air:

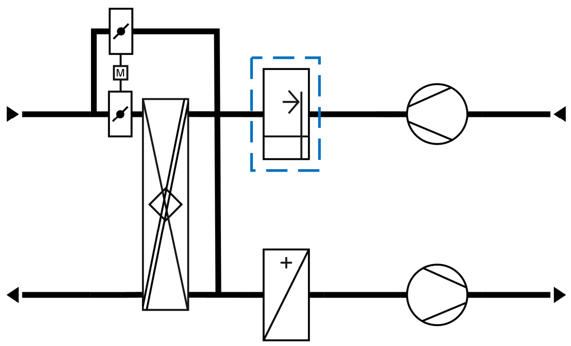
- Bringing the air in contact with a colder surface (surface cooling) see 3.5



- Spraying water into the supply air flow (evaporative cooling) see 3.6



- Spraying water into the return air flow (evaporative cooling) see 3.6



3.5 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, $\theta_{CO} > \theta_{DP}$).
- The cooling surface temperature lies below the dew point temperature of the air to be cooled (wet cooling surface, $\theta_{CO} < \theta_{DP}$).

Cooling without water condensation

When the temperature of the cooling surface θ_{CO} is higher than the dew point temperature θ_{DP} 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. The state change in the psychrometric chart (Fig. 3-4) 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 (see 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:

$$\Delta h = (h_1 - h_2)\ \frac{kJ}{kg}$$

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 supply and return water temperature of the cooling water:

$$\theta_{CO} = \frac{\theta_{ST} + \theta_{RT}}{2} + 1\ldots2\ K$$

In Fig. 3-4 the mass flow q_{mA} = 1 kg/s (= 3600 kg/h) of air is cooled to θ_2 = 20 °C from θ_1 = 29 °C and φ_1 = 40% RH using a surface cooler having an average cooling surface temperature of θ_{CO} = 18 °C. The cooling water in turn heats up from θ_{ST} = 15 °C to θ_{RT} = 19 °C. A quantity of heat of:

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

The cooling output needed for this is:

$$\Phi_{CO} = q_{mA} \cdot \Delta h = 1\ \frac{kg}{s} \cdot 9.5\ \frac{kJ}{kg} = 9.5\ \frac{kJ}{s} = 9.5\ kW$$

The mass flow of water is calculated in the same manner as for heating air except that there is a much smaller temperature difference $\Delta\theta$ available with cooling.

$$q_{mW} = \frac{\Phi_{CO}}{c_W \cdot \Delta\theta_W} = \frac{9.5\ \frac{kJ}{s}}{4.19\ \frac{kJ}{kg \cdot K} \cdot (19 - 15)\ K} = 0.57\ \frac{kg}{s}$$

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.

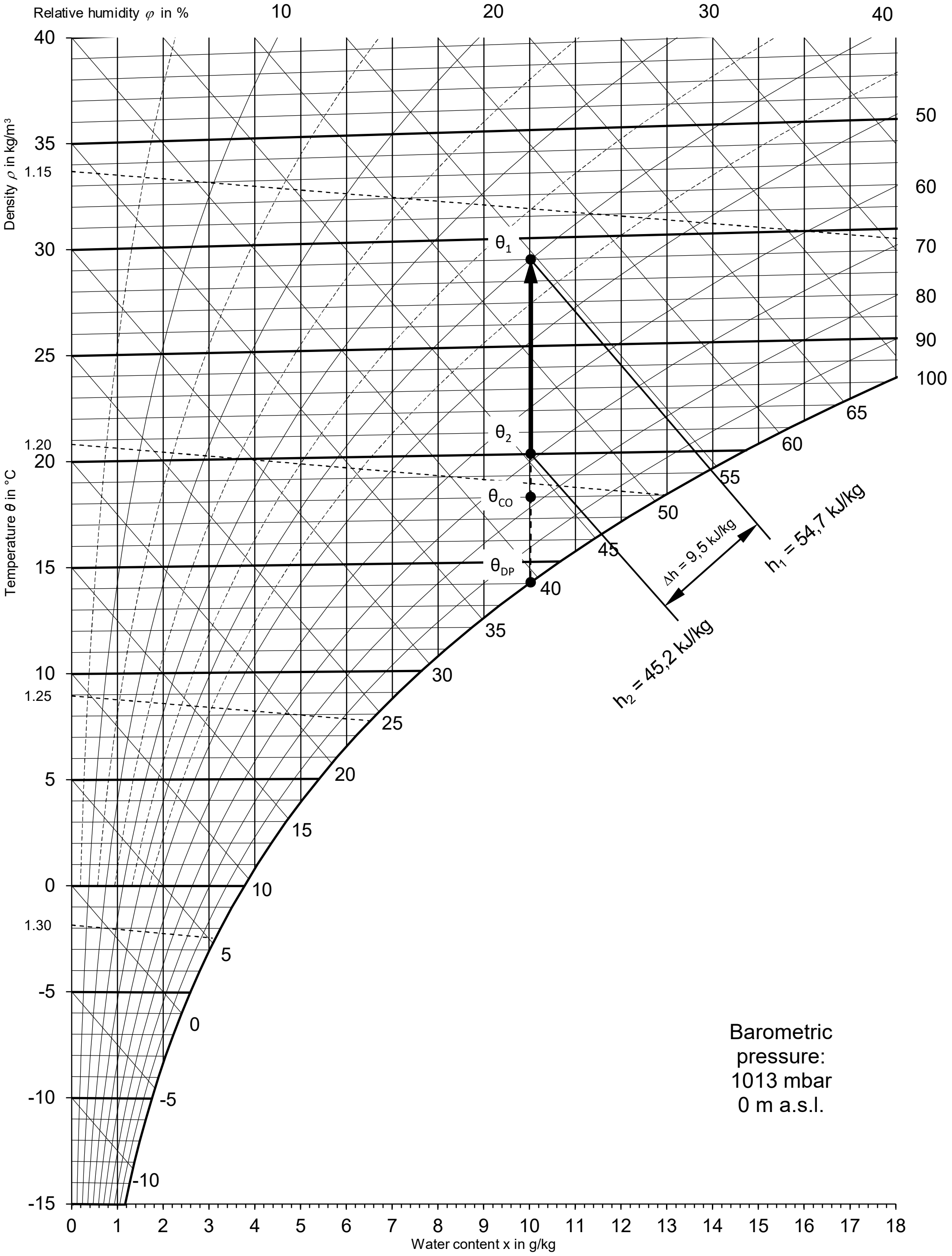


Fig. 3-4 Air cooling using a dry cooling surface

Cooling with water condensation

If the temperature of the cooling surface θ_{CO} is under the dew point θ_{DP} 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 (see section: Bypass effect).

The state change in the psychrometric chart (Fig. 3-5) 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 q_{mA} = 1 kg/s (= 3600 kg/h) air is cooled to θ_2 = 20 °C from θ_1 = 29 °C and φ_1 = 40 % RH, using a surface cooler having an average cooling surface temperature of θ_{CO} = 10 °C. At the same time, the cooling water heats up from θ_{ST} = 6 °C to θ_{RT} = 12 °C. The quantity of heat removed from one kg air thus must be:

$$\Delta h = h_1 - h_2 = 54.7 - 42.2 = 12.5\ \frac{kJ}{kg}$$

The cooling output required for this is:

$$\Phi_{CO} = q_{mA} \cdot \Delta h = 1\ \frac{kg}{s} \cdot 12.5\ \frac{kJ}{kg} = 12.5\ \frac{kJ}{s} = 12.5\ kW$$

The mass flow of water is calculated in the same manner as for heating the air.
The available temperature difference $\Delta\theta$ is, however, much smaller.

$$q_{mW} = \frac{\Phi_{CO}}{c_W \cdot \Delta\theta_W} = \frac{12.5\ \frac{kJ}{s}}{4.19\ \frac{kJ}{kg \cdot K} \cdot (12 - 6)\ K} = 0.5\ \frac{kg}{s}$$

The volume of condensed water per kg of air is:

$$\Delta x = x_1 - x_2 = 10.0 - 8.9 = 1.1\ \frac{g}{kg}$$

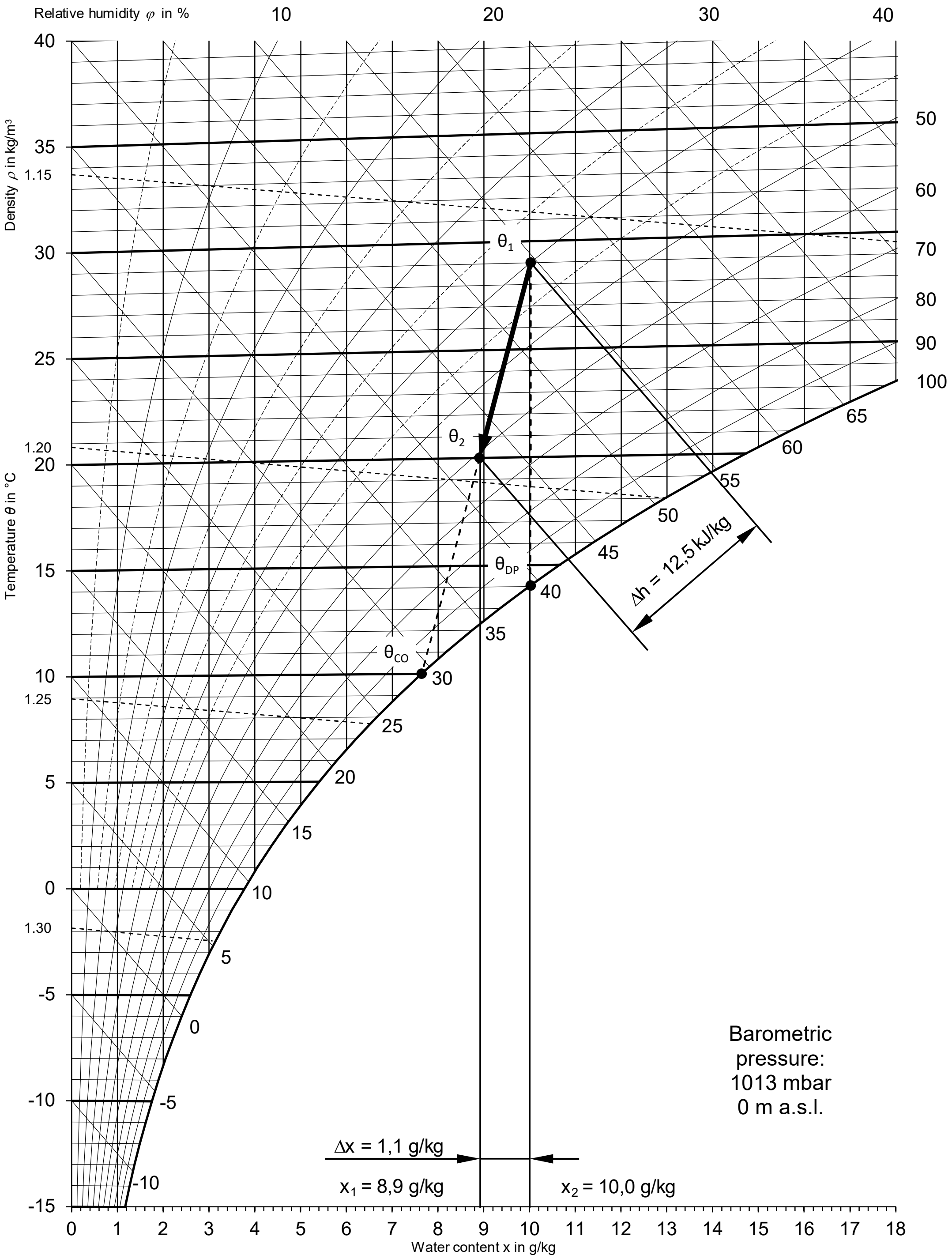


Fig. 3-5 Cooling with water condensation Δx in g/kg

As the examples (Fig. 3-4 and Fig. 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 $\Delta\Phi_{CO}$, the heat of vaporization of the extracted water is conducted away, thus:

$$\Delta\Phi_{CO} = r \cdot \Delta x = 2450\ \frac{kJ}{kg} \cdot 0.0011\ \frac{kg}{kg} = 2.7\ \frac{kJ}{kg}$$

When cooling by 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_{CO} (Fig. 3-6). The bend in the curve is, among other things, co-determined by the hydronic circuit applied.

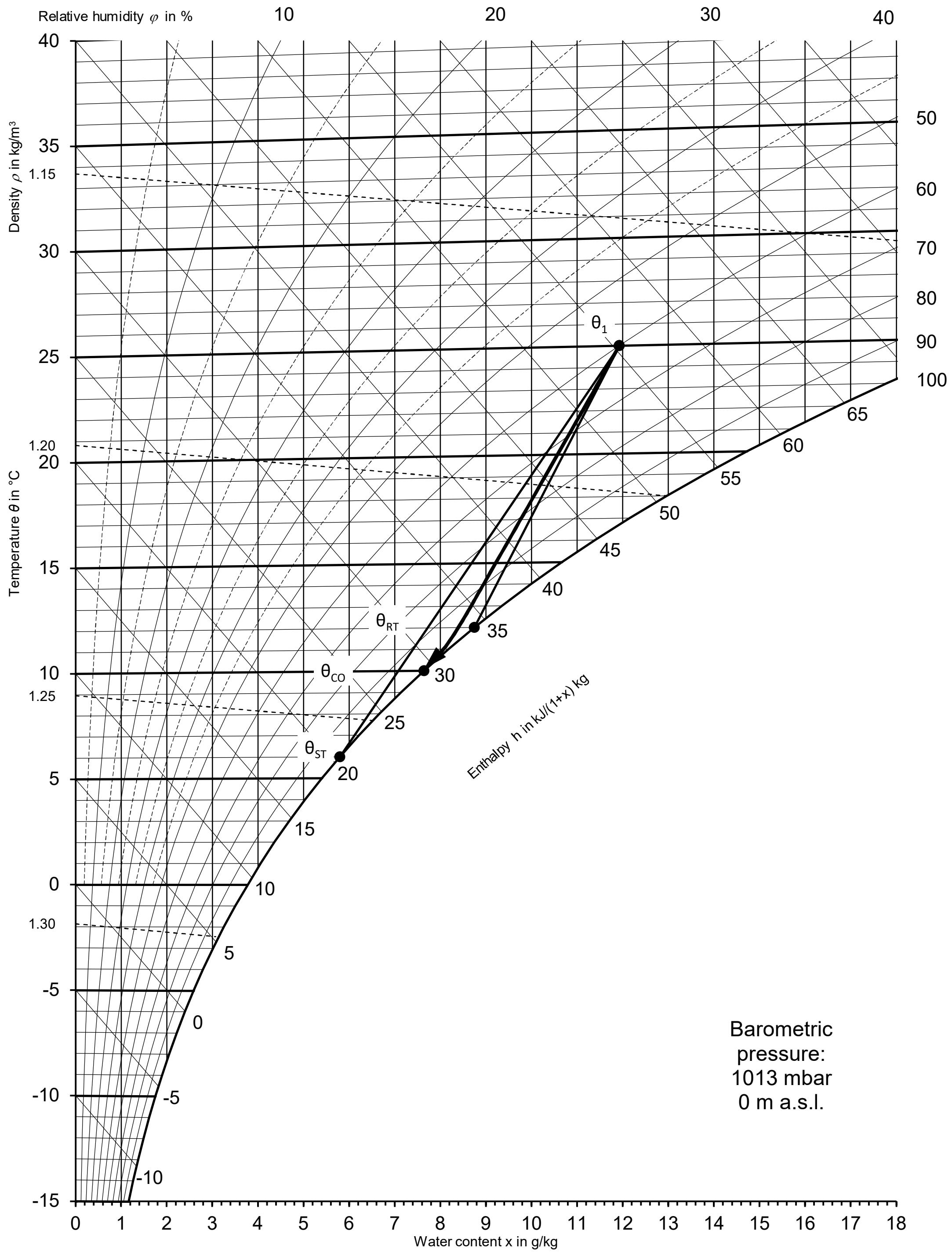


Fig. 3-6 Actual cooling process for a wet cooling surface

3.6 Evaporative Cooling

In the process of cooling the air using the evaporative cooling method, water is being sprayed into the jet chamber. Part of this water is being absorbed by the air and therefore results in saturating and cooling the air. For further information on this topic refer to 3.9 “Air humidification”. In Fig. 3-7 the mass q_{mA} = 1 kg/s (= 3600 kg/h) air at θ_1 = 29 °C and φ_1 = 40 % RH is cooled down adiabatically to θ_2 = 20 °C using an evaporative cooler (refers to schematic in 3.4). The air reaches the desired temperature, but has a relative humidity of over 90%. Most often the air cannot be supplied to the room in this air state because of comfort reasons. The common process of cooling the air is shown in Fig. 3-8, where the return air is being cooled and then further used in a heat recovery system.

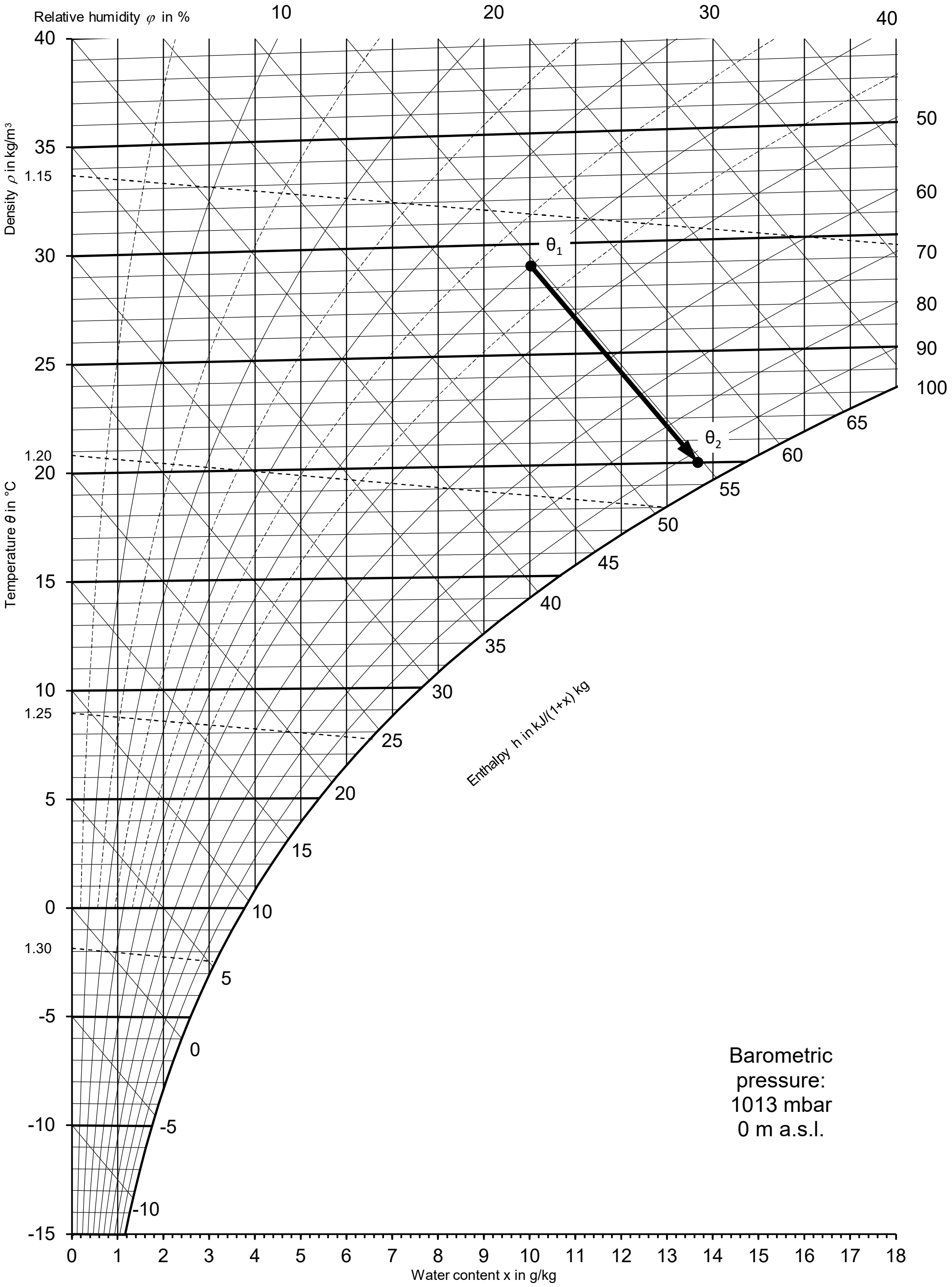


Fig. 3-7 Cooling of the supply air using the evaporative method

In Fig. 3-8 the process of cooling the return air using the evaporative method, as well as the following heat transfer of sensible heat energy using the heat recovery system is shown (refer to schematic in 3.4). The heat recovery system has a maximum thermal efficiency of η_{th} = 85 %.

In this example the return air at θ_3 = 25°C and φ_3 = 50 % RH is cooled down to θ_4 = +18 °C using an evaporative cooling system. This cooled return air cools down the outside air of θ_1 = 29 °C and φ_1 = 35 % RH to the desired temperature of θ_2 = 20 °C using a plate heat exchanger. At the same time the return air gets heated to θ_5 = 27 °C.

For further information refer to 3.8 “Heat recovery”.

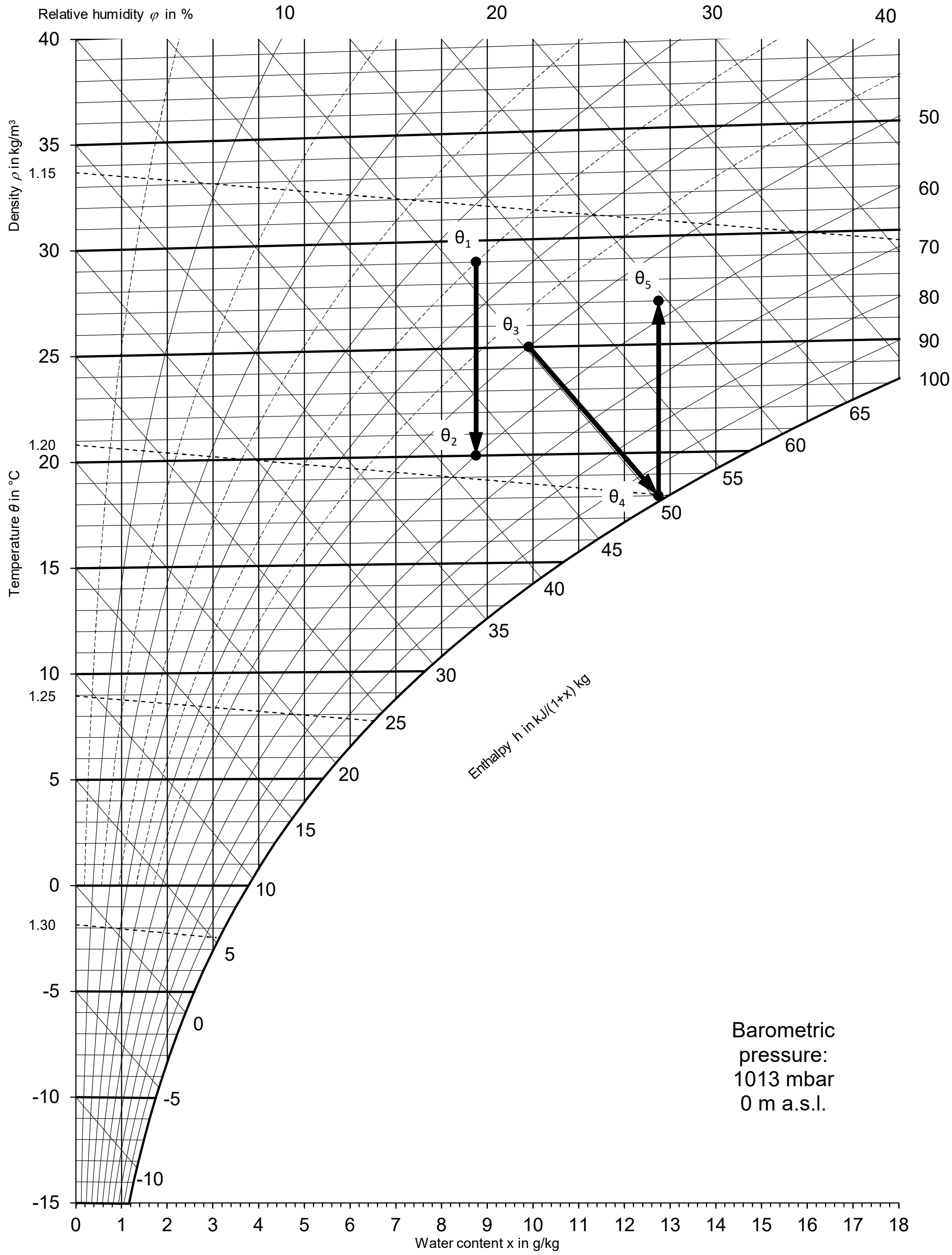


Fig. 3-8 Cooling of the return air using the evaporative method

3.7 Cooling output

In air-conditioning systems, state changes (e.g., cooling and humidifying) are normally not 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 θ_1 = 25 °C and x_1 = 9 g/kg is cooled and dehumidified to θ_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 heating factor for this example as follows:

$$\mathrm{SHF} = \frac{Q_{sens}}{Q_{total}} = \frac{\Delta h_{sens}}{\Delta h_{total}} = \frac{(48.5 - 38.5)\ \frac{kJ}{kg}}{(48.5 - 35.0)\ \frac{kJ}{kg}} = 0.74$$

whereas ($\Delta h_{total} = \Delta h_{sens} + \Delta h_{lat}$)

If the psychrometric chart (Fig. 3-9) has a scale of Q_{sens} / Q_{total} with an associated pivot point (e.g. θ = 21 °C, x = 8 g/kg), the process of the state change can be determined graphically by a parallel displacement (SHF = 0.74). 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.

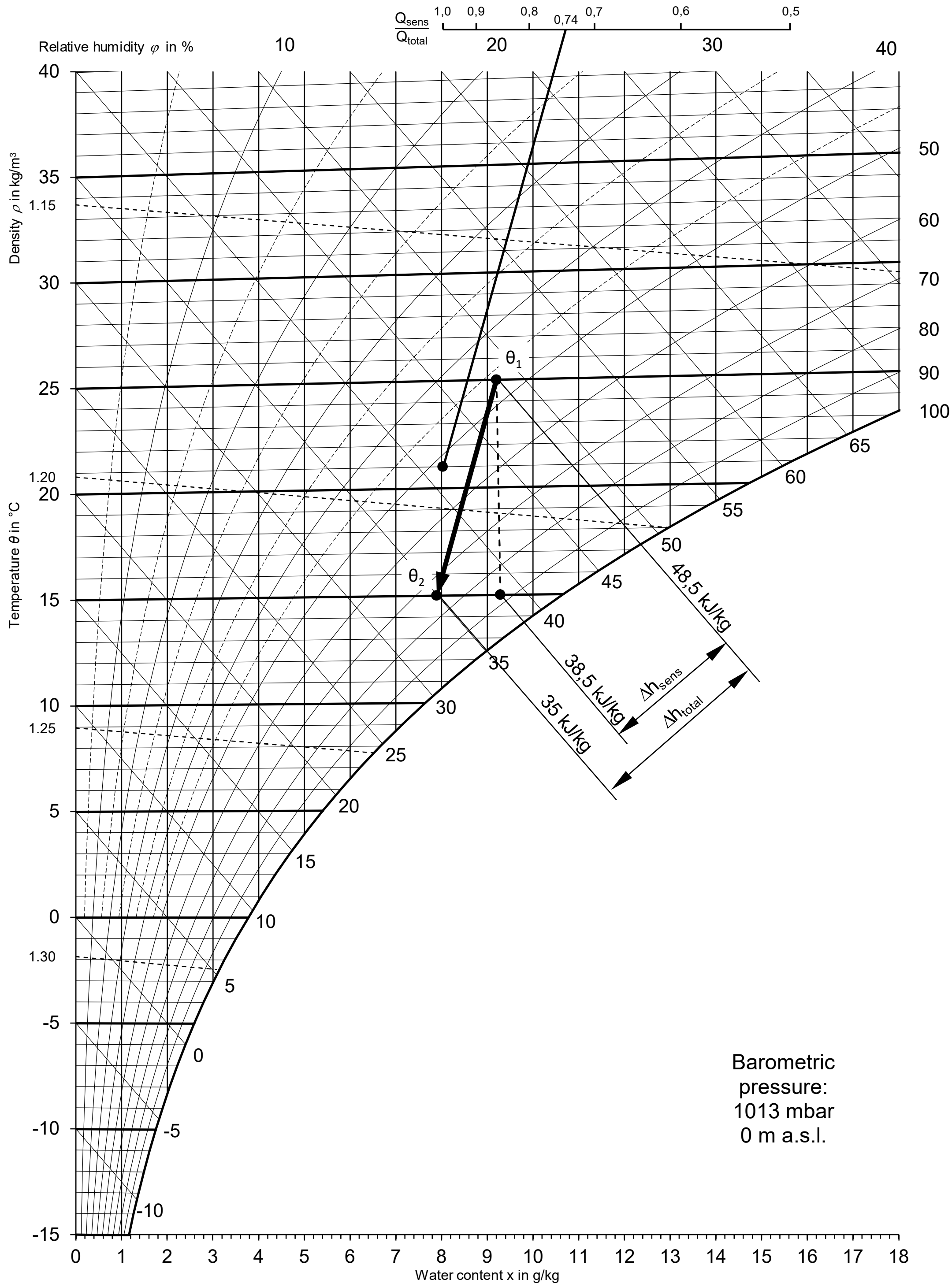


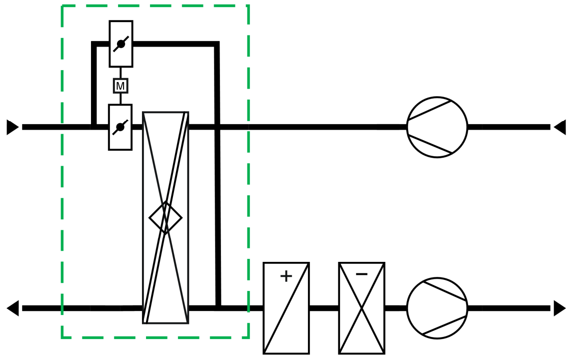
Fig. 3-9 Cooling process with help of the SENSIBLE HEAT FACTOR (SHF)

3.8 Heat recovery

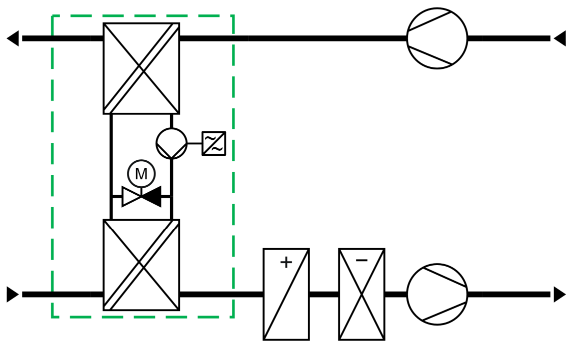
During winter operation the heat recovery system takes the heat energy out of the return air before it leaves the system as exhaust air. This heat energy can then be transferred through the heat recovery system to the (cold) outside air. Through this process a substantial amount of primary energy, which would have been used to heat the outside air, can be saved.

There are different systems to recover heat. Following are the three most commonly used heat recovery systems:

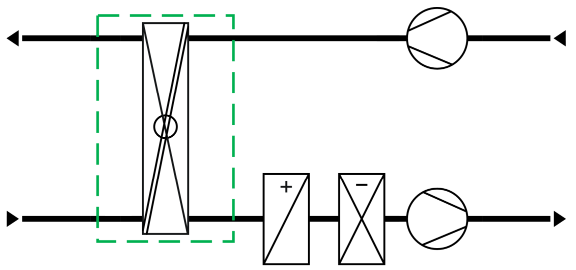
- Transfer of sensible heat energy using a plate heat exchanger



- Transfer of sensible heat energy using a cycle compound system



- Transfer of sensible heat energy or of sensible and latent heat energy using a rotary heat exchanger



Transfer of sensible heat energy

In example (Fig. 3-10) the transfer of sensible heat energy is shown. The example shows a heat transfer with a thermal efficiency of η_{th} = 55 %.In Fig. 3-10 the mass outside air q_{mA} = 1 kg/s (= 3600 kg/h) air at θ_1 = +2 °C and φ_1 = 60 % RH is heated to θ_2 = +13 °C. At the same time the return air at θ_3 = +22 °C and φ_3 = 40 % RH is cooled down to θ_4 = +11 °C. The conditions shown below are true, if during winter operation the mass of outside and exhaust air are the same and if there is no condensation in the return air.

The example (Fig. 3-10) can be used for the plate heat exchange system, as well as for the cycle compound system or the rotary heat exchange system if sensible heat is exchanged.

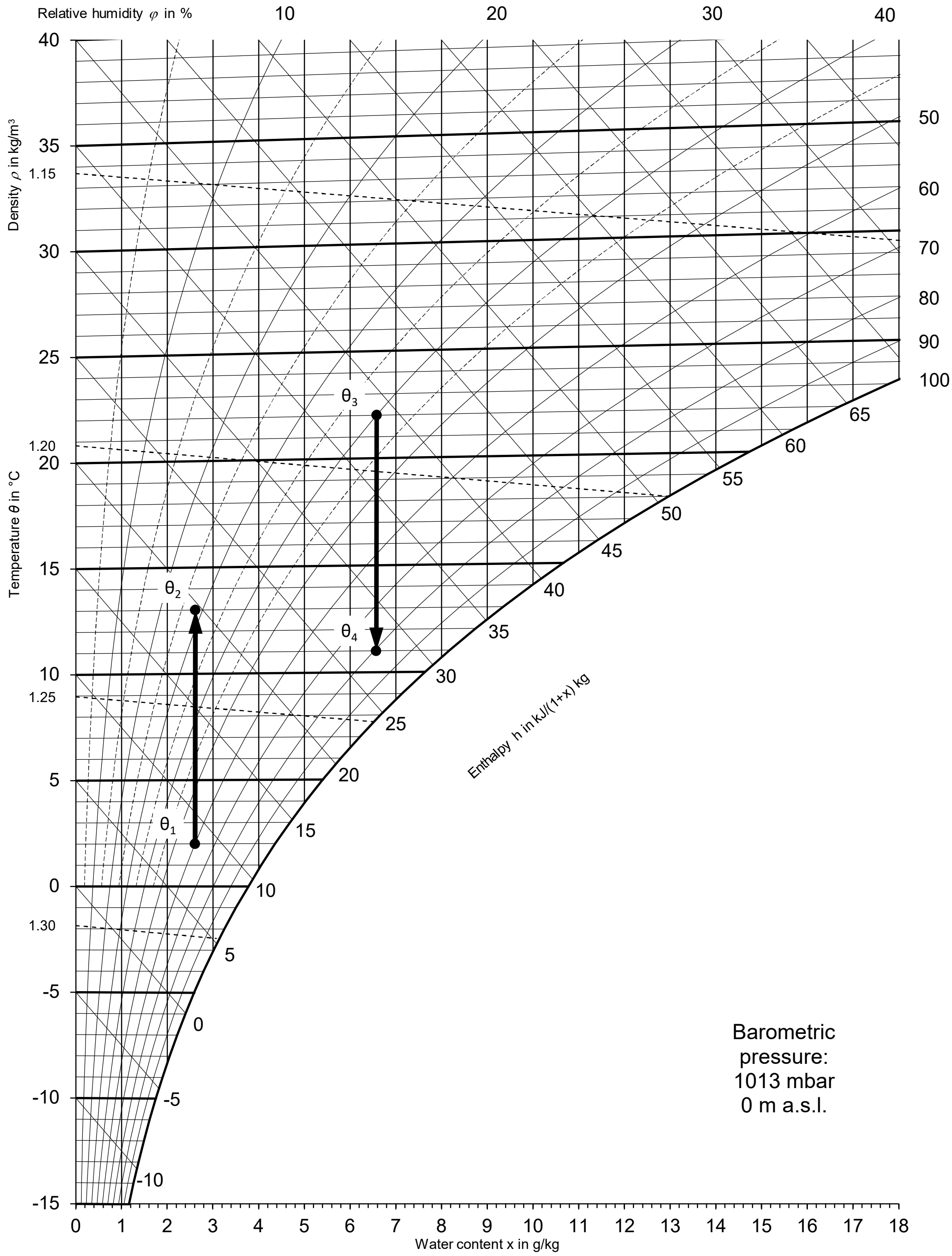


Fig. 3-10 Heat recovery for sensible heat energy

Transfer of sensible and latent heat energy

The example (Fig. 3-11) can be used if the rotary heat exchanger transfers sensible and latent heat (humidity). In this example the transfer of sensible and latent heat is shown. The heat exchange system in this example has a thermal efficiency of η_{th} = 75 % and a humidity efficiency of η_h = 70 %.

In Fig. 3-11 the mass outside air q_{mA} = 1 kg/s (= 3600 kg/h) air at θ_1 = +2 °C and φ_1 = 60 % RH and x_1 = 2.6 g/kg is heated to θ_2 = +17 °C, φ_2 = 59.2 % RH and x_2 = 7.1 g/kg. At the same time, the return air at θ_3 = +22 °C and φ_3 = 55 % RH and x_3 = 9.0 g/kg is cooled down and dehumidified to θ_4 = +7 °C, φ_4 = 73.4 % RH and x_4 = 4.5 g/kg. In this cooling process the condensed water per kg air is 4.5 g/kg. The conditions shown below are true, if during winter operation the mass of outside and exhaust air are the same.

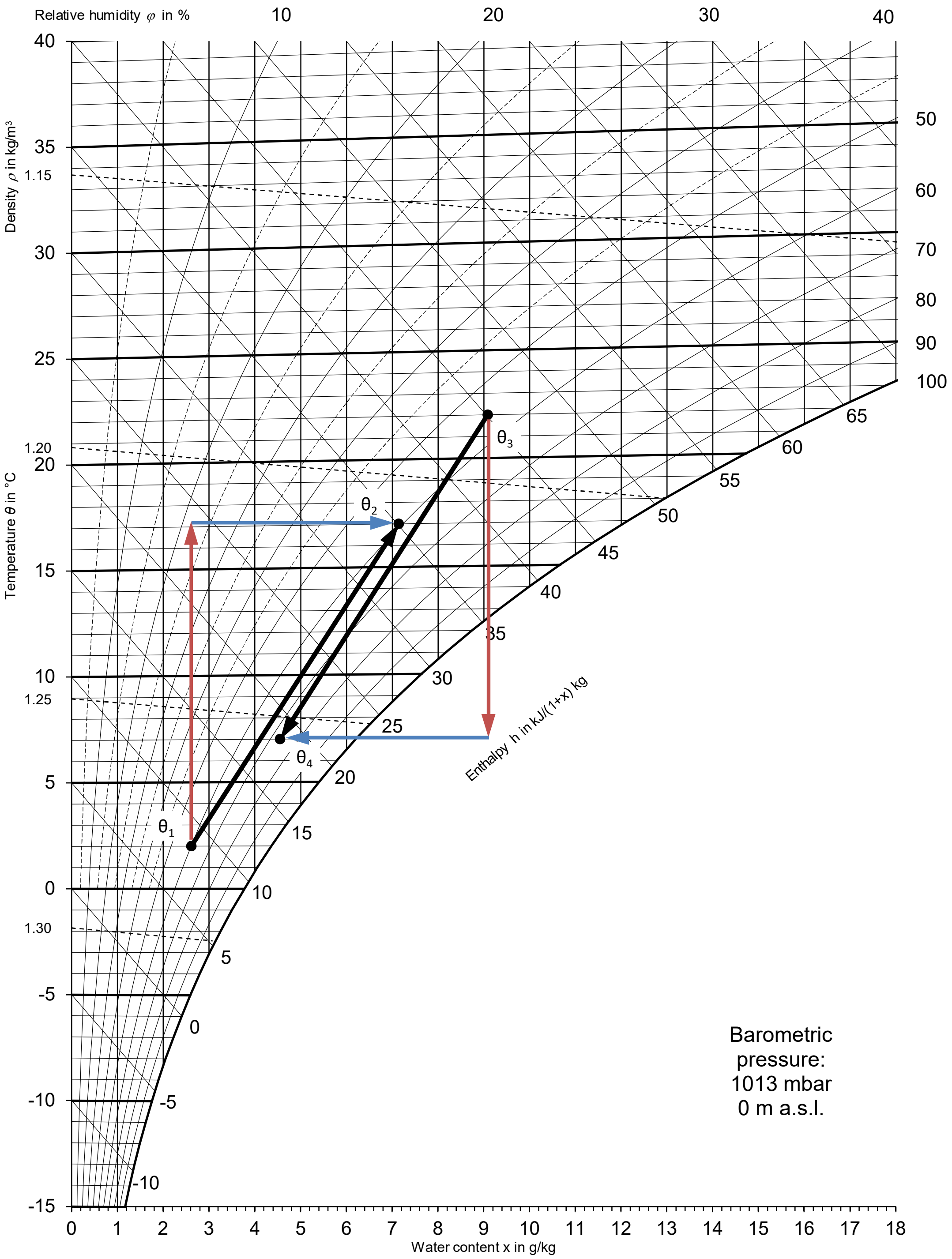


Fig. 3-11 Heat recovery for sensible and latent heat energy

Danger of icing with heat recovery systems

The example (Fig. 3-12) shows a plate exchanger HRU with the heat recovery coefficients (dry / wet) of approx. 82 % / 85 %. The mass extract air q_{mA} = 1 kg/s (= 3600 kg/h) air at θ_3 = 22 °C and φ_3 = 40 % RH is cooled down to θ_4 = 4.8 °C and there is also condensate Δx occurring. The heat transferred to the outside air stream includes the heat of condensation released and is Δh = (38.8-18.4) kJ/kg = 20.4 kJ/kg. This warms the outside air from θ_1 = -2 °C and φ_1 = 70% RH to θ_2 = 18.4 °C (Δh = 20.4 kJ/kg).

In winter this can lead to icing of the plate heat exchanger if the temperature of the outside air is even lower than shown in this example. This is also true for a closed circuit system. In the case of a rotary exchange system the icing only occurs at very low outside air temperatures.

For additional information on how to prevent icing by controls refer to “Controls of ventilation and air conditioning plants”.

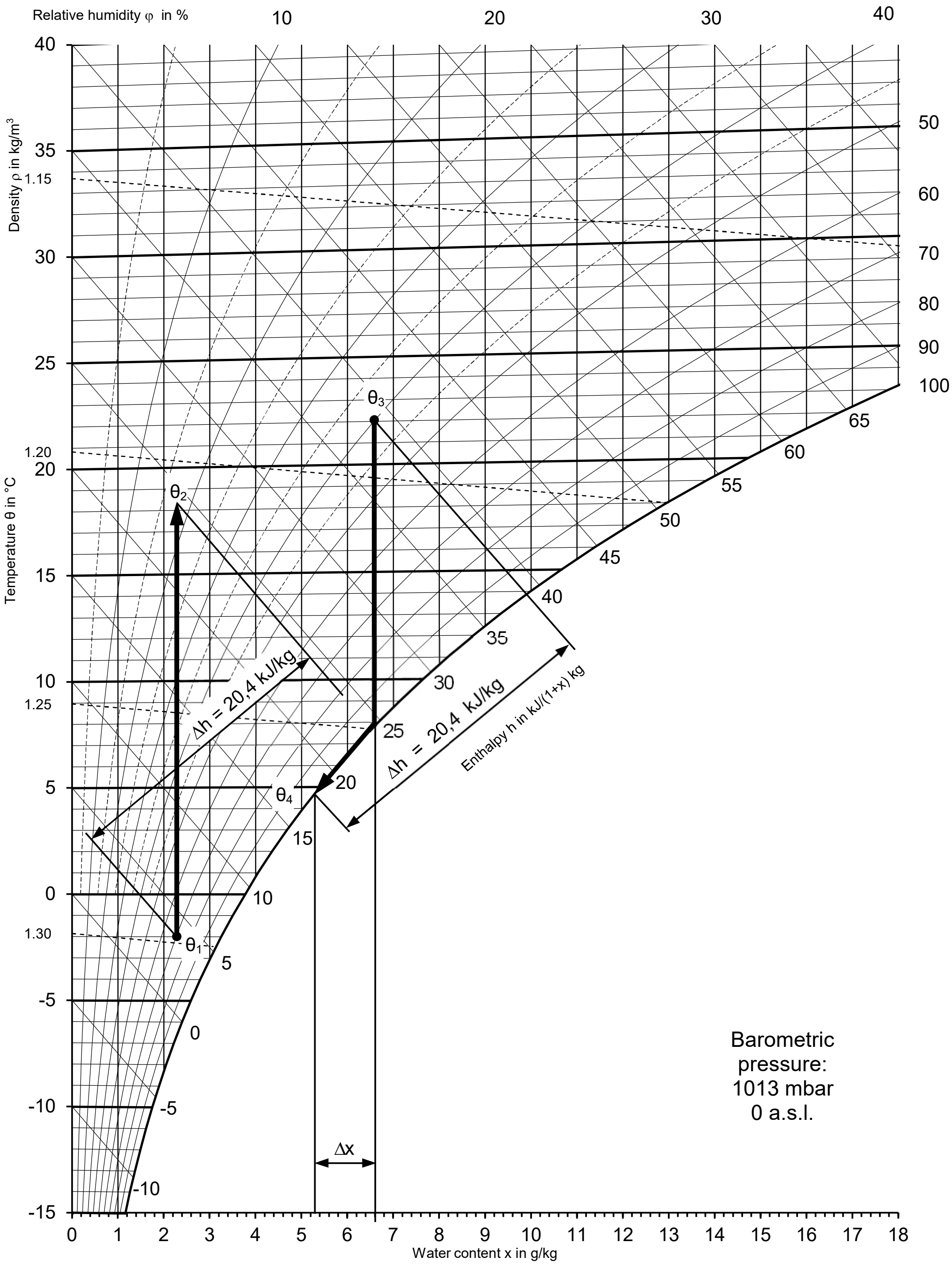


Fig. 3-12 Condensation Δx with sensible heat recovery (possible danger of icing)

3.9 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, h_1, then not only the absolute humidity x does change, but also the enthalpy h is increased by the enthalpy of the added water or vapor.

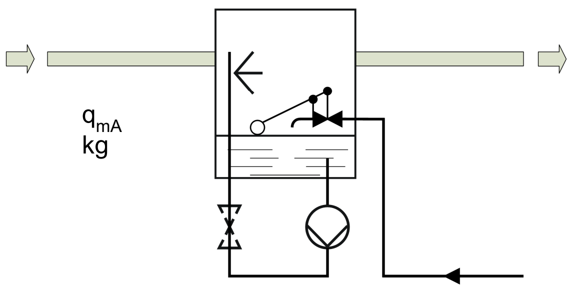
Δx is the weight and Δh is the enthalpy of the added water or vapor.

The state of the humidified air x_2, h_2 is then:

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

3.10 Humidifying with water (excess water)

Water is sprayed in the jet chamber of the 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 float controller ensures that the small amount of water vaporized and absorbed by the air is replaced (Fig. 3-13).



Humidifying using a circulating water

For the consideration of the circulating water humidification process, we take the initial state P of the air to be θ_A = 23 °C, φ = 25 % RH, and the temperature of the water in the collector to be θ_W = 17 °C.

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 isenthalpic lines (adiabatic lines) 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_{WB} (see section 2.2.8).

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.

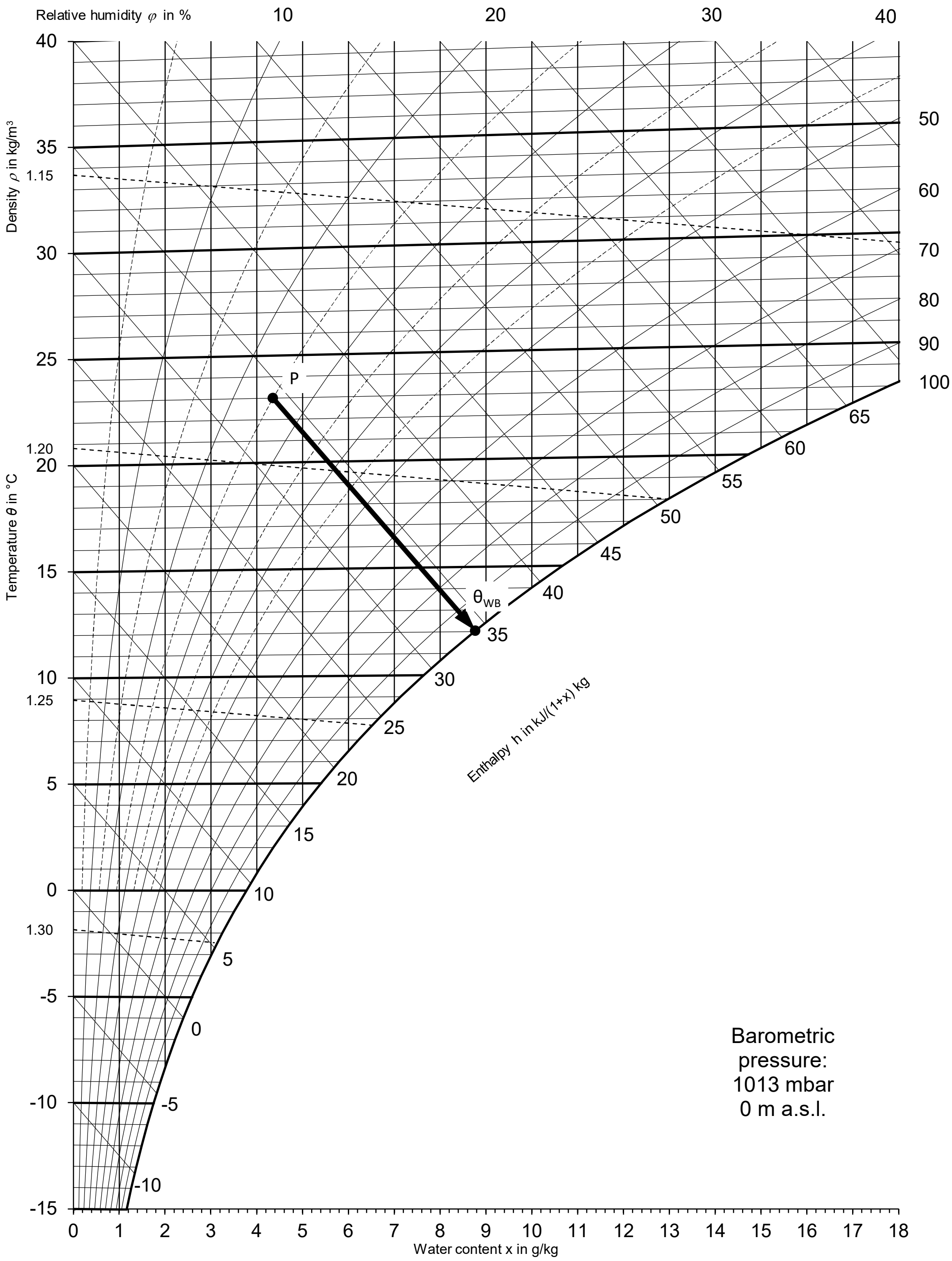


Fig. 3-13 Humidifying with circulation water

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_{WB}, 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 increases or decreases by the enthalpy of the water being absorbed, (depending on the temperature and volume of the water added). In the important areas for air-conditioning operation this error, however, is of negligible importance.

If air of state P1, with θ_1 = 27 °C, φ_1 = 30 % RH is to be adiabatically humidified to state P_2 with φ_2 = 80 % RH, then we first have to calculate the humidification efficiency (Fig. 3-14). 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: x1 = 6.6 [g/kg]
- At the humidifier discharge: x2 = 10.3 [g/kg]
- At the saturation state: x3 = 11.2 [g/kg]

The required humidification efficiency η_h can be calculated as follows:

$$\eta_h = \frac{x_2 - x_1}{x_3 - x_1} \cdot 100\ \% = \frac{(10.3 - 6.6)\ \frac{g}{kg}}{(11.2 - 6.6)\ \frac{g}{kg}} \cdot 100\ \% = \frac{3.7}{4.6} \cdot 100\ \% = 80.4\ \%$$

The data sheets of a humidifier manufacturer provide the required information, that, for example, a humidifier having two jet nozzles 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.

This line of thought can also be applied to other types of evaporating humidifiers such as contact humidifier, cold vapor generator etc.

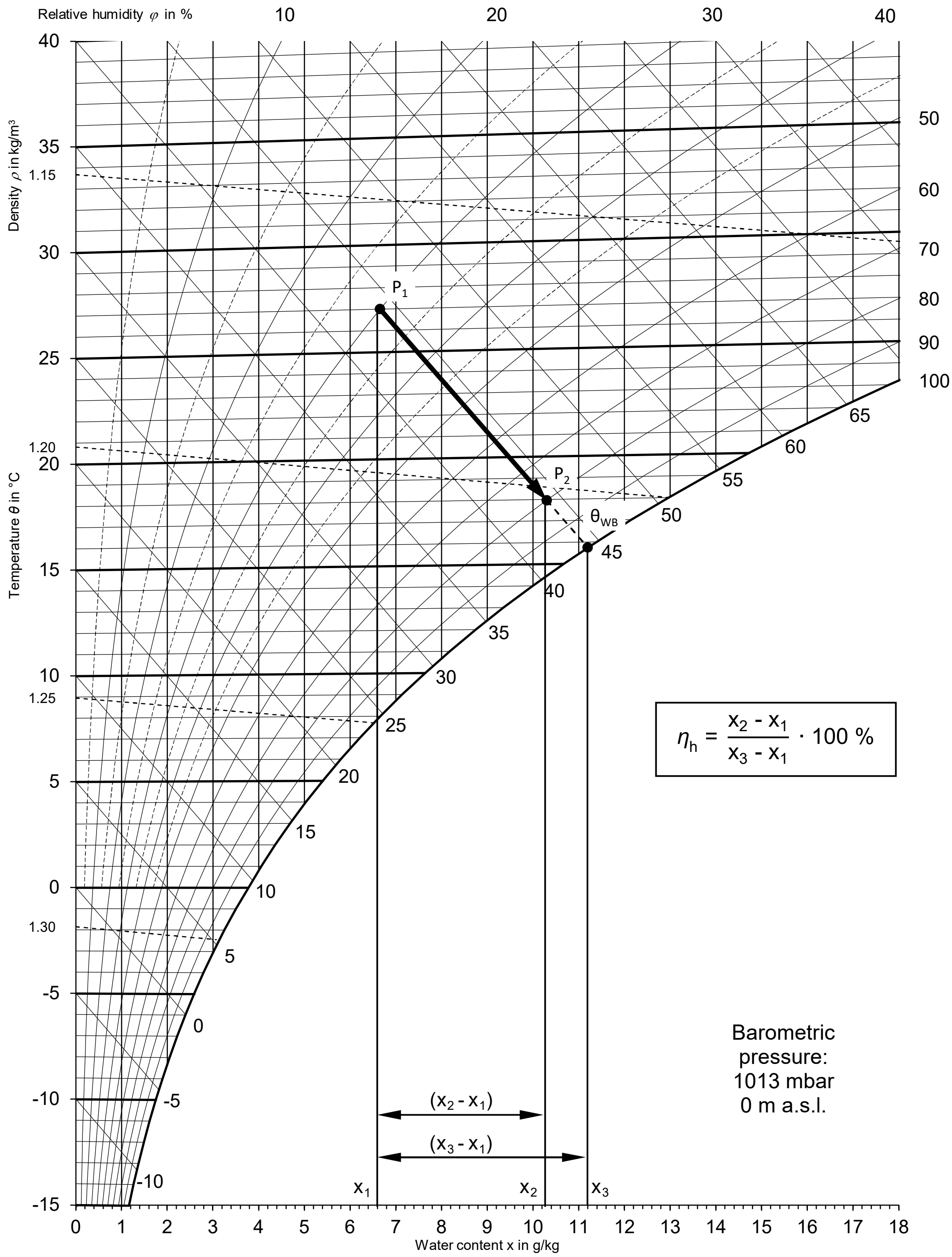


Fig. 3-14 Determination of humidification efficiency η_h

3.11 Humidifying with steam

This process involves blowing saturated steam into the air channel. The steam is withdrawn from a local steam air-humidifier or from a central steam generation system (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 2676 kJ/kg for steam at 100 °C (see tables in the end of this chapter).

The enthalpy increase during the steam humidification amounts to:

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

If 1 kg of air at e.g., θ_1 = 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. 3-15), then the amount of heat added is:

$$\Delta h = \Delta x \cdot h_S = 0.006 \frac{kg}{kg} \cdot 2676 \frac{kJ}{kg} \approx 16.1 \frac{kJ}{kg}$$

The state of the humidified air consists of:

$$h_2 = h_1 + \Delta h = (32.5 + 16.1) \frac{kJ}{kg} = 48.6 \frac{kJ}{kg}$$

$$x_2 = x_1 + \Delta x = (5 + 6) \frac{g}{kg} = 11 \frac{g}{kg}$$

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 h_S of the added steam. Because of this, some 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. 3-16).

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 chapter).

The direction of the state change is thus determined as follows:

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_S (e.g., 2676 kJ/kg) on the direction scale.
3. Slide this straight line parallel to itself to starting state θ_1 and possibly extend it to the vertical line at x_2. The intersection with this vertical line shows 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 i.e. there is practically no temperature change, rather only a latent increase of the enthalpy and of the water vapor content of the air.

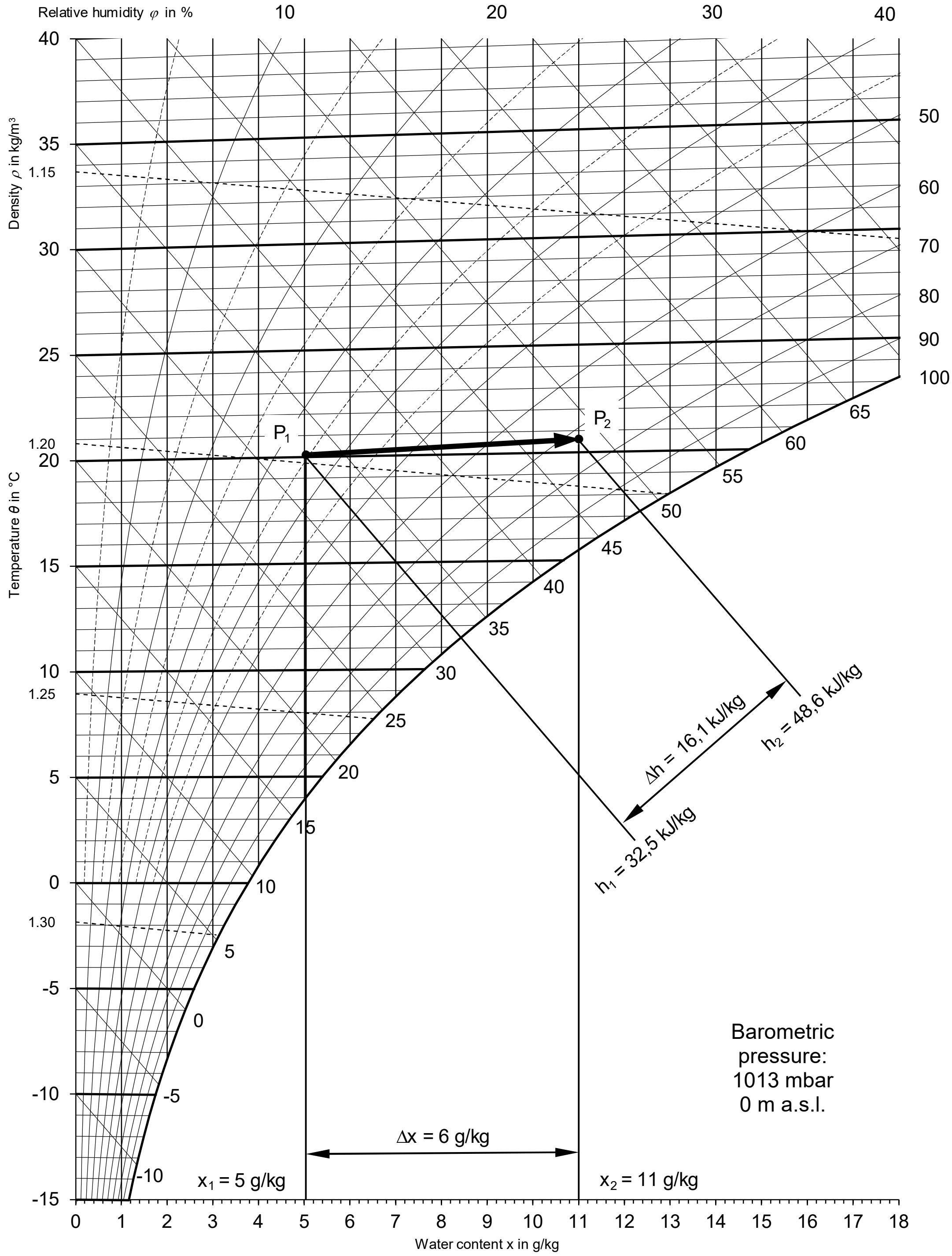


Fig. 3-15 Steam humidification using saturated steam (calculation method)

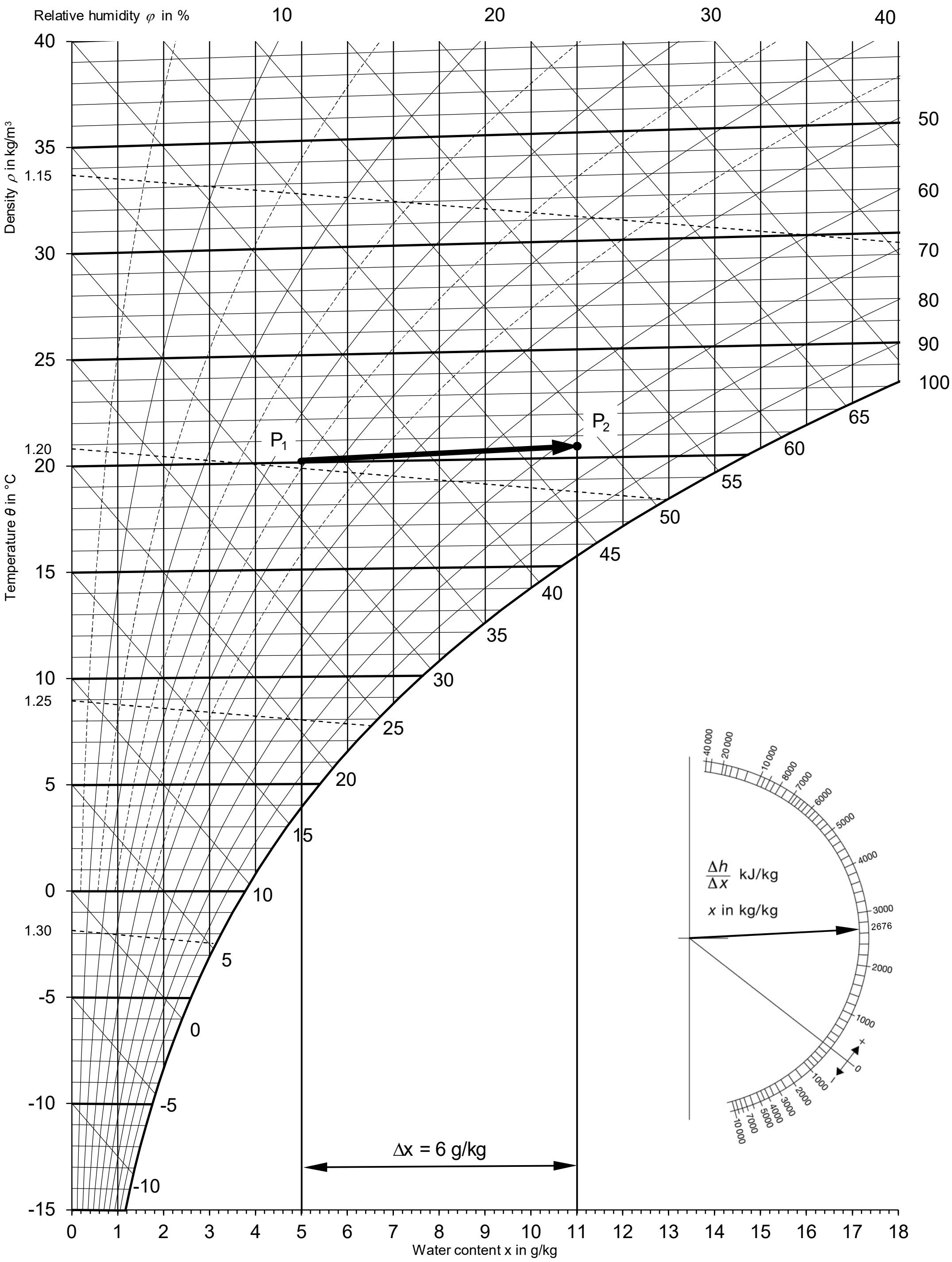


Fig. 3-16 Steam humidification (state change using the direction scale)

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

Temperature [°C]	**Absolute pressure** [bar]	**Enthalpy** [kJ/kg]		**Heat of evaporation** [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

Table 3-1 Variables as a function of temperature

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

Absolute pressure [bar]	**Temperature** [°C]	**Enthalpy** [kJ/kg]		**Heat of evaporation** [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

Table 3-2 Variables as a function of pressure

3.12 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 (undercooling method)
- Absorption of the water by absorption material (absorption method)
- Mixing in dry air.

3.12.1 Undercooling 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. 3-17) 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.

The amount of water condensing out depends on the cooling power. Water vapor content decreases by Δx (P_1 - P_2), 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 (P_2 – P_3). Heating, of course, lowers the relative humidity.

3.12.2 Absorption 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. 3-18 shows this (adiabatic) state change in the psychrometric chart.

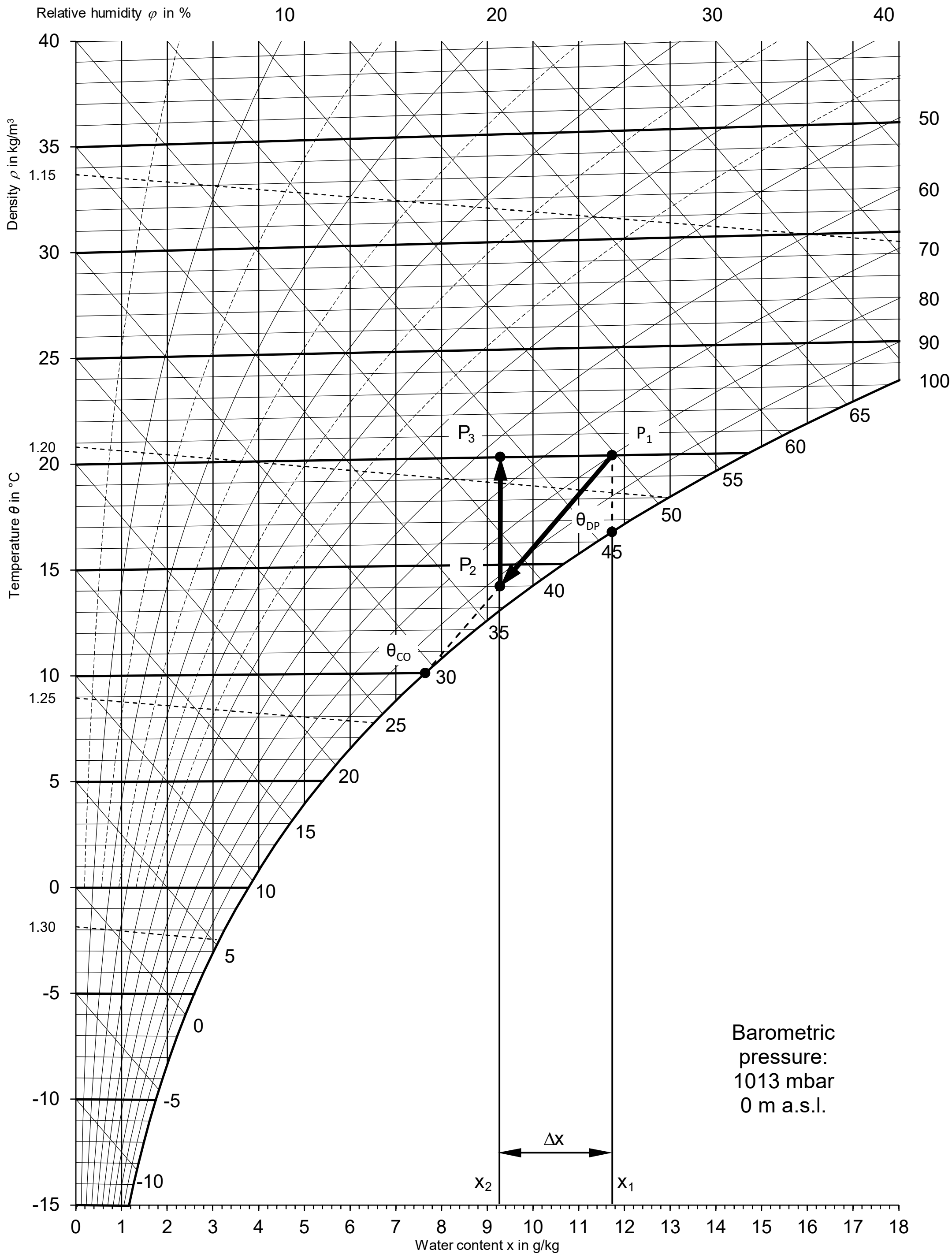


Fig. 3-17 Air drying by Δx using the undercooling method (with reheating)

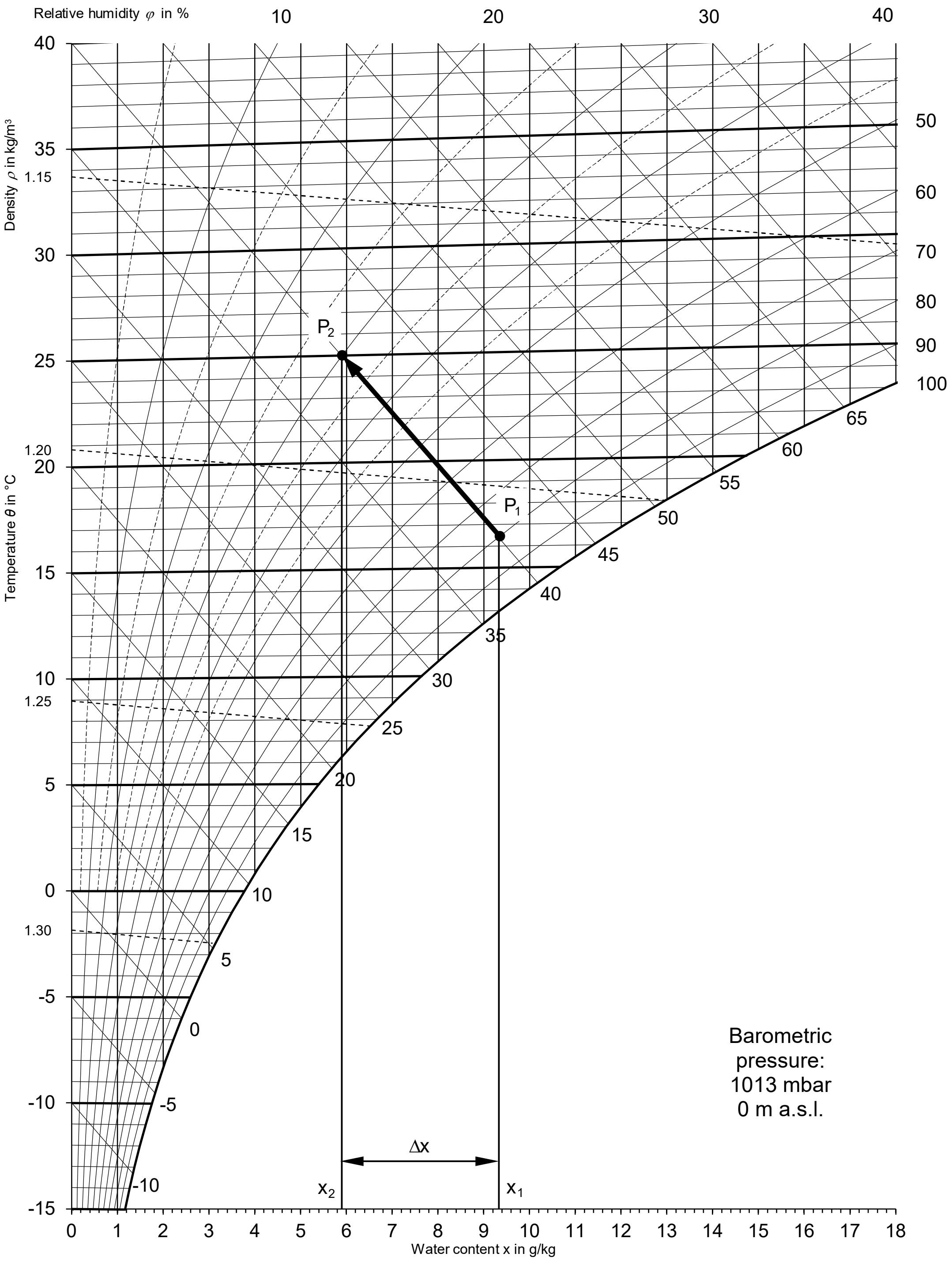


Fig. 3-18 Air drying by Δx using the absorption method

3.12.3 Air drying by mixing in dry air

This method is shown in Fig. 3-19. Here (mostly cold) air L_2 is mixed with humid air L_1 (φ_1 = 70 %, x_1 = 14 g/kg). The vapor content of L_2 is considerably below that of L_1. The state of the mixed air M_1 is determined by the mixing ratio (see section 3.2 "Mixing two quantities of air").

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 original 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 φ_2 = 50 % (application: e.g., swimming pool systems).

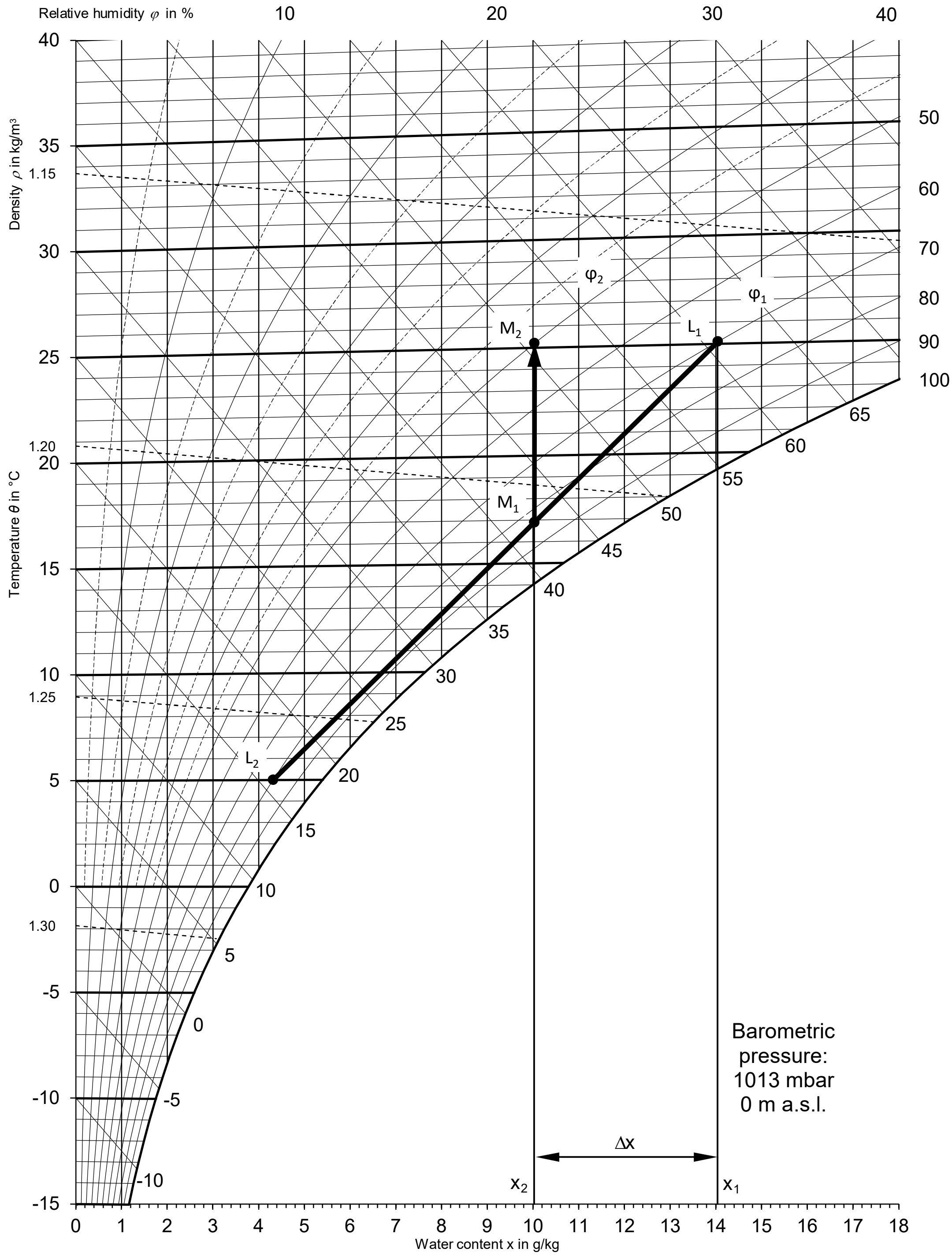


Fig. 3-19 Air drying by Δx by mixing and reheating

3.13 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. 3-20, for air state P_1, having temperature θ_1 = 15 °C and relative humidity φ_1 = 65 %, the air density is ρ_1 = 1.22 kg/m^3. For an air volume flow rate of q_{v1} = 1000 m^3/h the air mass flow rate is:

$$q_m = \rho_1 \cdot q_{v1} = 1.22\ \frac{kg}{m^3} \cdot 1000\ \frac{m^3}{h} = 1220\ \frac{kg}{h}$$

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 θ_2 = 25 °C, relative humidity φ_2 = 35 % and air density ρ_2 = 1.18 kg/m^3. The air volume flow rate q_{v1} = 1000 m^3/h then changes to:

$$q_{v2} = \frac{q_m}{\rho_2} = \frac{1220\ \frac{kg}{h}}{1.18\ \frac{kg}{m^3}} = 1034\ \frac{m^3}{h}$$

The air volume flow rate q_v changes from 1000 to 1034 m^3/h as a result of the temperature increase, while the air mass flow rate q_m (at constant absolute humidity x) remains unchanged.

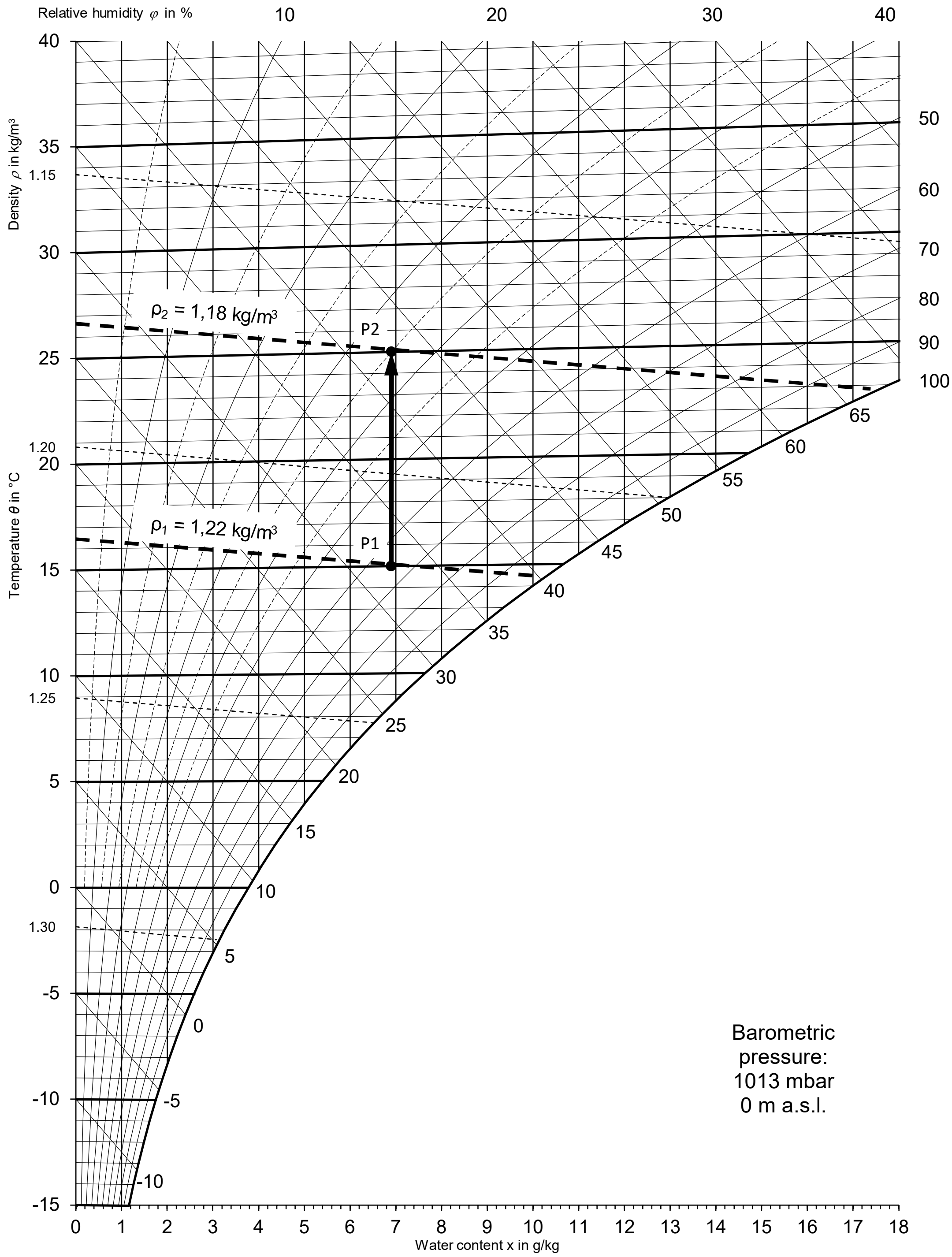


Fig. 3-20 Change in air density ρ due to a temperature increase

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 height above sea level. Thus, each psychrometric chart has a label specifying for which height above sea level or for which barometric pressure it is valid.

Units used for this pressure are:

meters above sea level [m a.s.l.], 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 = 1013 mbar = 760 torr.

How to convert or even redraw an existing psychrometric chart will only be discussed briefly in this chapter. Because other than manually converting or redrawing an existing chart, there are also selected charts available from suppliers for air conditioning units. Further various software-based solutions exist, which make psychometric chart calculations for an individual altitude possible.

4.2 Calculating the correction factors for *φ* and *ρ*

Most psychrometric charts include a table or diagram specifying correction factors for various heights above sea level (see tables in the end of this chapter). From the diagram you can determine the average barometric value or air pressure p for a specific height H.

Then, using the formula

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

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 height above sea level under consideration. An example:

At 1000 meters above sea level the average barometric pressure p_2 = 899 mbar.

The psychrometric chart, however, is referenced to p_1 = 1013 mbar.

The correction factor k is then:

$$k = \frac{p_2}{p_1} = \frac{899}{1013} \frac{\text{mbar}}{\text{mbar}} = 0.887$$

Using this correction factor, you can now determine the corrected values for the relative humidity lines. The values φ = 10 %, 20 %, 30 % relative humidity (RH), 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 % RH becomes a humidity line where φ = 8.9 % RH
- Humidity line φ = 20 % RH becomes a humidity line where φ = 17.7 % RH
- Humidity line φ = 100 % RH becomes a humidity line where φ = 88.7 % RH

5 Appendix

Conversion tables

Energy, work and quantity of heat

Unit:	J	kJ	kWh	kcal
1 J = 1 Nm = 1 Ws	1	0.001	$\approx 0.28\ 10^{-6}$	$\approx 0.24 \cdot 10^{-3}$
1 kJ = 1 kWs	1000	1	$\approx 0.28 \cdot 10^{-3}$	≈ 0.24
1 kWh	$3600 \cdot 10^{3}$	3600	1	860
1 kcal	4186	≈ 4.19	$\approx 1.16 \cdot 10^{-3}$	1

Power and heat flow (thermal output)

Unit:	W	kW	kJ/h	kcal/h
1 W = 1J/s	1	0.001	0.001	0.860
1 kW	1000	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	1163	$1.163 \cdot 10^{-3}$	≈ 4.19	1

Pressure

Unit:	Pa	Bar	mm WS	mm Hg (torr)
1 Pa = 1 N/m^2	1	10^{-5}	≈ 0.1	$\approx 7.5 \cdot 10^{-3}$
1 bar	10^5	1	$1.02 \cdot 10^4$	≈ 750
1 mm WS	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

Conversion factor for φ and ρ depending on elevation

H	**m a.s.l.**	**0**	**200**	**400**	**600**	**800**	**1000**	**1500**	**2000**
p	**kPa**	101.3	98.9	96.6	94.3	92.1	89.9	84.2	79.5
	mbar	1013	989	966	943	921	899	842	795
k	φ	1	0.976	0.953	0.931	0.901	0.887	0.831	0.785
	ρ	1	0.976	0.953	0.931	0.901	0.887	0.831	0.785

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