

Zeller's manual for air-conditioning (4 editions from 1994 to 1999)

Now we write the year 2021 and there are still requests for this book, which has been out of print for many years and is no longer reissued. Meanwhile, electronic distribution channels have largely replaced books, with the advantage that necessary corrections and additions can be implemented much easier. The book was developed from the experiences in the field of climatic engineering when handling humid air since 1970. It did not contain any fundamentally new insights. It was more than one compendium for interested professionals, engineers, lecturers and students. Furthermore it has been used gladly and actively as a supplementary specialist book to our varied multilingual software.

AHH Mollier diagram and Carrier psychrometric chart with air processes.
Range -100/300°C, 0/1000 g/kg, -5000/15000 m, 0.03/16 bar.
150 meteorological data, further locations from Meteonorm.
3 different ranges of comfort zone from DIN and ASHRAE.
Show your individual measurement points.

MDI Meteorological data interface: Define the service times.
Meteorological data based on Meteonorm.

AHU Air-handling unit configurator: Element handling per drag and drop.
Approximately measurements, weights, pressure drops, prices.

EAC Economy of AHU's with circuit connected heat recovery systems.
Variable air volume flows, amortization time, capital costs.

DEH Economy of AHU's with different heat recovery systems.
Variable air volume flows, amortization time, capital costs.

ESH Glycol Re-Cooler with axial fans, outside or in AHU's.
Dry, adiabatic and hybrid service, container measurements.

HEH Calculation of fin coils like heater, cooler, condenser, evaporator,
heater split, cooler split.

CCS Calculation of heat recovery circuit systems with fin coils.
Different systems with foreign energy in the circuit.

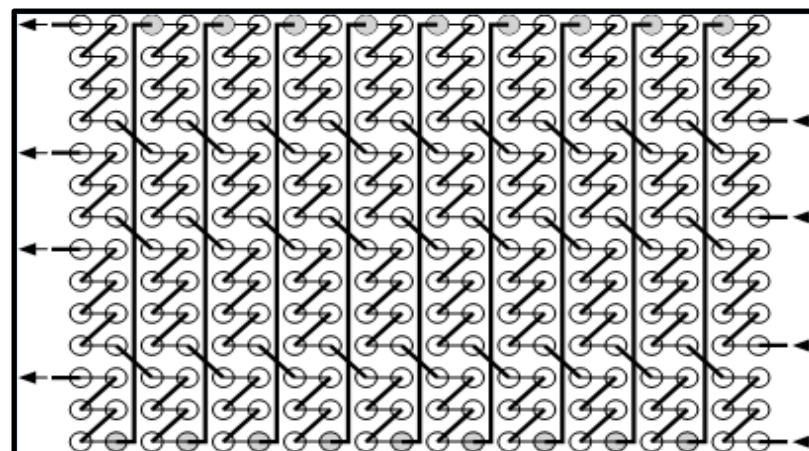
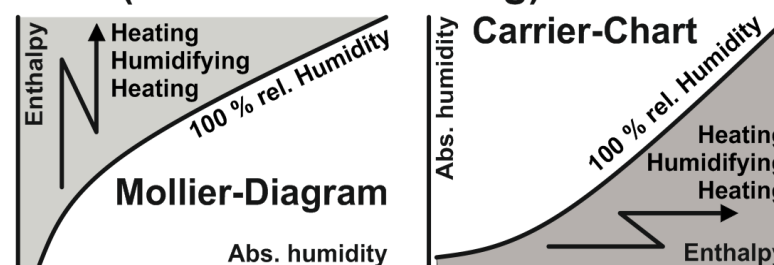
Diverse GHH, Mollier diagram for different types of gases and steams.
Spiral rib and circle rib heat exchangers.

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AHH (Air Humid Handling) = All in one!

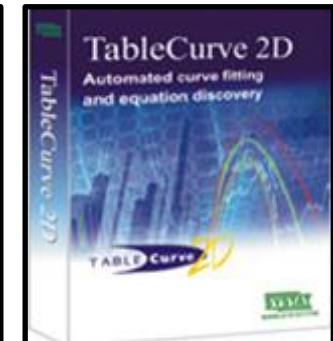
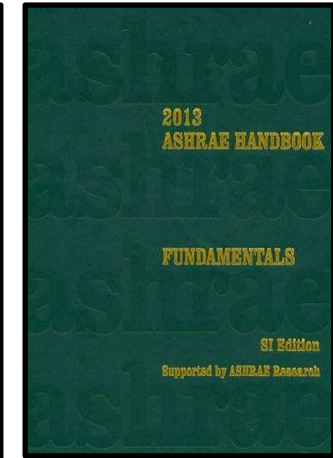


Mollier diagram / Carrier psychrometric chart (AHH) thematics

- 3 Representation according to Mollier and Carrier
- 4 Process data to the moist air, definition, area, thermodynamic properties
- 5 Thermodynamic properties
- 6 Approximation polynomials for the thermodynamic properties
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Fin coil heat exchanger (FCHE) thematics

- 11 FCHE measurements, corrugated fins, heat exchanger tubes inside grooved
- 12 FCHE measurements, fin thickness, basic equations
- 13 Average logarithmic temperature difference, counter flow, equal flow, cross flow, cross counter flow
- 14 Cross counter flow n-times, cooling process Δt_m , FCHE surface smooth and corrugated fins
- 15 FCHE surface inside and outside, influence on the heat transfer and pressure drop
- 16 Heat transfer coefficient outside
- 17 Heat transfer coefficient outside, k-coefficient, air cooler surface temperature
- 18 Pressure drop outside on air heater and air cooler sensible
- 19 Pressure drop outside on air heater hybrid and air cooler with condensate formation
- 20 Definition to the tube coupling and to the collectors, pressure drop in the collectors
- 21 Smooth and inside grooved tubes, pressure drop without aggregate state change and inside total
- 22 Heat transfer coefficient inside, agents without and with aggregate state change, condensation
- 23 Heat transfer coefficient inside, dry expansion evaporation, pump recirculation evaporation
- 24 Heat transfer coefficient with inside grooved tubes, pressure drop on the condensation, gradient
- 25 Pressure drop on the dry expansion evaporator, pressure drop in the capillaries
- 26 Pressure drop on the pump recirculation evaporator, CO₂ cooler in the supercritical area
- 27 CO₂ cooler in the supercritical area
- 28 Combination of gas mixtures (REFPROP from NIST) and condensable vapors
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- 30 Combination of gas mixtures and condensable vapors, application HEH-SR-G
- 31 Barometric altitude formula



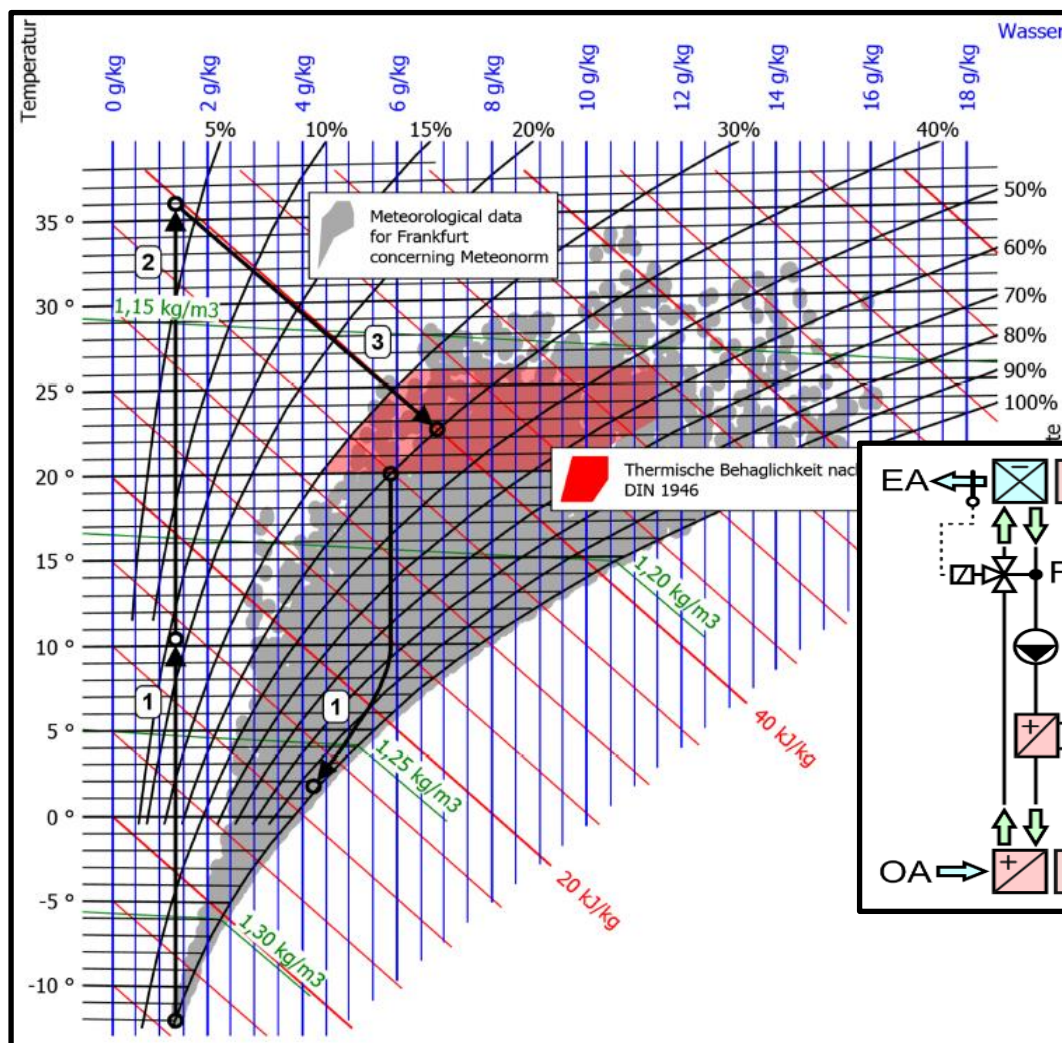
Dr.-Eng. Boris Slipcevic
Sulzer Escher Wyss
Lindau

Laboratory measurements were recorded by empirical equations. There followed numerous publications on heat transfer and pressure loss during condensation and evaporation (convective and bubble boiling) of refrigerants that flowed into the software.

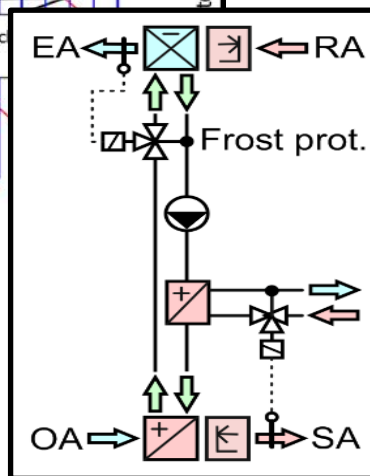
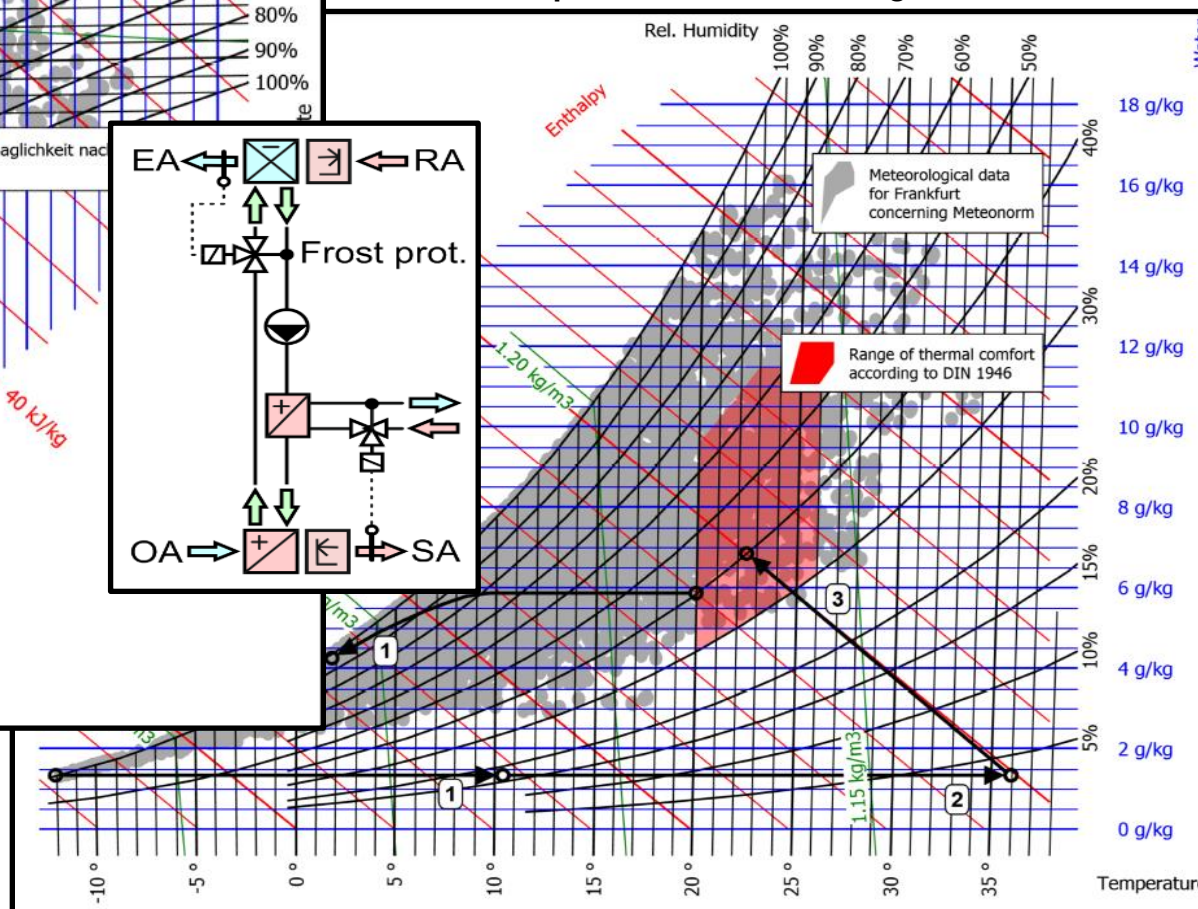
Cert.-Eng. Marin Zeller TU
Zeller Consulting Suisse
Ittigen

Laboratory measurements were recorded by empirical equations, especially to the lamella impression, to the contact efficiency between tubes and fins, to the cooling process with formation of condensate, the surface temperature and to the pressure drop wet of the moist air.

Representation according to Mollier



Representation according to Carrier



Process data to the moist air

Definition

The Mollier-hx-diagram represents the air water mixture. It is in such a way developed that the 0 °C isotherm is horizontal in the range of unsaturated air. The nebula isotherm of over saturated air by 0 °C go parallel to the enthalpy. On $t = 0$ °C and $x = 0$ kg/kg the enthalpy $h = 0$ J/kg, which leads to ranges with negative enthalpies. By exchange of the centerlines one receives the Carrier-xh-Diagram (Psychrometric chart) within the software AHH to be alternatively worked can.

Area

Usually the Mollier-hx-Diagram is based on a pressure of 1.013 bar according to sea level and exhibits a range, which permits not all applications. The software AHH permits the desired range for each application and supports the good clarity because of each stretching of the axes of coordinates.

Temperature	-100 to 300°C
Absolute humidity	0 to 1000 g/kg
Pressure absolute	0.1 to 16 bar
Height	-5000 to 15000 m

Thermodynamic properties

In specialized books one usually finds the specific temperature-referred thermal capacity. This value points out, how much energy must be spent, in order to warm up the medium with appropriate temperature around 1°C. If one wants to know, which energy is needed, in order the medium of t_1 to t_2 to warm up, the means of the specific temperature-referred thermal capacity must be determined. Below the average values were formed of 0 °C to t °C and combined into tables and approximation polynomials, which make a fast processing possible on computers.

1) Heat recovery - Circuit connect-system

Efficiency temperature	%	70.000
Efficiency hygroscopic	%	0.000
Efficiency humid	%	0.000
Capacity	kW	62.759
Mean temp.diff.	K	11.560
Coefficient	kW/K	5.429

		Cold air In	Cold air Out	Hot air In	Hot air Out
Temperature	°C	-12.000	10.400	20.000	1.771
Rel. Humidity	%	100.000	17.090	40.000	97.451
Abs. Humidity	g/kg	1.346	1.346	5.872	4.255
Density humid	kg/m³	1.330	1.225	1.182	1.261
Enthalpy humid	kJ/kg	-8.737	13.857	35.028	12.435
Volume flow humid	m³/h	7527.666	8173.325	8511.363	7961.565
Mass flow dry	kg/h	10000.000	10000.000	10000.000	10000.000
Condensed water	kg/h		0.000		16.170
Surface temperature	°C				-3.324

Danger of FREEZING!

2) Heating

Capacity	kW	71.780	
		Air In	Air Out
Temperature	°C	10.400	36.000
Rel. Humidity	%	17.090	3.641
Abs. Humidity	g/kg	1.346	1.346
Density humid	kg/m³	1.225	1.124
Enthalpy humid	kJ/kg	13.857	39.697
Volume flow humid	m³/h	8173.325	8911.214
Mass flow dry	kg/h	10000.000	10000.000

3) Moistening of air with water

Capacity	kW	0.952	
Moistening flow	kg/h	55.020	
Moistening temperature	°C	15.000	
Moistening enthalpy	kJ/kg	62.302	
		Air In	Air Out
Temperature	°C	36.000	22.488
Rel. Humidity	%	3.641	40.000
Abs. Humidity	g/kg	1.346	6.848
Density humid	kg/m³	1.124	1.171
Enthalpy humid	kJ/kg	39.697	40.040
Volume flow humid	m³/h	8911.151	8596.927
Mass flow dry	kg/h	10000.000	10000.000

Mollier diagram / Carrier psychrometric chart							Symbol	Unit	Description	Page 5
Thermodynamic properties							cp_d	J/kgK	Heat capacity from water vapor on Solidus	
t	cp_l	cp_d	p_d	h_w	h_d	r	cp_l	J/kgK	Heat capacity from dry air	
-100	1007.20	1815.40	0.00160				h_d	J/kgK	Enthalpy from water vapor on Solidus	
-90	1006.90	1817.50	0.00933				h_w	J/kgK	Enthalpy from water vapor on Liquidus	
-80	1006.63	1819.60	0.05333				p_d	Pa	Partial pressure from water vapor	
-70	1006.40	1821.70	0.258				r	J/kgK	Evaporation heat from water vapor	
-60	1006.20	1823.80	1.076				t	°C	Temperature	
-50	1006.07	1826.00	3.939							
-40	1006.00	1828.10	12.870							
-30	1005.97	1830.30	38.101							
-20	1006.00	1832.50	103.450							
-10	1006.08	1834.70	259.980							
0	1006.18	1836.90	610.480	0	2500500	2500500				
10	1006.31	1839.10	1230	42000	2518900	2476900				
20	1006.45	1841.40	2340	83900	2537300	2453400				
30	1006.60	1843.70	4240	125600	2555500	2429900				
40	1006.81	1846.00	7370	167300	2573500	2406200				
50	1007.03	1848.30	12300	209100	2591300	2382200				
60	1007.30	1850.60	19900	250900	2608800	2357900				
70	1007.60	1852.90	31100	292800	2625900	2333100				
80	1007.90	1855.30	47300	334700	2642500	2307800				
90	1008.30	1857.70	70100	376800	2658700	2281900				
100	1008.70	1860.10	101300	418900	2674400	2255500				
110	1009.00	1862.50	143300	461100	2689600	2228500				
120	1009.50	1864.90	198500	503500	2704200	2200700				
130	1009.90	1867.30	270100	546100	2718300	2172200				
140	1010.30	1869.80	361400	588900	2731800	2142900				
150	1010.80	1872.30	476000	631900	2744500	2112600				
160	1011.30	1874.80	618000	675200	2756500	2081300				
170	1011.80	1877.30	792000	718800	2767600	2048800				
180	1012.40	1879.80	1002700	762700	2777600	2014900				
190	1013.00	1882.40	1255200	807000	2786300	1979300				
200	1013.60	1884.90	1555100	851800	2793700	1941900				
210	1014.20	1887.50	1908000	897100	2799400	1902300				
220	1014.80	1890.10	2320100	943000	2803400	1860400				
230	1015.50	1892.70	2797900	989600	2805400	1815800				
240	1016.20	1895.30	3348000	1036900	2805100	1768200				
250	1016.90	1898.00	3978000	1085100	2802500	1717400				
260	1017.60	1900.60	4694000	1134300	2797400	1663100				
270	1018.40	1903.30	5505000	1184500	2789500	1605000				
280	1019.20	1906.00	6419000	1236100	2778700	1542600				
290	1020.10	1908.70	7445000	1289300	2764900	1475600				
300	1021.00	1911.40	8592000	1344200	2748000	1403800				

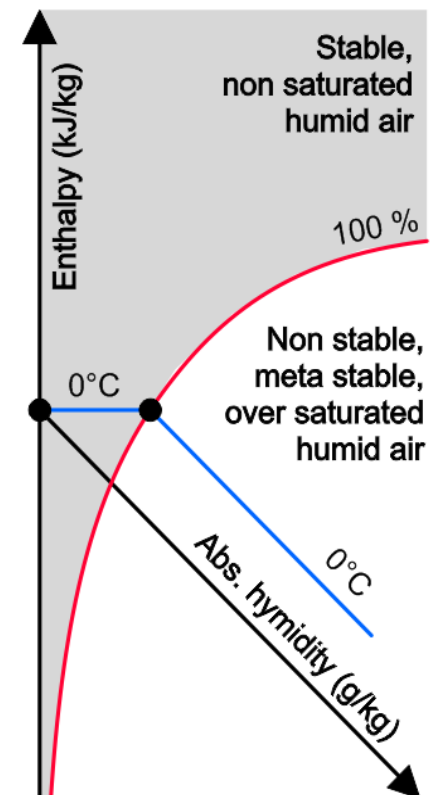
$$cp = \frac{\int_{t_1}^{t_2} cp_t dt}{t_2 - t_1}$$

$$t_1 = 0 \text{ and } t_2 = t \rightarrow cp = \frac{\int_0^t cp_t dt}{t}$$

Richard Mollier
1863 - 1935



W. H. Carrier
1876 - 1950



Approximation polynomials (-100 < t < 300°C)

a = 1.00617203411816E+03
b = -5.14584155927084E-04
c = -5.07744861271335E-01
d = -4.08693984761444E-06
e = -3.94830238325583E-03
f = 3.86998536082132E-10

$$cp_l = \frac{a + ct + et^2}{1 + bt + dt^2 + ft^3}$$

a = 1.83690225155577E+03
b = 2.96850242760703E-04
c = 7.68576185706328E-01
d = -8.23605125618347E-08

$$cp_d = \frac{a + ct}{1 + bt + dt^2}$$

a = 6.41424538282508E+00
b = 1.34952974449424E-02
c = 1.68771989526873E-01
d = 3.60425763984253E-05
e = 1.03764255356861E-03
f = -2.54470285416322E-09

$$-100 < t \leq 0 \rightarrow \ln(p_d) = \frac{a + ct + et^2}{1 + bt + dt^2 + ft^3}$$

a = 6.41425292688508E+00
b = -1.17398221741019E-02
c = -2.31391504282494E-03
d = 5.85324578180939E-05
e = -7.97076080224934E-04
f = 3.14129016158240E-07
g = 1.13870924045918E-05
h = -1.05815083120807E-09
i = -2.45705078974294E-08

$$0 > t \leq 300 \rightarrow \ln(p_d) = \frac{a + ct + et^2 + gt^3 + it^4}{1 + bt + dt^2 + ft^3 + ht^4}$$

a = 3.63051146855678E+00
b = -2.04547872985726E-03
c = 4.19231841629432E+03
d = -8.10031771231269E-07
e = -8.90297066641696E+00
f = 9.66233360384174E-10

$$h_w = \frac{a + ct + et^2}{1 + bt + dt^2 + ft^3}$$

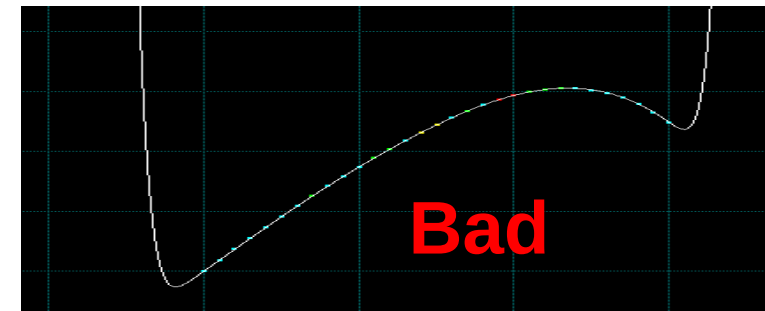
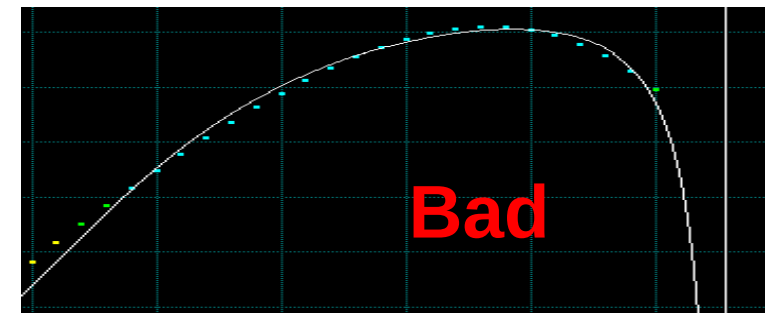
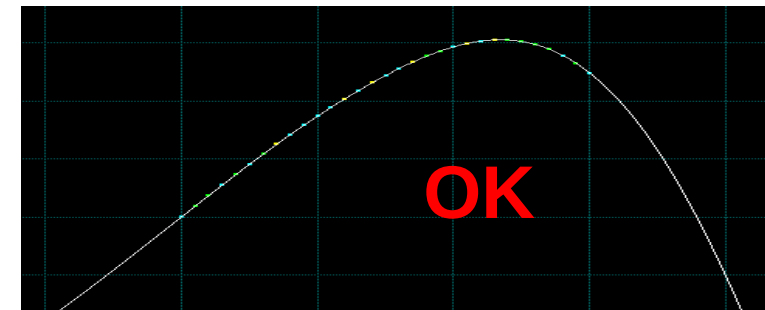
a = 2.50049979241906E+06
b = -1.09042949248609E-02
c = -2.54218801181540E+04
d = 6.03477944019292E-05
e = 1.30688801015895E+02
f = -1.24219636812250E-07
g = -2.06485565012501E-01
h = 1.15773454895717E-10
i = 1.21390880909374E-04

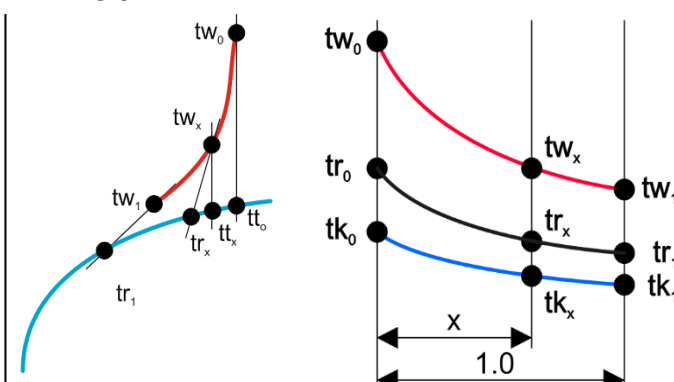
$$h_d = \frac{a + ct + et^2 + gt^3 + it^4}{1 + bt + dt^2 + ft^3 + ht^4}$$

$$r = h_d - h_w$$

The application TableCurve 2D offers more than 7000 equations, sorted according to the smallest error rate. However, not all equations are suitable to the same extent, but 3 criteria must be observed:

1. The waves within the data should be minimal.
2. Within the data, singularities are to be excluded.
3. The trend outside of the data should be continuous.



Mollier Diagramm / Carrier Psychrometric Chart	Symbol	Unit	Description	Page 7
Equations for moist air The air pressure depends on the height over sea level, the temperature and the humidity. As basis for Mollier diagram and Carrier chart the air pressure is to determine by the height over sea level, the middle yearly temperature (average 10°C for middle Europe) and the middle yearly humidity (average 80% for middle Europe). $T_{lf} = t_{lf} + 273.15 \quad z = \frac{M_l g H}{R T_{lf}} \frac{1+x}{1+x \frac{M_l}{M_w}} \rightarrow p_{lf} = 1.01325 e^{-z}$ $\rho_{lf} = \frac{M_l p_{lf}}{R T_{lf}} \frac{1+x}{1+x \frac{M_l}{M_w}} \quad x_s = \frac{M_w}{M_l} \frac{p_d}{p_{lf} - p_d} \quad x = \frac{M_w}{M_l} \frac{\phi_{lf} p_d}{p_{lf} - \phi_{lf} p_d}$ $\phi_{lf} = \frac{p_{lf} x \frac{M_l}{M_w}}{p_d \left(1 + x \frac{M_l}{M_w}\right)} \quad h_{lf} = c_{p_l} t + x(r_0 + c_{p_d} t) \quad \dot{Q}_{lf} = \dot{M}_{lt} \Delta h_{lf}$ $t_{lf} = \frac{h_{lf} - x r_0}{c_{p_l} + x c_{p_d}} \quad \dot{M}_{lt} = \frac{\dot{V}_{lf} \rho_{lf}}{1+x} \quad \dot{V}_{lf} = \frac{\dot{M}_{lt} (1+x)}{\rho_{lf}}$	H h_{lf} M_l $\dot{M}_{lt}$ M_w p_{lf} ϕ_{lf} $\dot{Q}_{lf}$ R $r_0(0^\circ\text{C})$ ρ_{lf} t_{lf} T_{lf} $\dot{V}_{lf}$ x x_s	m J/kg kg/kMol kg/s kg/kMol Pa kg/m3 W J/kMolK J/kg kg/m3 °C K m3/s kg/kg kg/kg	Height over sea level Enthalpy from the moist air Molecular weight from air = 28.949 kg/kMol Massflow from the dry air Molecular weight from water = 18.015 kg/kMol Pressure from the moist air Relative humidity from the moist air Capacity with moist air Universal gas constant = 8314.46 J/kMolK Evaporation heat from water vapor = 2500500 J/kg Density from the moist air Temperature from the moist air Temperature from the moist air Volume flow from the moist air Absolute humidity from the moist air Maximally absolute humidity from the moist air	
Cooling process 	In the software the cooling process in the heat exchanger is divided in 15 cells in air direction. Here, it is assumed that a high degree of cross-counter-flow. The surface temperature tr_x plays in each cell a crucial role. When this is less than the dew point tx , condensate forms. For small $tx - tr_x$, the droplets of condensate are small too. These can be separated in a 1st demister only, which produce bigger droplets. A 2nd demister can separate this big droplets. Demisters with less than 100 Pa pressure drop have a bad separation. This play the big rule, when the dehumidification during the cooling process is important.			

Mollier Diagramm / Carrier Psychrometric Chart	Symbol	Unit	Description	Page 8
Humidification (Water, wet steam, saturated steam) The humidification direction is carried out in the Mollier diagram on paper with the aid of the edge measuring rod. This is not possible in the software AHH because the two axes can be freely selected. $h_{b(Wasser)} = h_w \quad h_{b(Nassdampf)} = h_w + a_d r \quad h_{b(Sattdampf)} = h_d$ $\Delta h = h_a - h_e = \Delta x h_b = h_b \frac{\dot{M}_b}{\dot{M}_l} \quad \Delta x = x_a - x_e = \frac{\dot{M}_b}{\dot{M}_l} = \frac{\Delta h}{h_b}$ $\dot{M}_b = \Delta x \dot{M}_l \quad \dot{Q} = \Delta h \dot{M}_l$	a_d Δh Δx h_a h_b h_e $\dot{M}_b$ x_a x_e	--- J/kg kg/kg J/kg J/kg J/kg kg/s kg/kg kg/kg	Wet part of steam Enthalpy difference Humidity difference Enthalpy outlet Humidification enthalpy Enthalpy inlet Humidification mass flow Humidity outlet Humidity inlet	
Examples to the Mollier diagram on the right Humidification with water of 0°C $h_b = h_w = 0 \text{ J/kg}$ Humidification with water of 50°C $h_b = h_w = 209'100 \text{ J/kg}$ Humidification with wet steam of 110°C, wet part of steam 50 % $h_b = h_w + a_d r = 461'100 + 0.5 \cdot 2'228'500 = 1'575'350 \text{ J/kg}$ Humidification with saturated vapor of 150°C $h_b = h_d = 2'744'500 \text{ J/kg}$				
Note that moistening with constant enthalpy can only be achieved with water of 0°C. During humidification with water > 0°C the enthalpy increases, albeit little. If the desired relative humidity at the outlet is desired as an input value, this must be done by means of iteration, which is the case in the software.				
The Mollier diagram plots Temperature (°C) on the y-axis (20 to 40) against Humidity (g/kg) on the x-axis (0 to 20). It includes saturation curves for water and steam, and various process lines for constant enthalpy, constant humidity, and constant wet-bulb temperature. A specific humidification path is highlighted with points 1, 2, 3, and 4. Point 1 is at 30°C and 4 g/kg. Point 2 is at 30°C and 7 g/kg. Point 3 is at 30°C and 10 g/kg. Point 4 is at 30°C and 15 g/kg. The path from 1 to 2 is horizontal (constant enthalpy). The path from 2 to 3 is vertical (constant humidity). The path from 3 to 4 is diagonal (constant wet-bulb temperature). A text box provides details about the air conditions and the humidification process. Air pressure 1.01325 bar Air on inlet 10 kg/s / 30 °C / 40 % / 2 g/kg abs. humidity Air on outlet 7 g/kg abs. humidity 1) Humidification with water from 0 °C 2) Humidification with water from 50 °C 3) Humidification with wet steam 110 °C / 50 % 4) Humidification with saturated steam from 150 °C				

Meteorological data and a correct cooler design for sweltering midsummer

The German standard DIN 4710 recorded 87'600 events à 0.1 hours per year as average values of the period from 1961 to 1990 and therefore forms a large area in the Mollier diagram. The software from Meteonorm recorded 8'760 events à 1.0 hours, and therefore forms a smaller area in the Mollier diagram than the German standard DIN 4710 what is a risk in the design of coolers in sweltering midsummer.

A = 32.0°C / 40 % / 63.0 kJ/kg = Usual calculation
 B = 31.0°C / 58 % / 73.7 kJ/kg = Meteonorm for 2020
 C = 32.0°C / 65 % / 82.9 kJ/kg = DIN 4710, 1961-1990
 D = 12.3°C / 100 % / 35.2 kJ/kg = Cooling
 E = 18.0°C / 69 % / 41.0 kJ/kg = Reating

A - D: Big risk management

$63.0 - 35.2 = 27.8 \text{ kJ/kg} = 58.28 \%$

Cooler size loss = 41.72 %

B - D: Middle risk management

$73.7 - 35.2 = 38.5 \text{ kJ/kg} = 80.71 \%$

Cooler size loss = 19.29 %

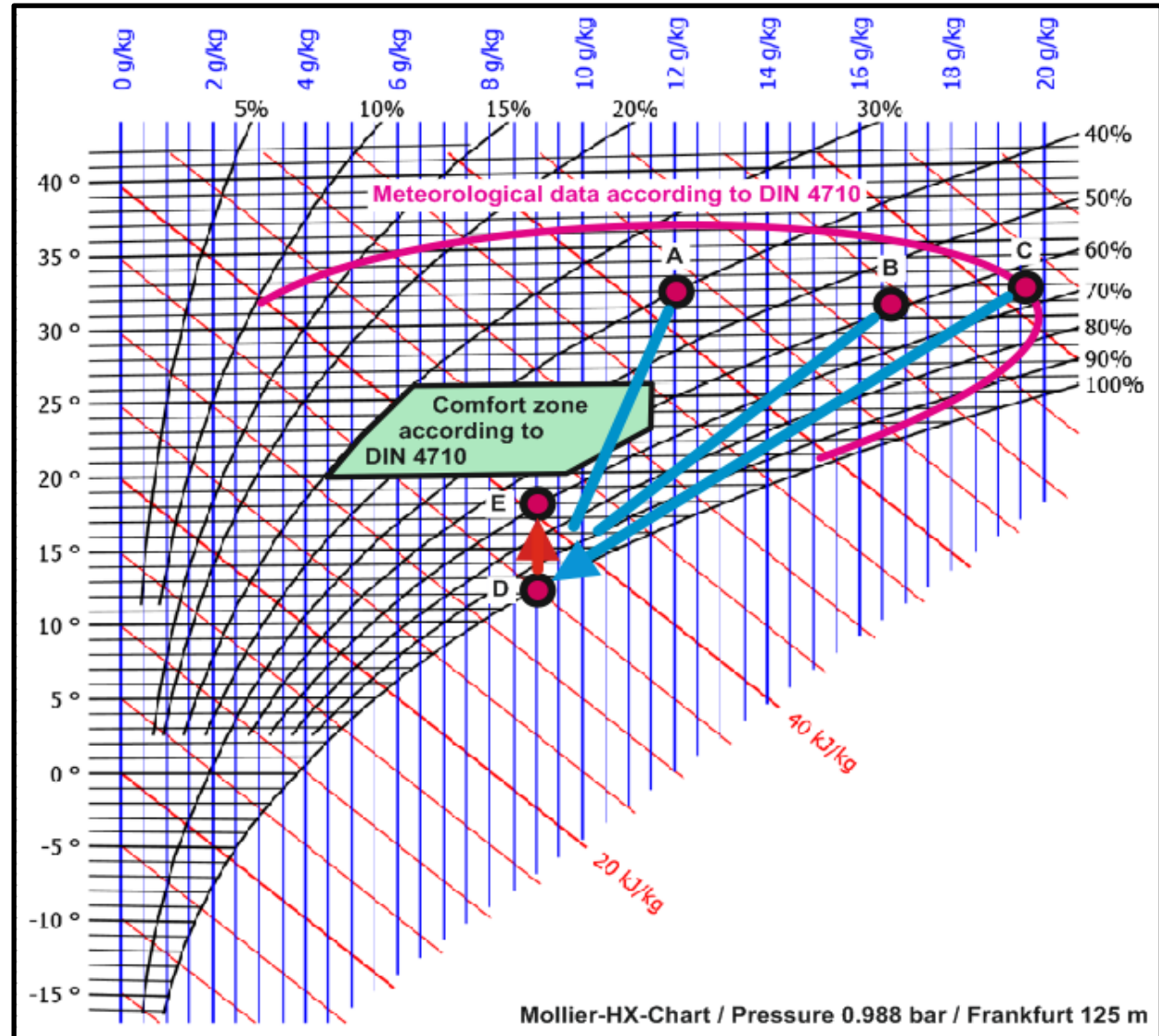
C - D: No risk management

$82.9 - 35.2 = 47.7 \text{ kJ/kg} = 100.00 \%$

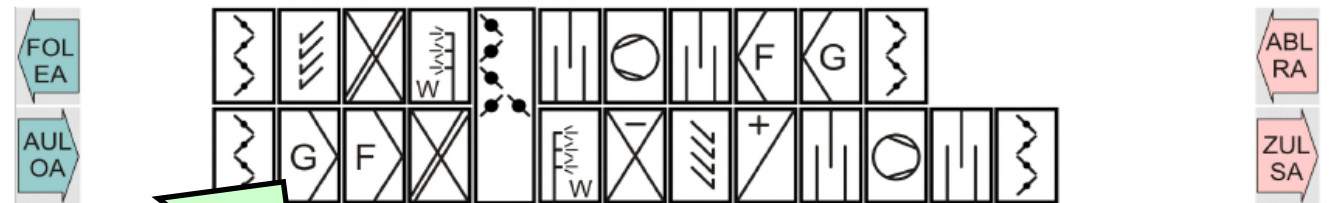
Correct cooler size

Summary

The correct cooler calculation does not depend on the highest summer temperature but on the highest enthalpy in the summer.



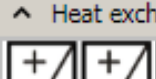
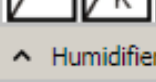
As a result, all relevant data is obtain for the two air-handling units and this at a time expenditure of just a few minutes.

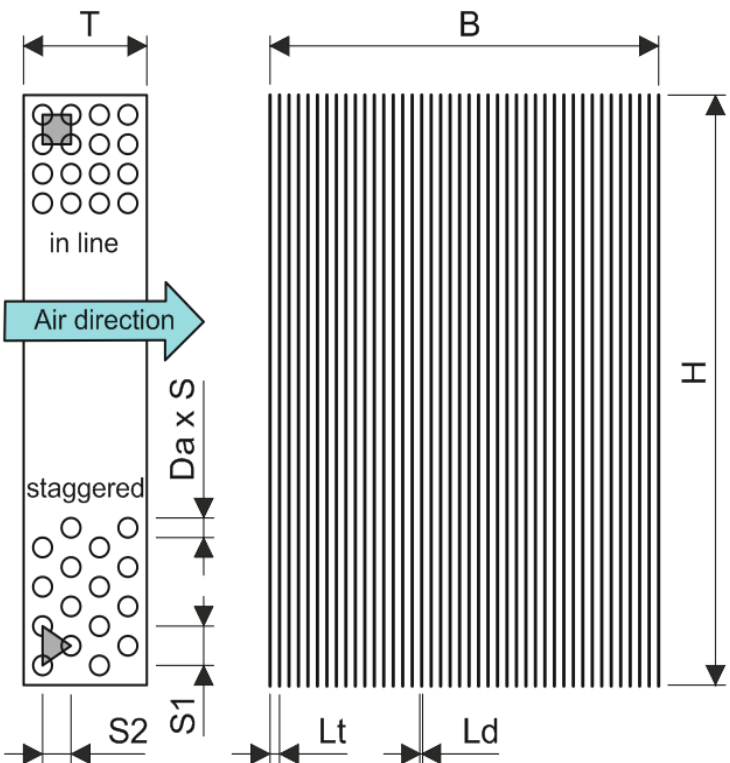
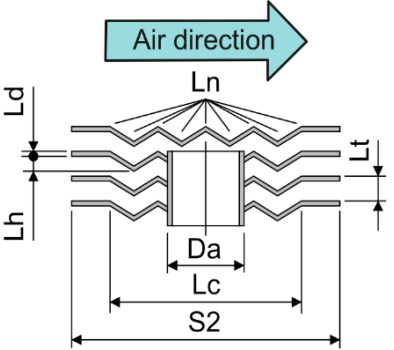
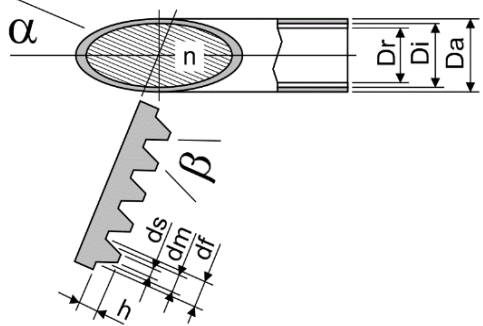


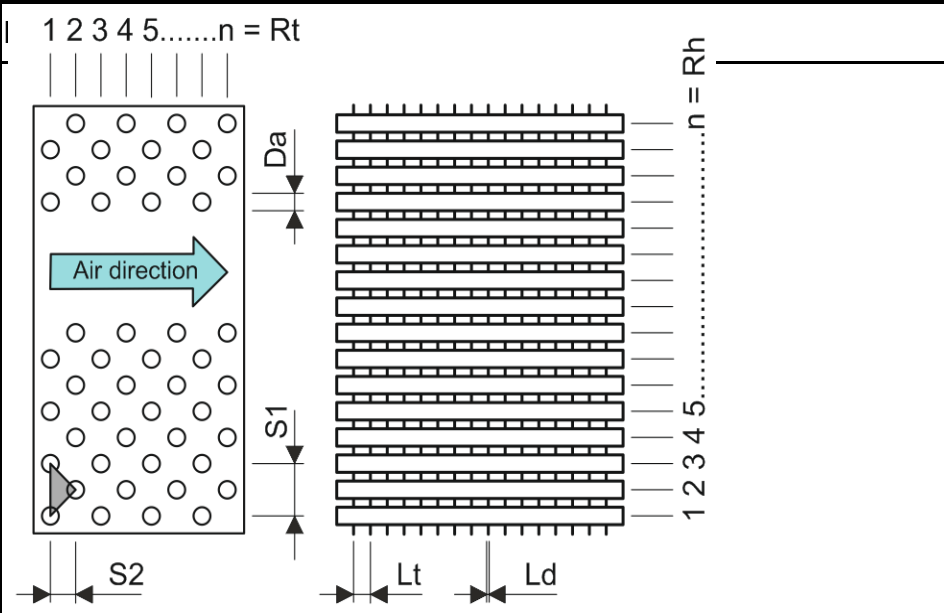
Air-Handling Unit Outside air (7)	= 1960 x 2570 mm) - Filter 1.87)	Length mm	Weight kg	Pressure drop Pa	Price EUR
Outside air				100.00	
Empty part with flaps		350.00	100.00	27.00	2180.00
Filter		450.00	140.00	109.00	2290.00
Filter		650.00	210.00	144.00	3390.00
CO ₂		650.00	590.00	87.00	9250.00
Empty part big with flaps		350.00	80.00	18.00	1160.00
water		1300.00	300.00	88.00	5560.00
		400.00	320.00	63.00	5280.00
separator		150.00	70.00	88.00	1500.00
er		200.00	200.00	23.00	3270.00
and absorber		1300.00	300.00	53.00	5560.00
an - Efficiency 88.00 % - Capacity 12.01 kW		2200.00	780.00	88.00	12900.00
sound absorber		1300.00	300.00	53.00	5560.00
Empty part little with flaps		350.00	100.00	27.00	2180.00
Supply air				300.00	
Total		9650.00	3490.00	1268.00	60080.00

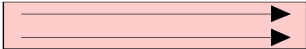
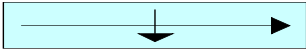
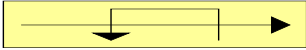
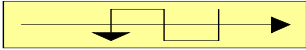
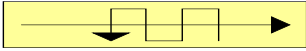
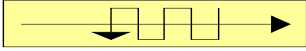
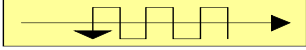
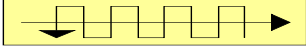
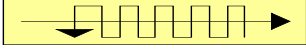
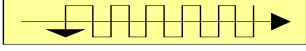
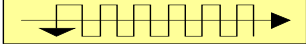
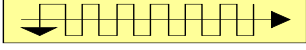
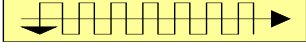
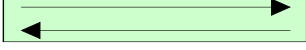
Air-Handling Unit (H x W = 1960 x 2570 mm) Return air (28000 m³/h - Filter 1.74)	Length mm	Weight kg	Pressure drop Pa	Price EUR
Return air			150.00	
Empty part little with flaps	350.00	100.00	23.00	2180.00
Filter G	450.00	140.00	95.00	2290.00
Filter F	650.00	210.00	126.00	3390.00
Sound absorber	1300.00	300.00	46.00	5560.00
Fan - Efficiency 75.00 % - Capacity 9.75 kW	2200.00	780.00	76.00	12900.00
Sound absorber	1300.00	300.00	46.00	5560.00
Empty part big with flaps	350.00	80.00	16.00	1160.00
Humidifier water	1300.00	300.00	76.00	5560.00
CC-System	650.00	590.00	87.00	9250.00
Droplet separator	150.00	70.00	76.00	1500.00
Empty part little with flaps	350.00	100.00	23.00	2180.00
Exhaust air			100.00	
Total	9050.00	2970.00	940.00	51530.00

^ HR

 ^ Heat exchanger

 ^ Humidifier


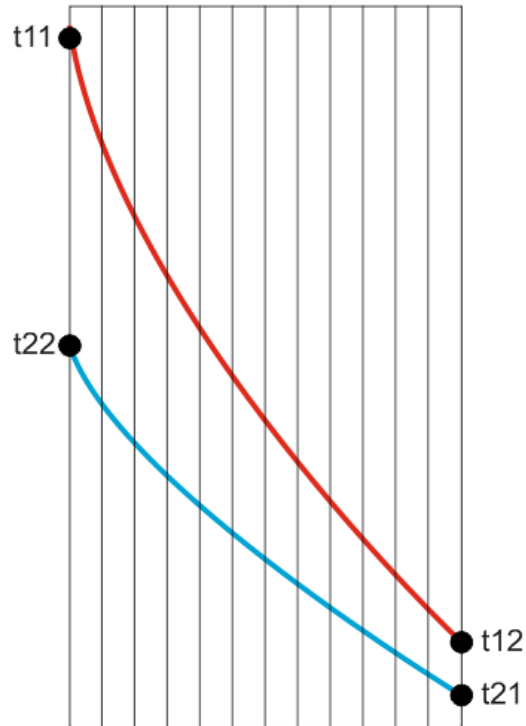
Fin coil heat exchanger	Symbol	Unit	Description	Page 11
 Corrugated fins  Heat exchanger tubes inside grooved 	α	°	Inside grooved tubes twist angle	
	B	m	Fin package width	
	β	°	Inside grooved tubes flank angle	
	Da	m	Heat exchanger tubes outside diameter	
	Di	m	Heat exchanger tubes inside diameter	
	Dr	m	Heat exchanger tubes grooves diameter	
	df	m	Grooves thickness at the foot	
	dm	m	Grooves thickness average	
	ds	m	Grooves thickness on the head	
	H	m	Fin package height	
	h	m	Grooves height	
	Lc	m	Fin corrugation depth per tube row	
	Ld	m	Fin thickness	
	Lh	m	Fin corrugation height	
	Ln	m	Number of fin creases per tube row	
	Lt	m	Fin spacing	
	n	---	Number of grooves	
	S	m	Heat exchanger tubes thickness	
	S1	m	Heat exchanger tubes interval on the height	
	S2	m	Heat exchanger tubes on the depth	
	T	m	Fin package depth	

	Symbol	Unit	Description	Page 12
 $\eta_{KRL} = 1.00$ for fin thickness ≥ 0.20 mm $\eta_{KRL} = .95$ or fin thickness ≥ 0.18 and < 0.20 mm $\eta_{KRL} = 0.90$ for fin thickness ≥ 0.16 and < 0.18 mm $\eta_{KRL} = 0.85$ for fin thickness ≥ 0.14 and < 0.16 mm $\eta_{KRL} = 0.80$ for fin thickness ≥ 0.12 and < 0.14 mm $\eta_{KRL} = 0.75$ for fin thickness ≥ 0.10 and < 0.12 mm $p = 4R[(L_t - L_d)D_a\pi + 2L_dS_1S_2]/(L_tS_1S_2)$ $q = 1 - e^{-p}$ $\eta_{BPL} = q^{0.25}$ $r = L_d\lambda_l/(A_a/A_i)$ $s = 1 - e^{-r}$ $\eta_{WRL} = s^{0.25}$ $\eta_{WTT} = \eta_{KRL}\eta_{BPL}\eta_{WRL}$ $\dot{Q} = \dot{M}_{lt}\Delta h_{lf}$ $\dot{Q} = \dot{M}\Delta h_m$ $\dot{Q} = k_a A_a \Delta t_m$ $\frac{1}{k_a} = \frac{1}{\alpha_a} + f_a + \frac{A_a \delta_w}{A_i \lambda_w} + \frac{A_a}{A_i} \frac{1}{\alpha_i} + \frac{A_a}{A_i} f_i$	A_a	m ²	Heat exchanger surface outside	
	A_i	m ²	Heat exchanger surface inside	
	α_a	W/m ² K	Heat transfer coefficient outside	
	α_i	W/m ² K	Heat transfer coefficient inside	
	Δh_{lf}	J/kg	Enthalpy difference of the moist air	
	Δh_m	J/kg	Enthalpy diff. from the heating or cooling agent	
	Δt_m	K	Average logarithmic temperature difference	
	δ_w	m	Heat exchanger tubes thickness	
	η_{BPL}	---	Bypass efficiency between the fins	
	η_{KRL}	---	Contact efficiency tube / fins	
	η_{WRL}	---	Heat transfers efficiency tube / fins	
	η_{WTT}	---	Heat exchanger efficiency total	
	f_a	m ² K/W	Fouling factor outside	
	f_i	m ² K/W	Fouling factor inside	
	k_a	W/m ² K	K-coefficient concerning the outside surface	
	λ_w	W/mK	Heat exchanger tubes thermal conductivity	
	$\dot{M}$	kg/s	Mass flow from the heating or cooling agent	
	$\dot{M}_{lt}$	kg/s	Mass flow from the dry air	
	$\dot{Q}$	W	Capacity	
	R_h	Piece	Number of tube rows on the height	
	R_t	Piece	Number of tube rows on the depth	

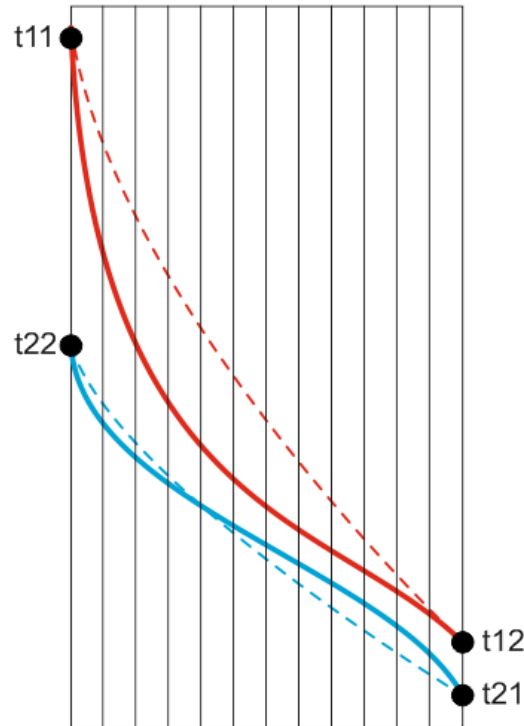
Fin coil heat exchanger	Symbol	Unit	Description	Page 13
Average logarithmic temperature difference	t_{11}	°C	Inlet temperature hot agent	
Counter flow	t_{12}	°C	Outlet temperature hot agent	
$\Delta t_1 = t_{11} - t_{22} \quad \Delta t_2 = t_{12} - t_{21} \quad \Delta t_1 = \Delta t_2 \rightarrow \Delta t_m = \Delta t_1$	t_{21}	°C	Inlet temperature cold agent	
$\Delta t_1 \neq \Delta t_2 \rightarrow \Delta t_m = (\Delta t_1 - \Delta t_2) / \ln(\Delta t_1 / \Delta t_2)$	t_{22}	°C	Outlet temperature cold agent	
Equal flow	n	Piece	Number of counter flow packages	
$\Delta t_1 = t_{11} - t_{21} \quad \Delta t_2 = t_{12} - t_{22} \quad \Delta t_1 = \Delta t_2 \rightarrow \Delta t_m = \Delta t_1$	i	---	Cell number	
$\Delta t_1 \neq \Delta t_2 \rightarrow \Delta t_m = (\Delta t_1 - \Delta t_2) / \ln(\Delta t_1 / \Delta t_2)$			Equal	
Cross flow			Cross flow 1-time	
$p = (t_{11} - t_{12}) / (t_{11} - t_{21}) \quad q = (t_{22} - t_{21}) / (t_{11} - t_{21})$			Cross counter flow 2-times	
Cross flow 1-time			Cross counter flow 3-times	
$r = 1 - (q/p) \ln(1/(1-p)) \quad s = q / \ln(1/r)$			Cross counter flow 4-times	
$\Delta t_m = s(t_{11} - t_{21})$			Cross counter flow 5-times	
Cross counter flow 2-times			Cross counter flow 6-times	
$p = q \rightarrow x = 1 - p / (2(p - 1))$			Cross counter flow 7-times	
$p \neq q \rightarrow x = \frac{\sqrt{(1-q)/(1-p)} - q/p}{1 - q/p}$			Cross counter flow 8-times	
$r = 1 - (q/p) \ln x \quad s = q / (2 \ln(1/r)) \quad \Delta t_m = s(t_{11} - t_{21})$			Cross counter flow 9-times	
Cross counter flow n-times			Cross counter flow 10-times	
$\Delta t_m = \dot{Q} / \sum_{i=1}^n k_{a(i)} A_{a(i)}$			Cross counter flow 11-times	
			Cross counter flow 12-times	
			Cross counter flow 13-times	
			Cross counter flow 14-times	
			Counter flow	

Fin coil heat exchanger	Symbol	Unit	Description	Page 14
Cross counter flow n-times	A_l	m ²	Heat exchanger fin surface	

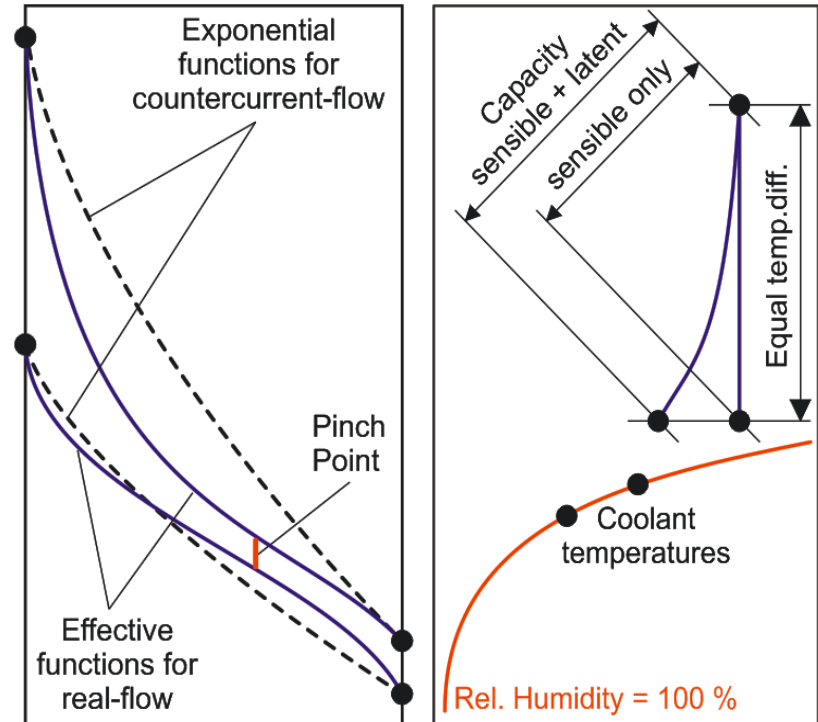
Cooling process sensible only
Temperature profile exponential
Optimal temperature difference
Cells $i = 1, 2, 3, 4, 5, \dots, n$



Cooling process sensible & latent
Temperature profile deformed
Reduced temperature difference
Cells $i = 1, 2, 3, 4, 5, \dots, n$



Heat exchanger manufacturers, which do not calculate so and specify the Δt_m for countercurrent, constantly supply to small air coolers, which is still too often the case, one hopes nevertheless cynically that it will not be measured.



Smooth fins

$$A_l = 2 \left(S_1 S_2 - \frac{D_a^2 \pi}{4} \right) R_t R_h \left(\frac{B}{L_t} - 1 \right)$$

Corrugated fins

$$\xi_a = \arctan \left(\frac{L_h (L_n - 1)}{L_c} \right) \frac{180}{\pi} \quad f_a = \frac{(S_2 + L_c) + \left(\frac{L_c}{\cos(\xi_a \pi / 180)} \right)}{S_2}$$

$$A_l = 2 f_a \left(S_1 S_2 - \frac{D_a^2 \pi}{4} \right) R_t R_h \left(\frac{B}{L_t} - 1 \right)$$

Flat or marginal corrugated fins, combined with tubes in line, show the smallest air pressure drops on the air side. Highly corrugated fins, combined with staggered tubes, show the highest air pressure drops on the air side. Manufacturers who offer strongly corrugated fins mostly have too thin fins. They only need heavily corrugated fins to add just a bit of stability. The claim that they are doing it for reasons of increasing performance, one can confidently deny.

Fin coil heat exchanger	Symbol	Unit	Description	Page 15
Heat exchanger surface outside	A_r	m ²	Heat exchanger surface tubes outside	
$A_r = D_a \pi (L_t - L_d) R_t R_h \left(\frac{B}{L_t} - 1 \right)$ $A_a = A_r + A_l$	f_h	---	Factor for the heat transfer	
	f_{dp}	---	Factor for the pressure drop	

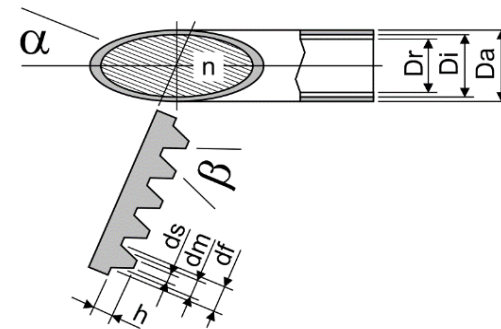
Heat exchanger surface inside with smooth tubes

$$A_i = D_i \pi B R_t R_h$$

Heat exchanger surface inside with grooved tubes

$$d_m = \frac{(D_i + D_r) \pi}{4n} \quad A_{r1} = n \left[\left(\frac{2h}{\cos\left(\frac{\beta\pi}{360}\right)} \right) + \left(d_m - h \tan\left(\frac{\beta\pi}{360}\right) \right) \right]$$

$$A_{r2} = A_{r1} + \left[n \left(d_m \left(h \tan\left(\frac{\beta\pi}{360}\right) \right) \right) \right] \quad A_i = A_{r2} B R_t R_h$$

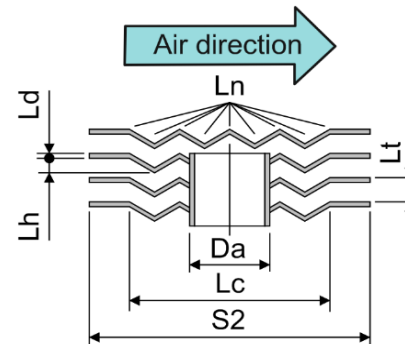


Corrugated fins

Influence on the heat transfer and the pressure drop

$$f_x = 1 + 0.3 \sin\left(\frac{L_n \pi}{180}\right) + 0.2 \left(\sin\left(\frac{L_n \pi}{180}\right) \right)^2 \quad f_y = (L_n + 1)^{0.05}$$

$$f_b = f_x f_y \quad f_e = \frac{L_h}{L_t - L_d} \quad f_h = f_b^{f_e} \quad f_{dp} = f_h^{L_n}$$



Heat transfer coefficient outside

The equations of the VDI-Atlases of the years 1955 to 1984 supply too high, the equations of the years 1985 to 1997 too low values for the heat transfer coefficient of the moist air. This fact is positive, because it allows calibrating based on laboratory measurements. Thus studies with TUEV showed that the mean value of the two approaches, so a relation of 1:1, resulted in a match with the measurements. VDI-Atlases of recent date are useless because various authors have applied double integrals from zero to infinity and similar impracticable mathematical excesses.

Fin coil heat exchanger	Symbol	Unit	Description	Page 16
Heat transfer coefficient outside (VDI 1955 to 1984) $a = \frac{S_1}{D_a} \quad b = \frac{S_2}{D_a} \quad \psi = \left(1 - \frac{\pi}{4a}\right) \frac{L_t - L_d}{L_t} \quad f_{a(staggered)} = 1 + \frac{2}{3b}$ $f_{a(in\ line)} = 1 + \frac{0.7 \left(\frac{b}{a} - 0.3\right)}{\psi^{1.5} \left(\frac{b}{a} + 0.7\right)^2} \quad f_k = \frac{1 + f_a(R_t - 1)}{R_t}$ $l_{hyd} = \frac{D_a \pi}{2} \quad V_{lfm} = \frac{\dot{M}_{lt}(1 + x_{lfm})}{\rho_{lfm}} \quad c_l = \frac{V_{lfm}}{BH}$ $Re_l = \frac{c_l l_{hyd} \rho_{lfm}}{\psi \eta_{lfm}} \quad cp_{lfm} = \frac{\Delta h_{lf}}{\Delta t_{lf}} \quad Pr_l = \frac{\eta_{lfm} cp_{lfm}}{\lambda_{lfm}}$	c_l cp_{lfm} Δt_{lf} η_{lfm} λ_l λ_{lfm} ρ_{lfm} V_{lfm} x_{lfm}	m/s J/kgK K Pas W/mK W/mK kg/m3 m3/s kg/kg	Average velocity of the moist air Average heat capacity of the moist air Temperature difference of the moist air Average dynamic viscosity of the moist air Thermal conductivity of the fins Average thermal conductivity of the moist air Average density of the moist air Average volume flow of the moist air Average absolute humidity of the moist air	
$Nu_1 = 0.644 Re_l^{0.5} Pr_l^{1/3} \quad Nu_2 = \frac{0.037 Re_l^{0.8} Pr_l}{1 + 2.443 Re_l^{-0.1} (Pr_l^{2/3} - 1)}$ $\rho_{me} = 1.28 \frac{S_2}{D_a} \sqrt{\frac{S_1}{S_2} - 0.2} \quad h_{me} = \frac{D_a}{2} (\rho_{me} - 1) (1 + 0.35 \ln \rho_{me}) \quad X = h_{me} \sqrt{\frac{2\alpha_l}{\lambda_l L_d}}$			$Nu_3 = 0.3 + \sqrt{Nu_1^2 + Nu_2^2} \quad Nu_l = f_k Nu_3 \quad \alpha_l = \frac{f_h Nu_l \lambda_{lfm}}{l_{hyd}}$ $\vartheta = \frac{Tanh X}{X} \eta_{KRL} \quad \eta_l = 1 - \frac{A_l}{A_a} (1 - \vartheta) \quad \alpha_{a(old)} = \eta_l \alpha_l$	
Heat transfer coefficient outside (VDI 1985 to 1997) $a = \frac{S_1}{D_a} \quad b = \frac{S_2}{D_a} \quad \psi = \frac{S_1 L_t}{(S_1 - D_a)(L_t - L_d)} \quad c_l = \frac{\psi V_{lfm}}{BH}$ $\vartheta = \frac{2 \left((S_1 S_2) - (D_a^2 \pi / 4) \right) + (D_a \pi (L_t - L_d))}{D_a \pi L_t} \quad R_t \leq 3 \rightarrow f_{a(in\ line)} = 0.2 \quad R_t > 3 \rightarrow f_{a(in\ line)} = 0.22$ $Re_l = \frac{c_l D_a \rho_{lfm}}{\eta_{lfm}} \quad Pr_l = \frac{\eta_{lfm} cp_{lfm}}{\lambda_{lfm}}$ $R_t \leq 2 \rightarrow f_{a(staggered)} = 0.33 \quad R_t = 3 \rightarrow f_{a(staggered)} = 0.36 \quad R_t > 3 \rightarrow f_{a(staggered)} = 0.38 \quad Nu_l = f_a Re_l^{0.6} \vartheta^{-0.15} Pr_l^{(1/3)}$ $\alpha_l = \frac{f_h Nu_l \lambda_{lfm}}{D_a} \quad l_r = \sqrt{\left(\frac{S_1}{2}\right)^2 + S_2^2} \quad S_2 < \frac{S_1}{2} \rightarrow \rho_{me(staggered)} = 1.27 \frac{2S_2}{D_a} \sqrt{\frac{l_r}{2S_2} - 0.3} \quad S_2 \geq \frac{S_1}{2} \rightarrow \rho_{me(staggered)} = 1.27 \frac{S_1}{D_a} \sqrt{\frac{l_r}{S_1} - 0.3}$				

Fin coil heat exchanger	Symbol	Unit	Description	Page 17
$S_1 < S_2 \rightarrow \rho_{me(in\ line)} = 1.28 \frac{S_1}{D_a} \sqrt{\frac{S_2}{S_1}} - 0.2$	$f_{w(old)}$	---	Factor for the heat transfer coefficient VDI old	
$S_1 \geq S_2 \rightarrow \rho_{me(in\ line)} = 1.28 \frac{S_2}{D_a} \sqrt{\frac{S_1}{S_2}} - 0.2$	$f_{w(new)}$	---	Factor for the heat transfer coefficient VDI new	
$h_{me} = \frac{D_a}{2} (\rho_{me} - 1) (1 + 0.35 \ln \rho_{me})$	k_g	---	Factor for the surface temperature	
$X = h_{me} \sqrt{\frac{2\alpha_l}{\lambda_l L_d}}$	t_l	°C	Temperature of the moist air	
$\vartheta = \frac{Tanh X}{X} \eta_{KRL}$	t_m	°C	Temperature of the coolant agent	
$\eta_l = 1 - \frac{A_l}{A_a} (1 - \vartheta)$	t_o	°C	Average surface temperature outside	
$\alpha_{a(new)} = \eta_l \alpha_l$				

Heat transfer coefficient outside on average

$$\alpha_a = \frac{f_{w(old)} \alpha_{a(old)} + f_{w(new)} \alpha_{a(new)}}{f_{w(old)} + f_{w(new)}}$$

k-coefficient concerning the outside surface

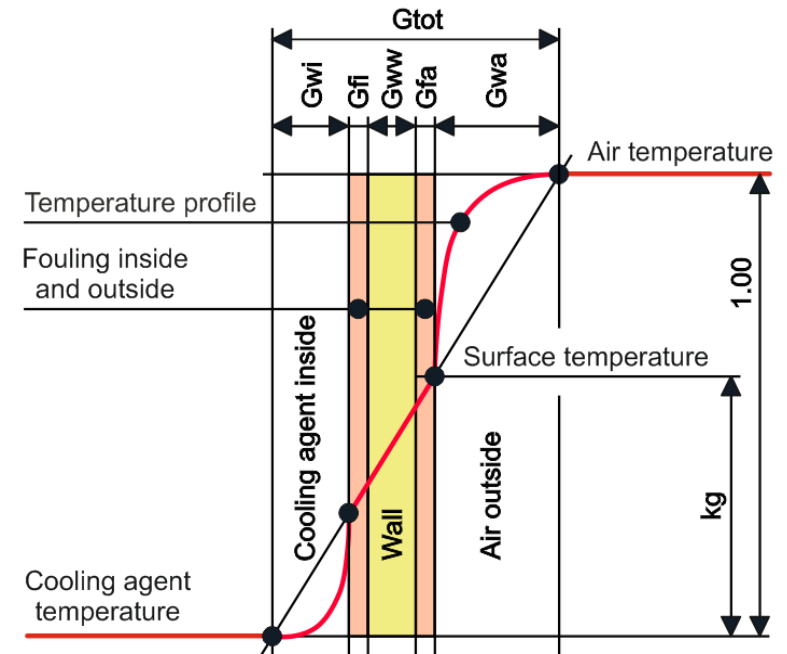
$$G_{wa} = \frac{1}{\alpha_a} \quad G_{fa} = f_a \quad G_{ww} = \frac{A_a \delta_w}{A_i \lambda_w} \quad G_{wi} = \frac{A_a}{A_i} \frac{1}{a_i} \quad G_{fi} = \frac{A_a}{A_i} f_i$$

$$G_{tot} = G_{wa} + G_{fa} + G_{ww} + G_{wi} + G_{fi} \quad k_a = \frac{1}{G_{tot}}$$

Air cooler surface temperature of the moist air

$$\eta_{WTT} = 1.00 \rightarrow k_g = \frac{G_{tot} - G_{wa}}{G_{tot}} \quad \eta_{WTT} < 1.00 \rightarrow m = \eta_{WTT}^4 \rightarrow k_g = \left(\frac{G_{tot} - G_{wa}}{G_{tot}} \right)^m$$

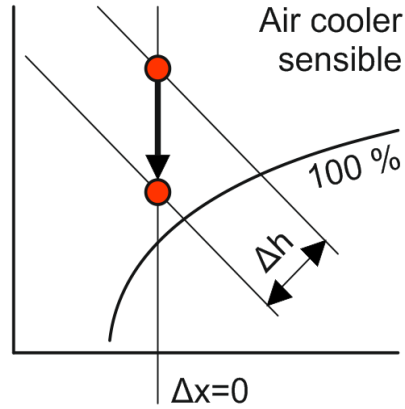
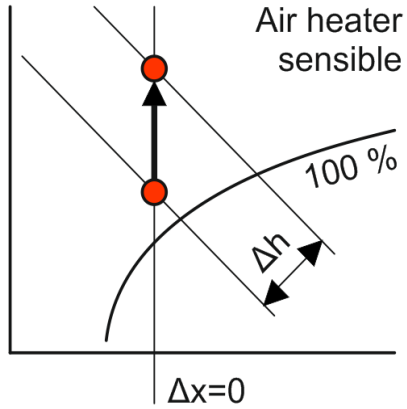
$$t_o = t_m + k_g (t_l - t_m)$$



The poorer the total heat exchanger efficiency is the higher is the surface temperature of the air. This is true for much too thin fins with poor contact to the tubes, for a few tubes in the depth and for large fins distribution due to the bypass effect. Such air coolers have too little latent power and can therefore hardly form condensate.

Pressure drop outside

$\Delta x = 0 \rightarrow$ Air heater and air cooler sensible



Δp_R

Pa

Pressure drop from the tubes

Δp_L

Pa

Pressure drop from the fins

$\Delta p_{l(dry)}$

Pa

Pressure drop from the air (dry)

$$c_l = \frac{V_{lfm}}{BH}$$

$$a = \frac{S_1}{D_a}$$

$$b = \frac{S_2}{D_a}$$

$$\Omega = \frac{S_1 - D_a L_t - L_d}{S_1 L_t}$$

$$Re_l = \frac{c_l D_a \rho_{lfm}}{\Omega \eta_{lfm}}$$

$$A_1 = \frac{280\pi (0.75 + (b^{0.5} - 0.6)^2)}{1.6 (1.4 \pi)}$$

$$\xi_1 = \frac{A_1}{Re_l}$$

$$A_{2(staggered)} = 2.5 + \frac{1.2}{(a - 0.85)^{1.08}} + 0.4 \left(\frac{b}{a} - 1 \right)^3 - 0.01 \left(\frac{a}{b} - 1 \right)^3$$

$$\xi_{2(staggered)} = \frac{A_2}{Re_l^{0.25}} + \frac{1}{a^2} \left(\frac{1}{R_t} - 0.1 \right)$$

$$\xi_{R(staggered)} = \xi_1 + \xi_2 \left(1 - e^{-\left(\frac{Re_l + 200}{1000} \right)} \right)$$

$$A_{2(in line)} = 0.03(a - 1)(b - 1) + \left(0.22 + 1.2 \frac{\left(1 - \frac{0.94}{b} \right)^{0.6}}{(a - 0.85)^{1.3}} \right) 10^{\left(\frac{b}{a} - 1.5 \right)}$$

$$\xi_{2(in line)} = \frac{A_2}{Re_l^{0.1} \left(\frac{b}{a} \right)} + \frac{1}{a^2} \left(\frac{1}{R_t} - 0.1 \right)$$

$$\xi_{R(in line)} = \xi_1 + \xi_2 \left(1 - e^{-\left(\frac{Re_l + 1000}{2000} \right)} \right)$$

$$\Delta p_R = \xi_R R_t \rho_l \frac{c_l^2}{2\Omega^2}$$

$$d_{hyd} = 1.8(L_t - L_d) + 0.1D_a$$

$$Re_l = \frac{c_l d_{hyd} \rho_{lfm}}{\Omega \eta_{lfm}}$$

$$\delta = \frac{L_t - L_d}{S_1 - D_a}$$

$$\beta = 0.84 + 0.66e^{-\frac{\delta}{0.33}}$$

$$r_l = 0.000078$$

$$\xi_1 = \beta \frac{64}{Re_l}$$

$$\xi_2 = \left(2 \log \left(\frac{d_{hyd}}{r_l} \right) + 1.14 \right)^{-2}$$

$$\xi_3 = \left(-2 \log \left(\frac{2.51}{Re_l \sqrt{\xi_3}} + \frac{r_l}{3.71 d_{hyd}} \right) \right)^{-2}$$

$$\xi_4 = \max(\xi_1, \xi_2, \xi_3)$$

$$\xi_{L(staggered)} = 0.25 + \frac{2}{\sqrt{3}} \frac{\xi_4 S_2}{d_{hyd}}$$

$$\xi_{L(in line)} = 0.25 + \frac{\xi_4 S_2}{d_{hyd}}$$

$$\Delta p_L = \xi_L R_t \rho_l \frac{c_l^2}{2\Omega^2} f_{dp}$$

$$\Delta p_{l(dry)} = \Delta p_R + \Delta p_L$$

Fin coil heat exchanger

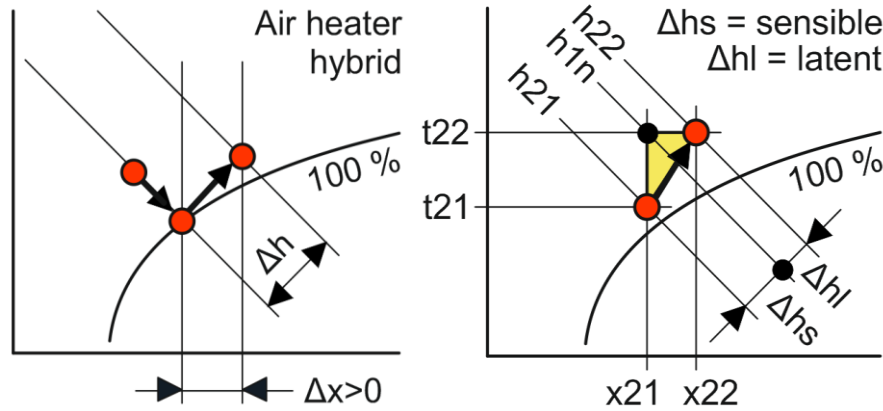
Symbol

Unit

Description

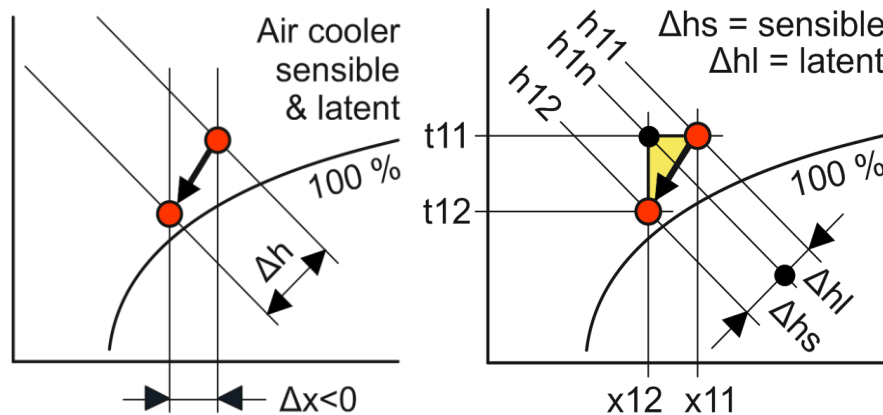
Page 19

$\Delta x > 0 \rightarrow$ Air heater hybrid (sensible and latent)



The equations are identical to those for sensitive air heaters and air coolers to and with the pressure drop of the air dry, see page 18. Now the calculation for the pressure drop with sprayed fins follows.

$\Delta x < 0 \rightarrow$ Air cooler sensible and latent



cp_{km}

J/kgK

Average heat capacity of the condensate

$\Delta h_{lf(sen)}$

J/kg

Enthalpy difference of the moist air (sensible)

$\Delta p_{l(moi)}$

Pa

Pressure drop of the moist air

η_{km}

Pas

Average dynamic viscosity of the condensate

$f_{l(moi)}$

Pressure drop factor (moist)

f_n

Factor for cells with condensate formation

λ_{km}

W/mK

Average thermal conductivity from the condensate

n

First from 15 cells with condensate formation

ρ_{km}

kg/m3

Average density from the condensate

$$\Delta h_{lf(sen)} = h_{1n} - h_{21} \quad f_{l(moi)} = \left(\frac{\Delta h_{lf}}{\Delta h_{lf(sen)}} \right)^{0.33} \quad \Delta p_{l(moi)} = f_{l(moi)} \Delta p_{l(dry)}$$

$$f_n = \frac{16 - n}{15}$$

$$f_u = 0.04 f_n (1 - e^{-\Delta x / 0.004})$$

$$f_v = \frac{\Delta x / \rho_{km}}{\left(((1 - \Delta x) / \rho_{lfm}) + (\Delta x / \rho_{km}) \right)}$$

$$\rho_{lfm} = \rho_{lfm}(1 - f_v) + \rho_{km} f_v$$

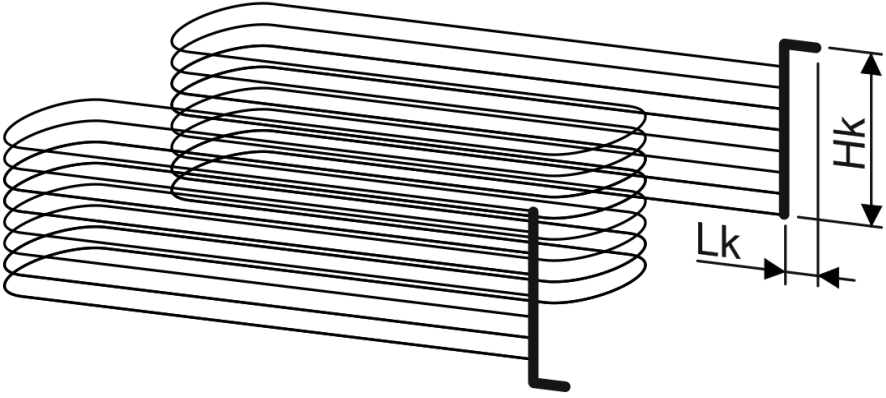
$$cp_{lfm} = cp_{lfm}(1 - \Delta x) + cp_{km} \Delta x$$

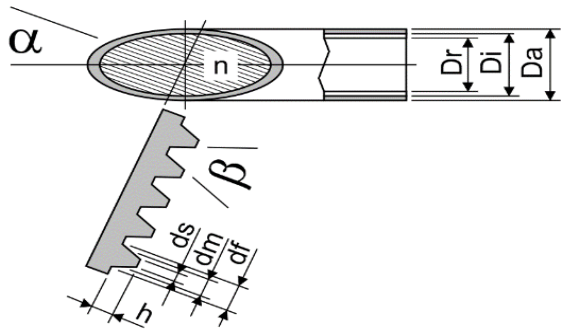
$$\lambda_{lfm} = \lambda_{lfm}(1 - f_u) + \lambda_{km} f_u$$

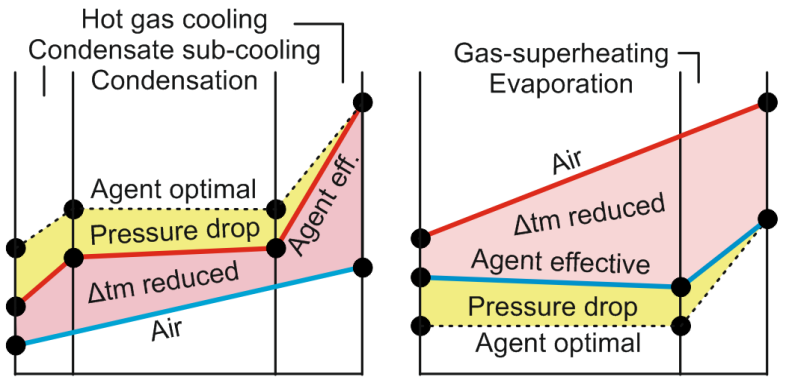
$$\eta_{lfm} = \eta_{lfm}(1 - \Delta x) + \eta_{km} \Delta x$$

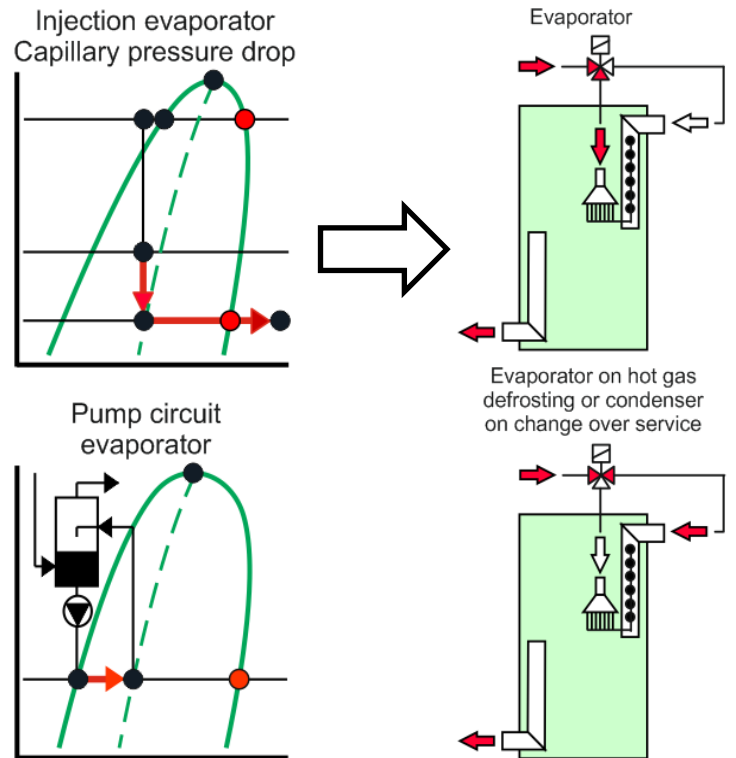
With the corrected material values (moist air with condensate), the equations for sensitive air heaters and air coolers can be applied to and with the pressure drop of the air dry, see page 18. Now the calculation for the pressure drop with condensate on the fins follows.

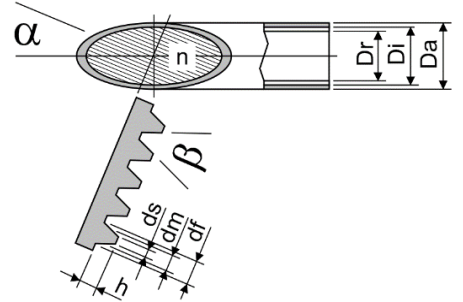
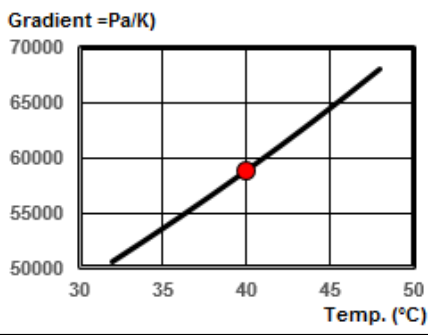
$$\Delta h_{lf(sen)} = h_{1n} - h_{12} \quad f_{l(moi)} = \left(\frac{\Delta h_{lf}}{\Delta h_{lf(sen)}} \right)^{0.33} \quad \Delta p_{l(moi)} = f_{l(moi)} \Delta p_{l(dry)}$$

Fin coil heat exchanger	Symbol	Unit	Description	Page 20
Definition to the tube coupling and to the collectors Example Number of circuits (NC) = 8 Number of passages per NC (PA) = 4	c_{km} cp_m D_{ki} Δp_k η_m λ_m NC PA Q_{ki} r_k ρ_m	m/s J/kgK m Pa Pas W/mK --- --- m ² m kg/m ³	Agent velocity inside of the collector Agent heat capacity Collector inside diameter Collector pressure drop inside Agent dynamic viscosity Agent thermal conductivity Number of circuits Number of passages per NC Collector cross section inside Roughness inside of the collector Agent density	
				
Pressure drop in the collectors $\Delta h_m = cp_m \Delta t_m \quad \dot{M} = \frac{\dot{Q}}{\Delta h_m} \quad \dot{V} = \frac{\dot{M}}{\rho_m} \quad Q_{ki} = \frac{D_{ki}^2 \pi}{4} \quad c_{km} = \frac{\dot{V}}{Q_{ki}} \quad Re_{km} = \frac{c_{km} D_{ki} \rho_m}{\eta_m}$ $r_{k(Copper)} = 0.000002 \quad r_{k(Stainless\ steel)} = 0.000080 \quad r_{k(Galvanized\ steel)} = 0.000160 \quad \xi_1 = \frac{64}{Re_{km}} \quad \xi_2 = 0.3164 Re_{km}^{-0.25}$ $\xi_3 = 0.0054 + 0.3964 (Re_{km}^{-0.3}) \quad \xi_4 = \left(\log (Re_{km} \sqrt{\xi_4}) \right)^{-2} \quad \xi_5 = \left(2 \log \left(\frac{D_{ki}}{r_{km}} \right) + 1.14 \right)^{-2} \quad \xi_6 = \left(-2 \log \left(\frac{2.51}{Re_{km} \sqrt{\xi_6}} + \frac{r_{km}}{3.71 D_{ki}} \right) \right)^{-2}$ $Re_{km} < 100 \rightarrow \xi_7 = \xi_1 \quad Re_{km} \geq 100 \rightarrow \xi_7 = \max(\xi_1, \xi_2, \xi_3, \xi_4, \xi_5, \xi_6) \quad \xi_{km} = \frac{\xi_7 (H_k + L_k)}{D_{ki}} + 3 \quad \Delta p_k = \xi_{km} \rho_m \frac{c_{km}^2}{2}$				

Fin coil heat exchanger	Symbol	Unit	Description	Page 21
Definition to the smooth tubes $Q_{ri} = NC \frac{D_i^2 \pi}{4} \quad D_{hyd} = D_i \quad r_{r(Copper)} = 0.000002$ $r_{r(Stainless\ steel)} = 0.000080 \quad r_{r(Galvanized\ steel)} = 0.000160$	Q_{ri} D_{hyd} Δp_r Δp_{ti}	m2 m Pa Pa	Tubes cross section inside total Hydraulic diameter from the tubes inside Pressure drop in the tubes inside Pressure drop total inside	
Definition to the inside grooved tubes $Q_{ri} = NC \frac{(D_i + D_r)^2 \pi}{16} \quad D_{hyd} = \frac{4Q_{ri}}{A_{r2}}$ $r_{rr(Copper)} = 0.000002$ $r_{rr(Stainless\ steel)} = 0.000080$ $r_{rr(Galvanized\ steel)} = 0.000160 \quad r_r = r_{rr} + h \sin\left(\frac{\alpha \pi}{180}\right)^4$				
Pressure drop in the tubes for agents without aggregate state change $c_{rm} = \frac{\dot{V}}{Q_{ri}} \quad Re_{rm} = \frac{c_{rm} D_{hyd} \rho_m}{\eta_m} \quad \xi_1 = \frac{64}{Re_{rm}} \quad \xi_2 = 0.3164 Re_{rm}^{-0.25} \quad \xi_3 = 0.0054 + 0.3964 (Re_{rm}^{-0.3}) \quad \xi_4 = \left(\log \left(Re_{rm} \sqrt{\xi_4} \right) \right)^{-2}$ $\xi_5 = \left(2 \log \left(\frac{D_{hyd}}{r_r} \right) + 1.14 \right)^{-2} \quad \xi_6 = \left(-2 \log \left(\frac{2.51}{Re_{rm} \sqrt{\xi_6}} + \frac{r_r}{3.71 D_{hyd}} \right) \right)^{-2} \quad \xi_7 = \max(\xi_1, \xi_2, \xi_3, \xi_4, \xi_5, \xi_6) \quad \xi_{rm} = \frac{PA \xi_7 B}{D_{hyd}} + (PA - 1)$ $\Delta p_r = \xi_{rm} \rho_m \frac{c_{rm}^2}{2}$				
Pressure drop inside total from the heat exchanger $\Delta p_{ti} = \Delta p_r + \Delta p_{k(Inlet\ collector)} + \Delta p_{k(Outlet\ collector)}$				

Fin coil heat exchanger	Symbol	Unit	Description	Page 22
Heat transfer coefficient inside	η_{liq}	Pas	Agent viscosity on Liquidus	
Agents without aggregate state change	g	m/s2	Fall acceleration (9.81 m/s2)	
This relates to liquids or gases, so also for condensers regarding hot-gas cooling and condensate sub-cooling and for injection evaporators regarding the suction gas superheat.	λ_{liq}	W/mK	Agent thermal conductivity on Liquidus	
	ρ_{liq}	kg/m3	Agent density on Liquidus	
	R	J/kg	Agent evaporation heat	
$Pr_m = \frac{\eta_m c p_m}{\lambda_m} \quad \xi_m = (1.82 \log(Re_{rm} - 1.64))^{-2}$ $Nu_1 = \frac{\frac{\xi_m}{8} (Re_m - 1000) Pr_m}{1 + 12.7 \left(Pr_m^{\frac{2}{3}} - 1 \right) \sqrt{\frac{\xi_m}{8}}} \quad Nu_2 = \sqrt[3]{3.66^3 + 1.61^3 Re_{rm} Pr_{rm} \frac{D_{hyd}}{B}} \quad Nu_3 = 0.664 Pr_m^{\frac{1}{3}} \sqrt{\frac{Re_m D_{hyd}}{B}} \quad Nu_m = \max(Nu_1, Nu_2, Nu_3)$ $\alpha_{i(Smooth tubes)} = \frac{Nu_m \lambda_m}{D_{hyd}}$				
Agents with aggregate state change (Boris Slipcevic)				
Condensation	$f_{lam} = 0.943 \left(\frac{\lambda_{liq}^3 \rho_{liq}^2 R g}{\eta_{liq} B} \right)^{0.25} \quad g i_{lam} = \frac{A_a}{A_i} \frac{1}{\alpha_{i(lam)}} \quad g t_{lam} = \frac{1}{k_a} \quad \Delta t_{lam} = \frac{g i_{lam}}{g t_{lam}} \quad \alpha_{i(lam)} = f_{lam} \Delta t_{lam}^{0.25}$ $f_{turb} = 0.003 \left(\frac{\lambda_{liq}^3 \rho_{liq}^2 B g}{\eta_{liq}^3 R} \right)^{0.5} \quad g i_{turb} = \frac{A_a}{A_i} \frac{1}{\alpha_{i(turb)}} \quad g t_{turb} = \frac{1}{k_a} \quad \Delta t_{turb} = \frac{g i_{turb}}{g t_{turb}} \quad \alpha_{i(turb)} = f_{turb} \Delta t_{turb}^{0.5}$ $\alpha_{i(Smooth tubes)} = \max(\alpha_{i(lam)}, \alpha_{i(turb)})$			

Fin coil heat exchanger	Symbol	Unit	Description	Page 23
Dry expansion evaporation $K_1 = \frac{\lambda_{liq} \rho_{liq}^{0.06}}{g^{0.3} \rho_{sol}^{0.66} \eta_{liq}^{0.575} \eta_{sol}^{0.225}} \quad B_1 = \sqrt{\frac{2S_t}{g(\rho_{liq} - \rho_{sol})}}$ $D_1 = 0.511B_1 \quad F_1 = 0.56 \sqrt{\frac{g}{D_1}} \quad Q_i = \frac{\dot{Q}}{A_i} \quad H_d = R(1 - F_{ge})$ $B_2 = \frac{2.059 \lambda_{liq}^{0.6} H_d^{0.133} r_r^{0.133} \rho_{sol}^{0.133}}{g^{0.2} (t_o + 273.16)^{0.4} S_t^{0.3} F_1^{0.266} D_1^{0.399} \rho_{liq}^{0.233}}$ $\dot{M} = \frac{\dot{Q}}{H_d} \quad \dot{m} = \frac{\dot{M}}{Q_{ri}} \quad \alpha_{i(kon)} = \frac{0.9(1 - F_{ge})^{0.1} K_1 \dot{m}^{1.4}}{D_{hyd}^{0.5}}$ $\alpha_{i(bub)} = \frac{0.9(1 - F_{ge})^{0.1} B_2 \dot{m}^{0.1} Q_i^{0.7}}{D_{hyd}^{0.5}} \quad \alpha_{i(Smooth tubes)} = \max(\alpha_{i(kon)}, \alpha_{i(bub)})$	η_{sol} F_{ge} H_d $\dot{m}$ P_u ρ_{sol} S_t t_o	Pas --- J/kg kg/sm2 --- kg/m3 N/m °C	Agent viscosity on Solidus Agent flashgas part at the inlet point Agent enthalpy difference Agent mass flow density Agent pump recirculation factor Agent density on Solidus Agent surface tension Evaporation temperature on Solidus	
Pump recirculation evaporation $K_1 = \frac{\lambda_{liq} \rho_{liq}^{0.06}}{g^{0.3} \rho_{sol}^{0.66} \eta_{liq}^{0.575} \eta_{sol}^{0.225}} \quad B_1 = \sqrt{\frac{2S_t}{g(\rho_{liq} - \rho_{sol})}} \quad D_1 = 0.511B_1$ $F_1 = 0.56 \sqrt{\frac{g}{D_1}} \quad Q_i = \frac{\dot{Q}}{A_i} \quad H_d = \frac{R}{P_u} \quad \dot{M} = \frac{\dot{Q}}{H_d} \quad \dot{m} = \frac{\dot{M}}{Q_{ri}}$ $B_2 = \frac{2.059 \lambda_{liq}^{0.6} H_d^{0.133} r_r^{0.133} \rho_{sol}^{0.133}}{g^{0.2} (t_o + 273.16)^{0.4} S_t^{0.3} F_1^{0.266} D_1^{0.399} \rho_{liq}^{0.233}} \quad \alpha_{i(kon)} = \frac{K_1 \dot{m}^{1.4}}{D_{hyd}^{0.5}}$ $\alpha_{i(bub)} = \frac{B_2 \dot{m}^{0.1} Q_i^{0.7}}{D_{hyd}^{0.5}} \quad \alpha_{i(Smooth tubes)} = \max(\alpha_{i(kon)}, \alpha_{i(bub)})$				

Fin coil heat exchanger	Symbol	Unit	Description	Page 24																																																			
Inside grooved tubes, heat transfer coefficient inside $d_f = d_m + \left(h \tan \left(\frac{\beta \pi}{360} \right) \right) \quad d_s = d_m - \left(h \tan \left(\frac{\beta \pi}{360} \right) \right)$ $d_{ir} = 0.75d_f + 0.25d_s \quad X = d_{ir} \sqrt{\frac{2 \alpha_i}{\lambda_w d_{ir}}} \quad Y = \frac{\tanhyp(X)}{X} \quad Z = 1 - \left(\frac{(1 - Y)A_{r1}}{A_{r2}} \right)$ $\alpha_{i(Grooved\ tubes)} = Z \alpha_{i(Smooth\ tubes)}$	$\Delta p_{r(K)}$ G_r	 K Pa/K	Pressure drop total inside Gradient																																																				
Pressure drop in tubes for agents with aggregate state change Pressure drop on condensation	Gradient, valid for condensation and evaporation																																																						
$f_v = \frac{\eta_{liq}}{\eta_{sol}} \quad f_d = \frac{\rho_{liq}}{\rho_{sol}} \quad f_{w1} = \frac{f_v \dot{m}^{0.5}}{62 \eta_{liq}^{1/6} g^{1/6} \rho_{liq}^{1/3} f_d^{0.1}}$ $f_{w2} = \sum_{n=1}^{10} 0.1 \left((0.1n - 0.05)^{\frac{14}{19}} + f_{w1} (0.1n - 0.05)^{\frac{14}{19}} (1.05 - 0.1n)^{0.5} \right)^{19/8}$ $Re_{rm} = \frac{\dot{m} D_{hyd}}{\eta_{liq}} \quad \xi_1 = \frac{64}{Re_{rm}} \quad \xi_2 = 0.3164 Re_{rm}^{-0.25}$ $\xi_3 = 0.0054 + 0.3964 (Re_{rm}^{-0.3}) \quad \xi_4 = \left(2 \log \left(Re_{rm} \sqrt{\xi_4} \right) \right)^{-2}$ $\xi_5 = \left(2 \log \left(\frac{D_{hyd}}{r_r} \right) + 1.14 \right)^{-2} \quad \xi_6 = \left(-2 \log \left(\frac{2.51}{Re_{rm} \sqrt{\xi_6}} + \frac{r_r}{3.71 D_{hyd}} \right) \right)^{-2}$ $\xi_7 = \max(\xi_1, \xi_2, \xi_3, \xi_4, \xi_5, \xi_6) \quad \xi_{rm} = PA \left(\frac{\xi_7 B}{D_{hyd}} + 1 \right) + 5$ $\Delta p_r = \left(\frac{\xi_{rm} f_{w2} \dot{m}^2}{2 \rho_{sol}} \right) + \left(\dot{m}^2 \left(\frac{1}{\rho_{liq}} - \frac{1}{\rho_{sol}} \right) \right) \quad \Delta p_{r(K)} = \frac{\Delta p_r}{G_r}$	<table> <tr> <th colspan="3">Refrigerant R410A (REFPROP)</th> <th>Polynomial approximation (Table Curve 2D)</th> </tr> <tr> <th>Temp. (t) °C</th> <th>Pressure (p) Pa</th> <th>Gradient (Gr) Pa/K</th> <th>$p = a + bt + ct^2 + dt^3$</th> </tr> <tr> <td>30.00</td> <td>1889338.04</td> <td></td> <td></td> </tr> <tr> <td>32.00</td> <td>1988619.80</td> <td>50611.13</td> <td>$a = 777887.5515$</td> </tr> <tr> <td>34.00</td> <td>2091782.57</td> <td>52578.39</td> <td>$b = 27693.40974$</td> </tr> <tr> <td>36.00</td> <td>2198933.37</td> <td>54600.09</td> <td>$c = 234.7441189$</td> </tr> <tr> <td>38.00</td> <td>2310182.92</td> <td>56678.16</td> <td>$d = 2.568982299$</td> </tr> <tr> <td>40.00</td> <td>2425646.03</td> <td>58814.84</td> <td></td> </tr> <tr> <td>42.00</td> <td>2545442.29</td> <td>61012.67</td> <td></td> </tr> <tr> <td>44.00</td> <td>2669696.70</td> <td>63274.56</td> <td></td> </tr> <tr> <td>46.00</td> <td>2798540.51</td> <td>65603.88</td> <td></td> </tr> <tr> <td>48.00</td> <td>2932112.21</td> <td>68004.53</td> <td></td> </tr> <tr> <td>50.00</td> <td>3070558.64</td> <td></td> <td></td> </tr> </table> Simplified calculation method see green cells above $G_{r(40)} = \frac{p_{42} - p_{38}}{t_{42} - t_{38}} = 58814.84$ The error is 0.0183 % against the exact method However, one go closer to the critical point, the error increases significantly. Gradient = Differential according to t $G_r = b + 2ct + 3dt^2$ $t = 40 \rightarrow G_r = 58804.05$ Gradient =Pa/K) 			Refrigerant R410A (REFPROP)			Polynomial approximation (Table Curve 2D)	Temp. (t) °C	Pressure (p) Pa	Gradient (Gr) Pa/K	$p = a + bt + ct^2 + dt^3$	30.00	1889338.04			32.00	1988619.80	50611.13	$a = 777887.5515$	34.00	2091782.57	52578.39	$b = 27693.40974$	36.00	2198933.37	54600.09	$c = 234.7441189$	38.00	2310182.92	56678.16	$d = 2.568982299$	40.00	2425646.03	58814.84		42.00	2545442.29	61012.67		44.00	2669696.70	63274.56		46.00	2798540.51	65603.88		48.00	2932112.21	68004.53		50.00	3070558.64		
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Capillary outside diameter

Capillary inside diameter

Pressure drop in the capillaries

Capillary thickness

Flashgas on capillary outlet

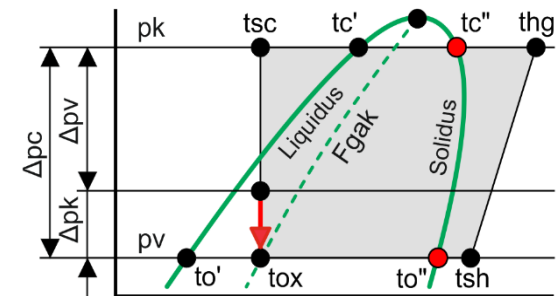
Capillary length

$$\max(\xi_1, \xi_2, \xi_3, \xi_4, \xi_5, \xi_6) \quad \xi_{rm} = PA \left(\frac{\xi_{7B}}{D_{hyd}} + 1 \right) + 5$$

pk tsc tc' tc'' thg

Δp

$$\xi_6 = \left(-2 \log \left(\frac{2.51}{Re_{km} \sqrt{\xi_6}} + \frac{r_r}{3.71 D_{ik}} \right) \right)^{-2}$$



Fin coil heat exchanger	Symbol	Unit	Description	Page 26
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Pressure drop on pump recirculation evaporation	F_{ga}	---	Flashgas part on the outlet point	
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$$f_v = \frac{\eta_{liq}}{\eta_{sol}} \quad f_d = \frac{\rho_{liq}}{\rho_{sol}} \quad f_{w1} = \frac{f_v \dot{m}^{0.5}}{62 \eta_{liq}^{1/6} g^{1/6} \rho_{liq}^{1/3} f_d^{0.1}} \quad F_{ga} = \frac{1}{P_u} \quad q = \frac{F_{ga}}{20}$$

$$f_{w2} = \sum_{n=1}^{10} 0.1 \left((q(2n-1))^{\frac{14}{19}} + f_{w1} (q(2n-1))^{\frac{14}{19}} (F_{ga} - q(2n-1))^{0.5} \right)^{19/8}$$

$$Re_{rm} = \frac{\dot{m} D_{hyd}}{\eta_{liq}} \quad \xi_1 = \frac{64}{Re_{rm}} \quad \xi_2 = 0.3164 Re_{rm}^{-0.25}$$

$$\xi_3 = 0.0054 + 0.3964 (Re_{rm}^{-0.3}) \quad \xi_4 = \left(2 \log (Re_{rm} \sqrt{\xi_4}) \right)^{-2} \quad \xi_5 = \left(2 \log \left(\frac{D_{hyd}}{r_r} \right) + 1.14 \right)^{-2} \quad \xi_5 = \left(2 \log \left(\frac{D_{hyd}}{r_r} \right) + 1.14 \right)^{-2}$$

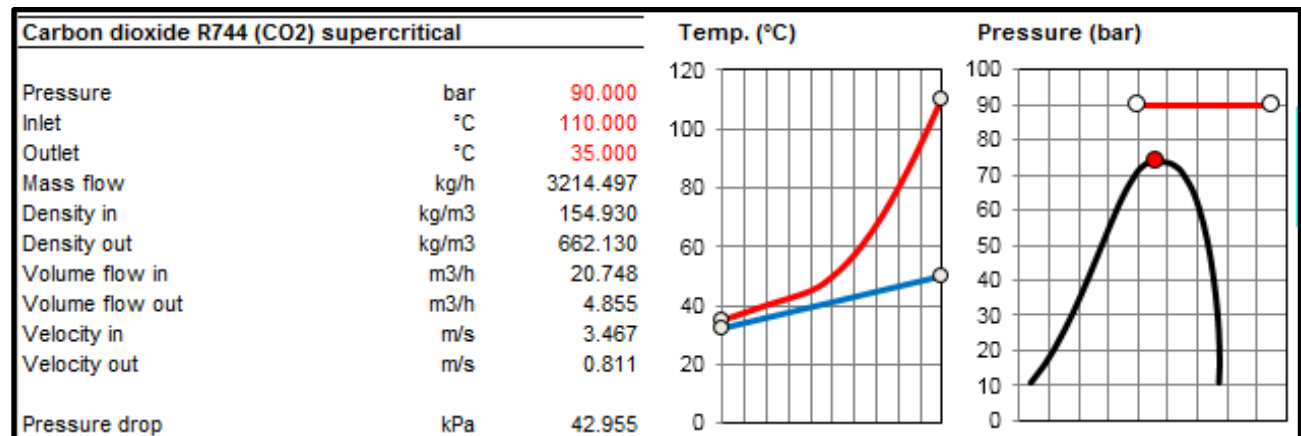
$$\xi_6 = \left(-2 \log \left(\frac{2.51}{Re_{rm} \sqrt{\xi_6}} + \frac{r_r}{3.71 D_{hyd}} \right) \right) \quad \xi_7 = \max(\xi_1, \xi_2, \xi_3, \xi_4, \xi_5, \xi_6) \quad \xi_{rm} = PA \left(\frac{\xi_7 B}{D_{hyd}} + 1 \right) + 5$$

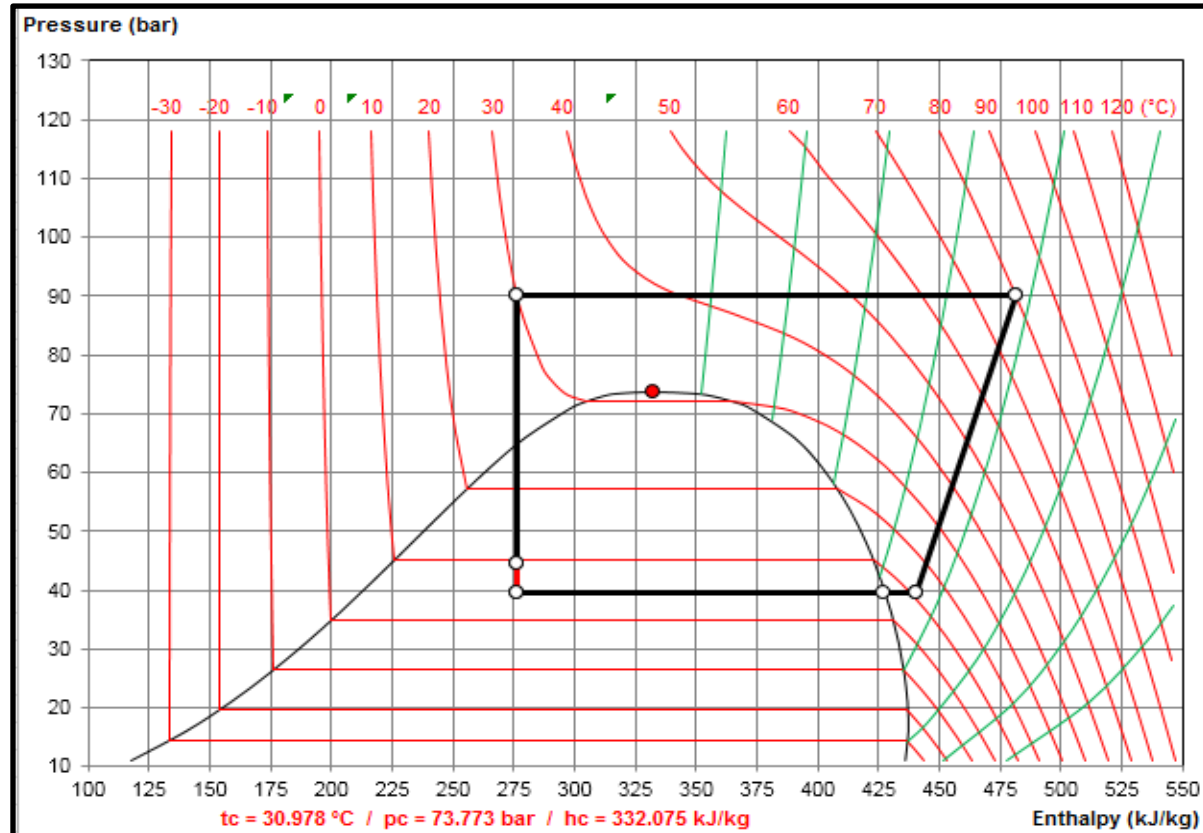
$$\Delta p_r = \left(\frac{\xi_{rm} f_{w2} \dot{m}^2}{2 \rho_{sol}} \right) + \left(\dot{m}^2 \left(\frac{1}{\rho_{sol}} - \frac{1}{\rho_{liq}} \right) \right) \quad \Delta p_{r(K)} = \frac{\Delta p_r}{G_r}$$

CO2-Cooler in the supercritical area

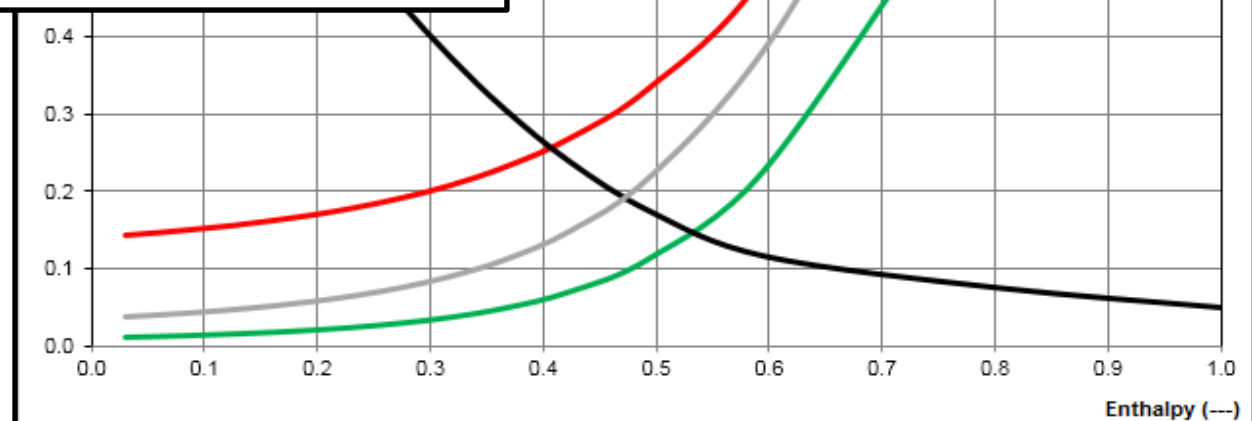
Because the thermodynamic values for cooling of CO2 in the supercritical range extremely change, the cooling process must be divided into 15 cells. The heat transfer coefficients, the mean logarithmic temperature difference and thus the need for a heat exchange surface change in the cells.

The calculation for the heat transfer coefficient and pressure drop occurs in the individual cells, as in the case of the media without a change in the state of the aggregate.





The precise determination of the injection point (flash gas) and the calculation of the capillaries are also a problem. These are carried out in the application HEH-DX-Evaporator.



There is very good software, such as REFPROP from NIST, which allows the calculation of the thermodynamic values of non-condensable mixed gases. A partial condensation of water is not taken into account. Software that would allow such calculations is not offered at a reasonable price. With the applications GHH and HEH-SR-G we offer the possibility to carry out such calculations at a reasonable price. By means of REFPROP from NIST, the mixed gas is first determined without condensable water vapor. Subsequently, the cooling process with condensable water vapor is calculated in the application GHH. GHH is therefore a Mollier diagram for non-condensable mixed gases with the possibility of calculating a partial condensation of water without determining the exact size of the heat exchanger. Alternatively, HEH-SR-G applications can be directly used when determining the size of the heat exchanger accurately.

Mixture Information



Mixture name: nitrogen/oxygen/carbon dioxide/carbon monoxide

Molar mass: 30.638 kg/kmol

Saturation fixed points

Critical Point

Cricondenther
(Max Temp.)

Temperature (°C)	Unknown	-67.404
Pressure (Pa)	Unknown	7091200.0
Density (kg/m³)	Unknown	160.34

Components and composition

Mass Fraction

nitrogen	0.55
oxygen	0.25
carbon dioxide	0.15
carbon monoxide	0.05

Example of a mixed gas of 4 non-condensable gases, thus without condensable water vapor, calculated with REFPROP from NIST. For our applications GHH and HEH-SR-G the values of -100 to 300°C are required in steps of 25 K even if, for example, only a cooling process from 150 to 30°C is to be calculated. Furthermore, the proportion of partially condensable water vapor has to be entered into these applications. In the following example, the cooling occurs with water from 10 to 40°C.

REFPROP - NIST Reference Fluid Properties (DLL version 9.1)

File Edit Options Substance Calculate Plot Window Help Cautions

1: nitrogen/oxygen/carbon dioxide/carbon monoxide: p = 100000.0 Pa (55/25/15/5)

	Temperature (°C)	Pressure (Pa)	Density (kg/m³)	Cp (J/kg-K)	Therm. Cond. (W/m-K)	Viscosity (Pa-s)	Prandtl
1	-100.00	100000.	2.1395	965.59	0.015077	0.000011493	0.73607
2	-75.000	100000.	1.8660	966.75	0.017183	0.000012951	0.72864
3	-50.000	100000.	1.6550	969.52	0.019194	0.000014351	0.72492
4	-25.000	100000.	1.4872	973.15	0.021158	0.000015697	0.72200
5	0.00000	100000.	1.3504	977.31	0.023085	0.000016994	0.71944
6	25.000	100000.	1.2367	981.86	0.024974	0.000018244	0.71727
7	50.000	100000.	1.1407	986.74	0.026826	0.000019453	0.71555
8	75.000	100000.	1.0586	991.93	0.028641	0.000020624	0.71430
9	100.00	100000.	0.98756	997.43	0.030419	0.000021760	0.71352
10	125.00	100000.	0.92545	1003.2	0.032162	0.000022864	0.71320
11	150.00	100000.	0.87071	1009.3	0.033873	0.000023939	0.71331
12	175.00	100000.	0.82209	1015.7	0.035553	0.000024986	0.71379
13	200.00	100000.	0.77861	1022.2	0.037204	0.000026008	0.71461
14	225.00	100000.	0.73951	1029.0	0.038828	0.000027006	0.71571
15	250.00	100000.	0.70416	1036.0	0.040427	0.000027981	0.71704
16	275.00	100000.	0.67203	1043.0	0.042002	0.000028936	0.71856
17	300.00	100000.	0.64271	1050.2	0.043555	0.000029871	0.72023

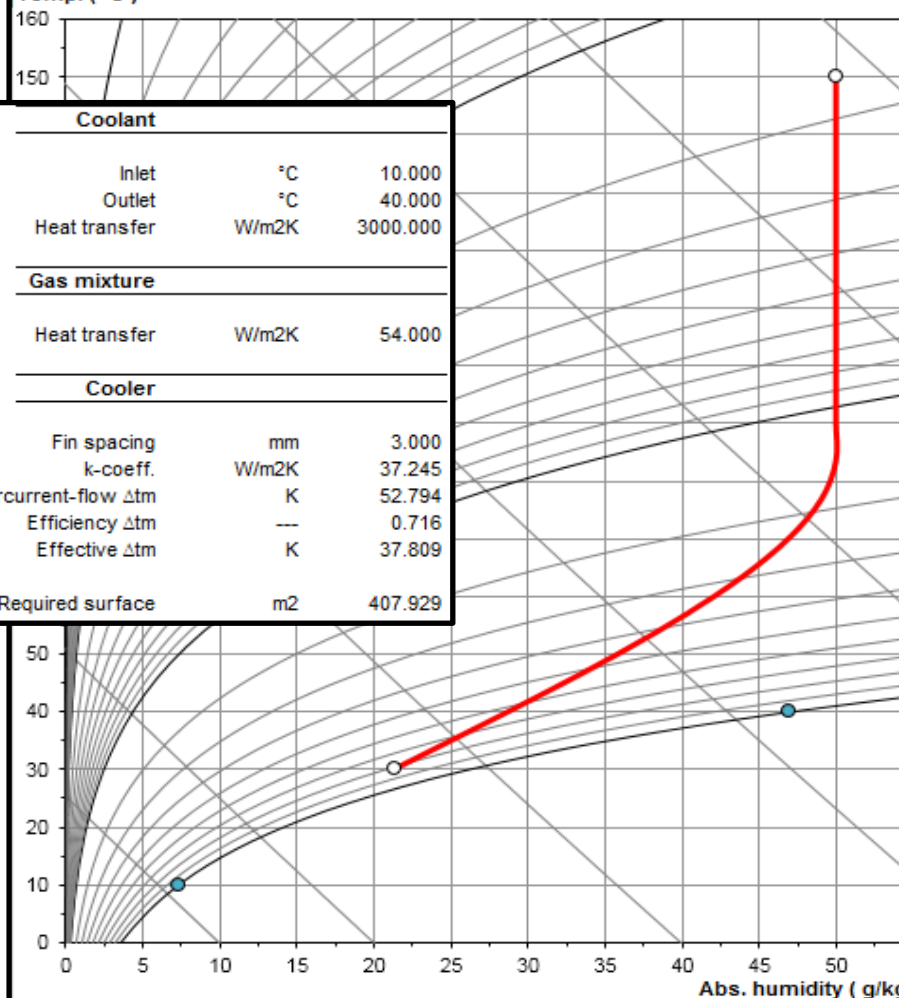
A water vapor quantity of 50 g/kg at the inlet was assumed. At the outlet it is still 20 g/kg. Accordingly, 30 g/kg water vapor condenses from. The size of the heat exchanger was determined only rudimentarily. A more precise calculation can be made with the 3 applications HEH-SR-G:

1. Cooling with liquid media
2. Cooling by injection evaporation
3. Cooling by pump-circulating evaporation

Gas mixture / Water		Inlet	Outlet
Pressure	barabs	1.000	
Temp.	°C	150.000	30.000
Rel. humidity	%	1.646	82.377
Abs. humidity	g/kg	50.000	21.317
Density humid	kg/m3	0.843	1.199
Enthalpy humid	kJ/kg	290.790	83.989
Wet bulb temperature	°C	51.887	27.439
Dew point temperature	°C	41.119	26.662
Volume flow humid	m3/h	12461.503	8520.704
Mass flow dry	kg/h	10000.000	10000.000
Condensate flow	kg/h		286.833
Capacity sensible	kW		352.322
Capacity latent	kW		222.127
Capacity frost	kW		0.000
Capacity total	kW		574.449

Mollier TX diagram for		Gas	Steam
Name		Gas mixture	Water
Formula		N2+O2+CO2+CO	H2O
CAS		---	7732-18-5
Molecular weight	kg/kMol	30.638	18.015
Triple point temperature	°C	0.000	0.010
Evaporation-Enthalpy (0.000 °C)	J/kg		2500900.000

Temp. (°C)



Coolant		
Inlet	°C	10.000
Outlet	°C	40.000
Heat transfer	W/m2K	3000.000
Gas mixture		
Heat transfer	W/m2K	54.000
Cooler		
Fin spacing	mm	3.000
k-coeff.	W/m2K	37.245
Countercurrent-flow Δtm	K	52.794
Efficiency Δtm	---	0.716
Effective Δtm	K	37.809
Required surface	m2	407.929



Company
Branch
Street
Country / ZIP / City

Phone: xxxxxxxxxxxx
Fax: xxxxxxxxxxxx
E-Mail
Homepage

City, 04.01.2017
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software by www.zcs.ch
created for the range
0.5 / 20 bar
-100 / 300 °C
0 / 1000 g/kg



Reduce the pollutants
by cooling,
condensation and
optimal separation

Cooler: 42/36/20-12R-30T-1800A-3.0PA-30C-AISI 316/AISI 316/AISI 316



Capacity	kW	570.980	----- sensible:	352.635
Surface reserve	%	7.321	latent:	218.345
Present surface	m ²	434.640	frost:	0.000
Required surface	m ²	404.989		
k-coeff.	W/m ² K	37.254	ffa:	5.000E-05
Mean temp. diff.	K	37.845		

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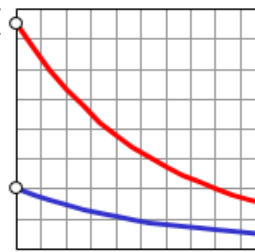
City, 04.01.2017
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Gas mixture / Water	Inlet	Outlet	Medium
Pressure	bar	1.000	
Temp.	°C	150.000	30.000
Rel. humidity	%	1.646	84.197
Abs. humidity	g/kg	50.000	21.805
Density humid	kg/m ³	0.843	1.198
Enthalpy humid	kJ/kg	290.790	85.237
Volume flow humid	m ³ /h	12461.503	8527.533
Mass flow dry	kg/h	10000.000	10000.000
Condensate flow	kg/h		281.950
Surface temperature	°C	77.812	16.875
Velocity	m/s	1.526	1.044
Pressure drop (dry 119 Pa)	Pa		139.544

Water	Inlet	Outlet	Medium
Temp.	°C	10.000	40.000
Density	kg/m ³		997.209
Spec. heat	kJ/kgK		4.180
Heat cond.	W/mK		0.611
Viscosity	Pas		8.901E-04
Volume flow	m ³ /h		16.438
Velocity	m/s		0.598
Pressure drop	kPa		15.066



Who prefers to let lay out the whole,
can contact us for what we need of
course all data. We will gladly provide
you with an offer on a fee base.

It should be noted, that we make such
calculations only against prepayment.

Cert.-Eng. Marin Zeller TU, VDI

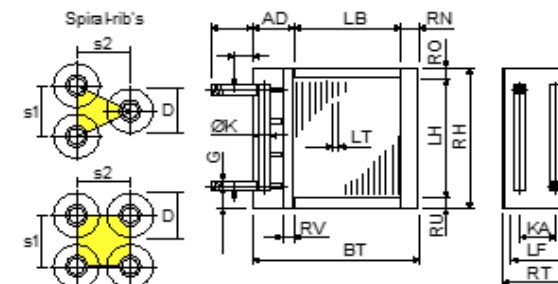


Finned width	LB	mm	1800
Finned depth	LF	mm	436
Frame on top	RO	mm	40
Frame on bottom	RU	mm	40
Frame in front	RV	mm	30
Frame on back (~84mm)	RN	mm	84
Collector-Diameter	K	mm	76
Collector covering	AD	mm	186
Collector distance	KA	mm	401
Fin spacing	LT	mm	3.000
Fin thickness	LD	mm	1.000
Tube diameter	d / D	mm	20.000 40.000
Tube thickness	S	mm	1.000
Tube interval on the height	S1	mm	42.000
Tube interval on the depth	S2	mm	36.373

Application	GHH	HEH-SR-G
Capacity sensible	352 kW	353 kW
Capacity latent	222 kW	218 kW
Capacity total	574 kW	571 kW
Required surface	407 m ²	405 m ²

Piece	360	Tubes:	flat	AISI 316
Piece	0		staggered	
Piece	0	Collectors:	1.23 m/s	AISI 316
Piece	0	Connections:	1.23 m/s	AISI 316
Piece	12	Fins:	ribbed	AISI 316
Piece	30	Frame:	2.00 mm	AISI 316
Piece	12	Circulations:	1	Default
Piece	30	Protection:		without

El. heat rods: ---
Air flow direction: horizontal
Special: Bottom plate perforated
for perfect condensate drain



Delivery:	5-6 weeks
Validity:	12 weeks
Condit.:	net, prepaid address
Payment:	30 days net
Non el. rods:	EUR 38802.00

