STUDIE DAY ON 16 NOVEMBER 2023

Simulation of an air cooler

Jean Lebrun

Atic

Heat and mass transfer

Dry regime:

Global heat transfer coefficient:

$$AU_{HEjk,dry} = \frac{A_{HEjk}}{\dfrac{1}{hc_{HEjk,prim}} + \dfrac{1}{hc_{HEjk,second}}}$$

Wet regime:

$$A_{HEjk,wet} = \varepsilon_{wetting,HEjk} \cdot A_{HEjk}$$

Fictitious specific heat:

$$c_{p,f,HEjk,second,wet} = \frac{h_{HEjk,second,su} - h_{HEjk,second,ex,wet}}{T_{wb,HEjk,second,su} - T_{wb,HEjk,second,ex,wet}}$$

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Fictitious heat transfer coefficient:

$$hcf_{HEjk,second,wet} = hc_{HEjk,second} \cdot \frac{c_{p,f,HEjk,second,wet}}{c_{p,HEjk,second,su}}$$

Global heat transfer coefficient:

$$AU_{HEjk,wet} = \frac{A_{HEjk,wet}}{\frac{1}{hc_{HEjk,prim}} + \frac{1}{hcf_{HEjk,second,wet}}}$$

Fictitious capacity flow rate:

$$\dot{C}_{f,HEjk,second,wet} = \dot{M}_{HEjk,second} \cdot c_{p,f,HEjk,second,wet}$$

Other equations are the same as in dry regime...

Convective heat transfer coefficients:

$$hc_{HEjk,prim} = Nusselt_{HEjk,prim} \cdot \frac{k_a}{D_{HEjk,prim}}$$

$$Nusselt_{HEjk,prim} = Stanton_{HEjk,prim} \cdot Re_{HEjk,prim} \cdot Prandtl$$

$$Stanton_{HEjk,prim} = \frac{j_{HEjk,prim}}{Prandtl^{(2/3)}}$$

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Local, continuous and global pressure drops and corresponding fans consumptions:

$$\Delta P_{local,HEjk,prim} = Kc_{HEjk,prim} \cdot \rho_{HEjk,prim,su} \cdot \frac{vel^2_{HEjk,channel,prim,su}}{2}$$

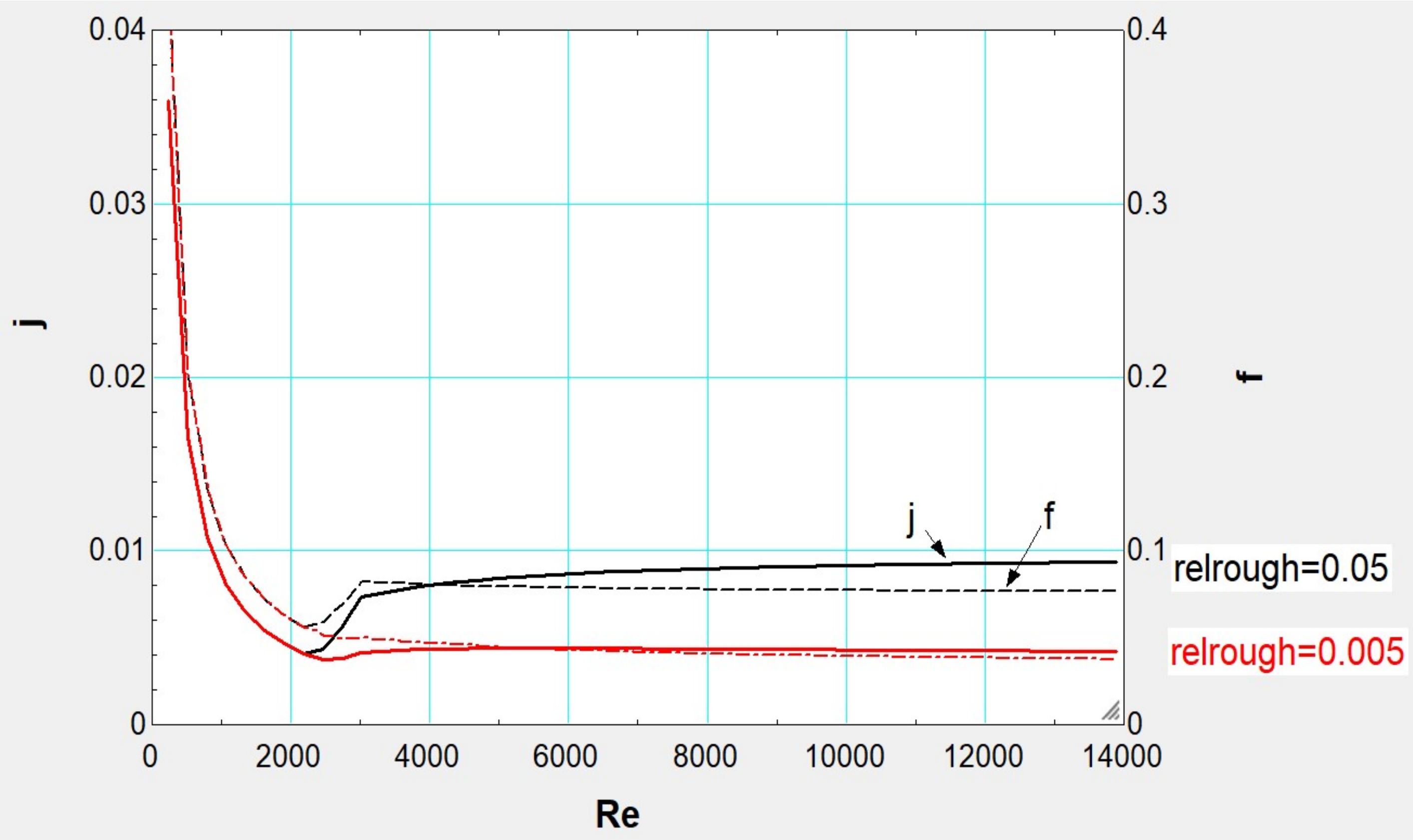
$$\Delta P_{cont,HEjk,prim} = f_{HEjk,prim} \cdot \frac{L_{HEjk,channel,prim}}{D_{HEjk,prim}} \rho_{HEjk,prim,su} \cdot \frac{vel^2_{HEjk,channel,prim,su}}{2}$$

$$\Delta P_{HEjk,prim} = \Delta P_{cont,HEjk,prim} + \Delta P_{local,HEjk,prim}$$

$$\dot{W}_{HEjk,fan,prim} = \dot{V}_{HEjk,prim,su} \cdot \frac{\Delta P_{HEjk,prim}}{\eta_{HEjk,fan,prim}}$$

Same equations on secondary side…

Typical evolutions of Colburn ("j") and friction ("f") factors as functions of the Reynolds inside a rectangular channel



In turbulent regime, both factors are almost only depending from the roughness and the following laws can be used in fair approximation:

simplified regressions:

$$f_{approx} = 0.044 + 0.77 \cdot RelRough$$

$$j_{approx} = 0.004 + 0.08 \cdot RelRough$$

The (real or fictitious) roughness factor appears as a convenient parameter to be tuned in order to fit with experimental results available.

Key issue: proportionality between “f” and “j”…

Simplified definition of the wet regime

The air is not supposed to be *pre-humidified* upstream to the secondary side of the heat exchanger.

The "*wetting effectiveness"* is defined as a ratio between the *wet* transfer area (actually or fictitiously) available and the *geometrical* area.

Parameters:

Heat exchanger geometry:

Plate sizes:

$L_{HEjk,1}$ = 0.47 [m]

$L_{HEjk,2}$ = 0.47 [m]

Plate thickness:

$Plthick_{HEjk}$ = 0.00014 [m]

Surface relative roughness:

$Relrough_{HEjk,prim}$ = 0.04 [-]

$Relrough_{HEjk,second}$ = 0.04 [-]

Number of plates:

Npl_{HEjk} = 119 [-]

Pitch:

pt_{HEjk} = 0.00335 [m]

Flows arrangement:

counter or cross flow

Flows directions:

Primary flow along L1

Secondary flow:

along $L_{HEjk,1}$ in counter flow;

along $L_{HEjk,2}$ in cross flow.

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Local pressure drop factors:

$Kc_{HEjk,prim} = 2$ [-]

$Kc_{HEjk,second} = 2$ [-]

(hypothetical)

Wetting effectiveness:

$\varepsilon_{wetting,HEjk} = 0.8$ [-]

(hypothetical)

Fans efficiencies:

$\eta_{HEjk,fan,prim} = 0.6$ [-]

$\eta_{HEjk,fan,second} = 0.6$ [-]

(hypothetical)

Both plates sizes “LHEjk1” and “LHEjk2” can be tuned, in order to fit with the actual performances.

Also tune the roughness.

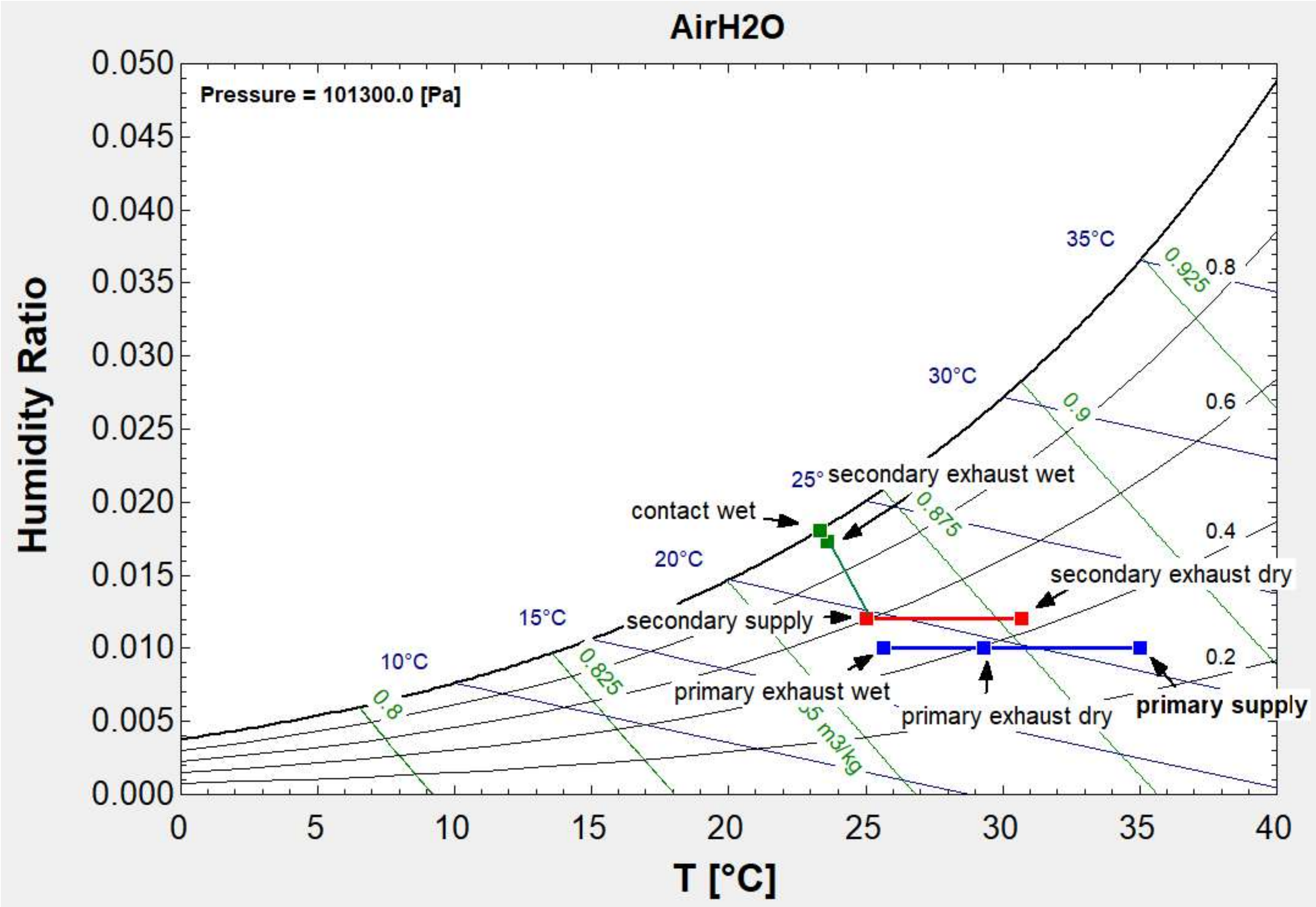
These tunings have direct impacts on *both heat and mass transfers and pressure drops* .

In turbulent regime, the number of plates “Npl”, can be tuned to adjust the continuous pressure drops *without any effect on global heat transfer coefficient*.

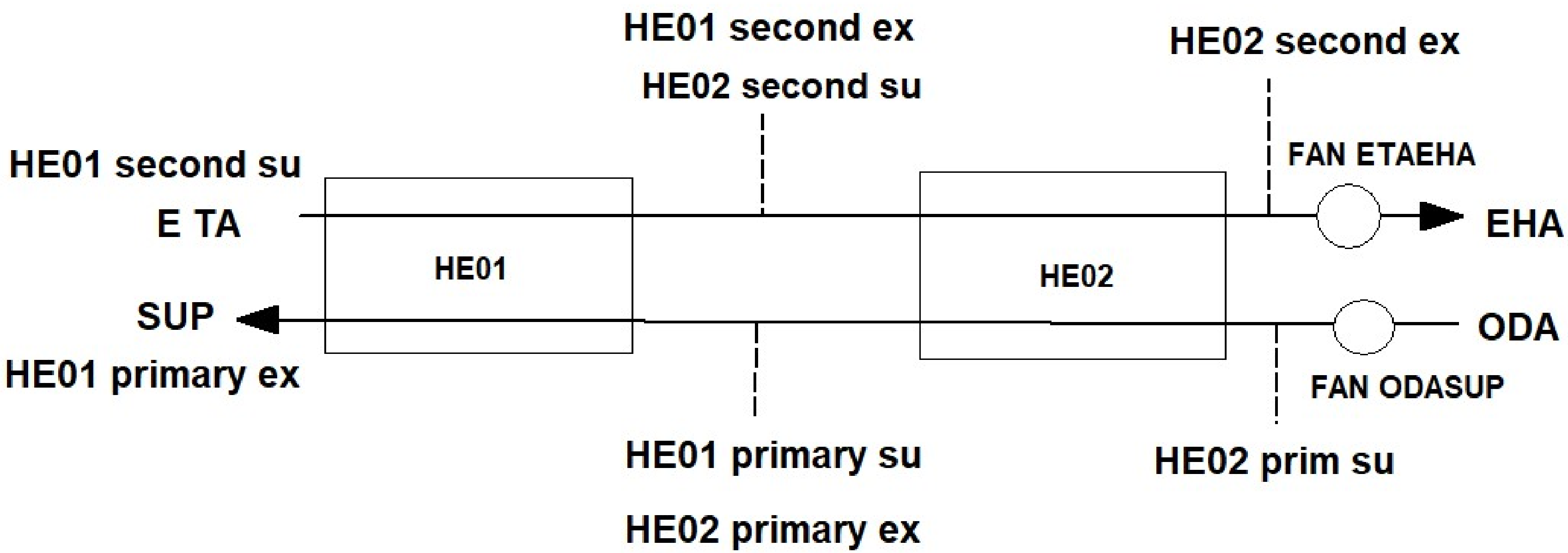
Same idea with the local pressure drop factors “Kc”.

“Kc” may include the both continuous and local pressure drops occurring upstream and downstream of the heat exchanger.

Main air states on both sides of the heat exchanger



Example of real system



Analysis of manufacturer data in "mode 1"

	Mode n°1 (Winter)
Temperature ODA [°C]	-9.6
Phi ODA [%]	87
Temperature SUP [°C]	18.8
Phi SUP [%]	11
Temperature ETA [°C]	22
Phi ETA [%]	40
Temperature EHA [°C]	0.5
Phi EHA [%]	100

	Mode n°1 (Winter)
Friction losses (heat exchanger) ODA - SUP [Pa]	90
Friction losses (heat exchanger) ETA-EHA [Pa]	112
Fan pressure ODA - SUP [Pa]	768
Fan pressure ETA - EHA [Pa]	652
Standard airflow ODA - SUP [m³/h]	4000
Standard airflow ETA - EHA [m³/h]	4000
Power consumption of the fan ODA - SUP [kW]	1.3
Power consumption of the fan ETA - EHA [kW]	1.12
Condensate water (ETA-EHA) [kg/h]	12.6

Tuning of the parameters to fit with manufacturer data

In this "*mode 1*", it appears that the most realistic hypotheses consist in assuming a *fully dry regime in the first heat exchanger* and a *fully wet regime in the second one*. *This wet regime is due to "natural" condensation of the water vapor(as inside a cooling coil), without any injection of liquid water.*

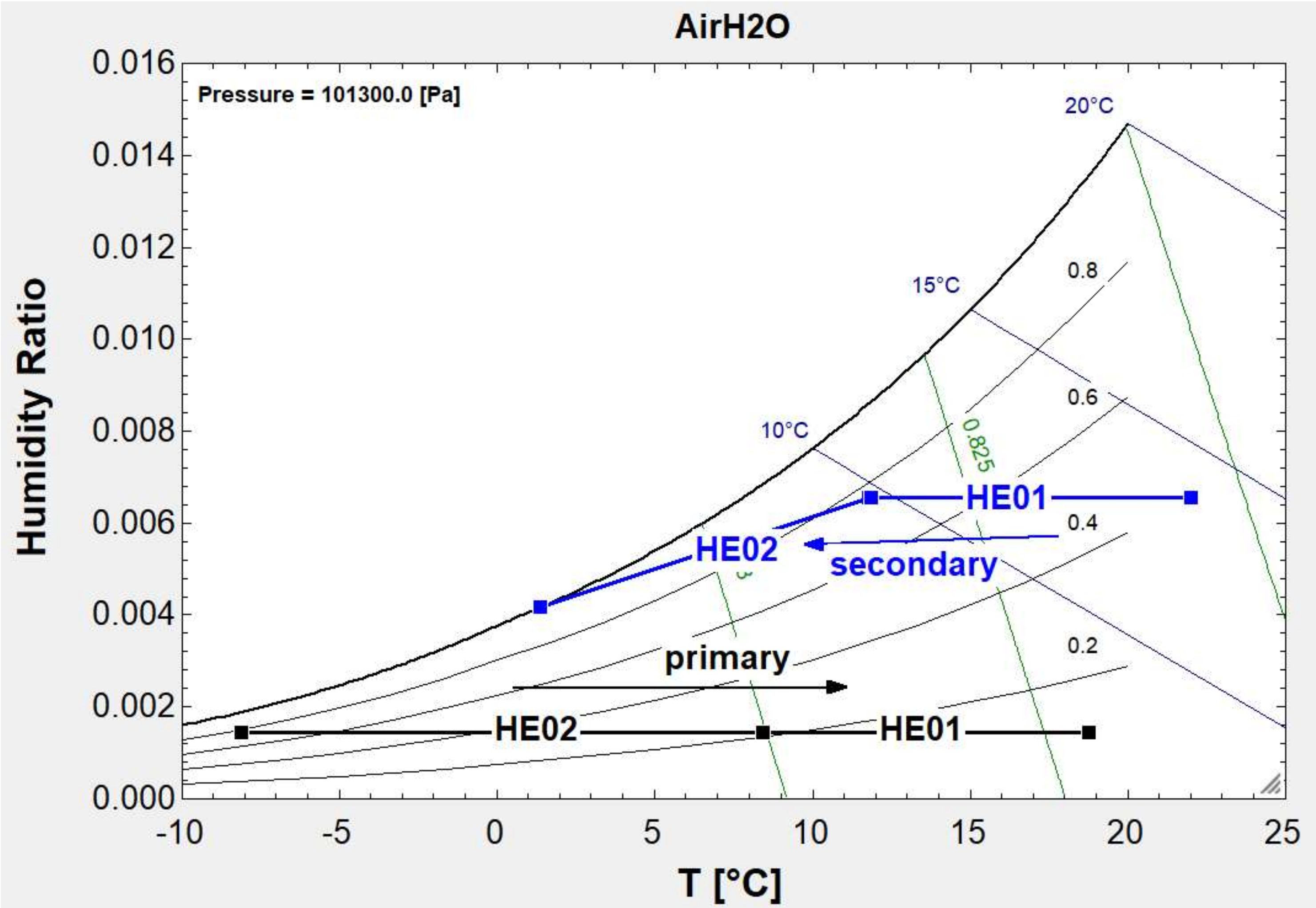
A first tuning is performed with the help of a *sizing factor:*

$$L_{HE01,1} = F_{sizing} \cdot 0.47 \ [m] \qquad L_{HE02,1} = F_{sizing} \cdot 0.47 \ [m]$$

$$L_{HE01,2} = F_{sizing} \cdot 0.47 \ [m] \qquad L_{HE02,2} = F_{sizing} \cdot 0.47 \ [m]$$

A satisfactory result **TSUP** is obtained with **Fsizing=2.5**

Psychrometric plotting of “mode 1” (with **Fsizing=2.5)**



Analysis of manufacturer data in mode 2

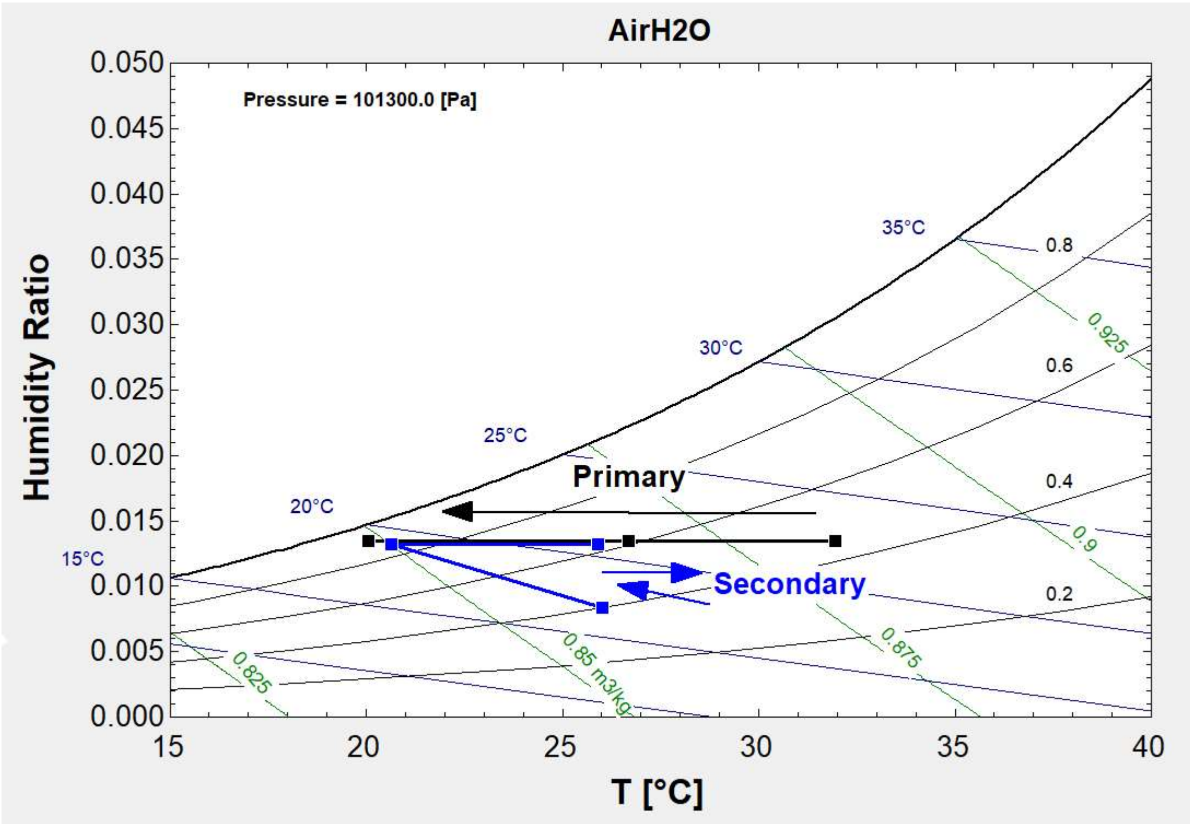
This mode is very different from the previous one: the *evaporation* (and no more *condensation*) “wet” regime, occurring on the secondary side of the *first* (and no more the *second*) heat exchanger, is obtained by artificial wetting of this surface.

	Mode n°2 (IEC 1)
Fan pressure ODA - SUP [Pa]	776
Fan pressure ETA - EHA [Pa]	641
Standard airflow ODA - SUP [m³/h]	4086
Standard airflow ETA - EHA [m³/h]	4086
Power consumption of the fan ODA - SUP [kW]	2.02
Power consumption of the fan ETA - EHA [kW]	1.69

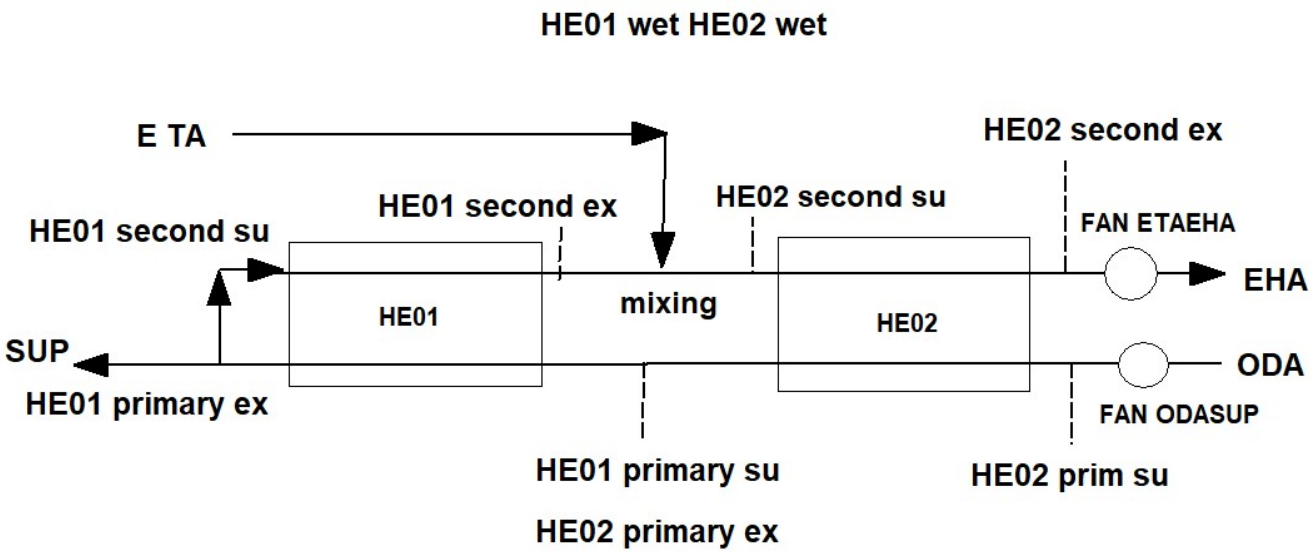
	Mode n°2 (IEC 1)
Temperature ODA [°C]	30,5
Phi ODA [%]	49
Temperature SUP [°C]	20
Phi SUP [%]	91
Temperature ETA [°C]	26
Phi ETA [%]	55

A satisfactory result **TSUP** is obtained with **Fsizing=0.67,** i.e. *much smaller than in mode 1*. *This strong difference has still to be explained*

Psychrometric plotting of “mode 2”



Association of two heat exchangers in series with partial recirculation



Analysis of manufacturer data in "mode 3"

	Mode n°3 (IEC 2)
Fan pressure ODA - SUP [Pa]	881
Fan pressure ETA - EHA [Pa]	646
Standard airflow ODA - SUP [m³/h]	6236
Standard airflow ETA - EHA [m³/h]	6236
Power consumption of the fan ODA - SUP [kW]	3.32
Power consumption of the fan ETA - EHA [kW]	2.47

Return air flow rate: 2150 Nm3/h

	Mode n°3 (IEC 2)
Temperature ODA [°C]	30,4
Phi ODA [%]	49
Temperature SUP [°C]	19,2
Phi SUP [%]	95
Temperature ETA [°C]	26
Phi ETA [%]	55

A satisfactory result **TSUP** is obtained with **Fsizing=1.5 (**i.e. *somewhere between modes 1 and 2).*

A tuning of the number of plates can be performed in such a way to reconciliate measured and simulated powers of the fan ODASUP.

The best value is **FNpl=1.391**

A reconciliation between measured and simulated powers of the fan ETAEHA Tuning of KC on secondary side

Best value: **Kc=6.73**

Analysis of monitoring results
- Period selected: ***12 to 15h***
- Mode 3

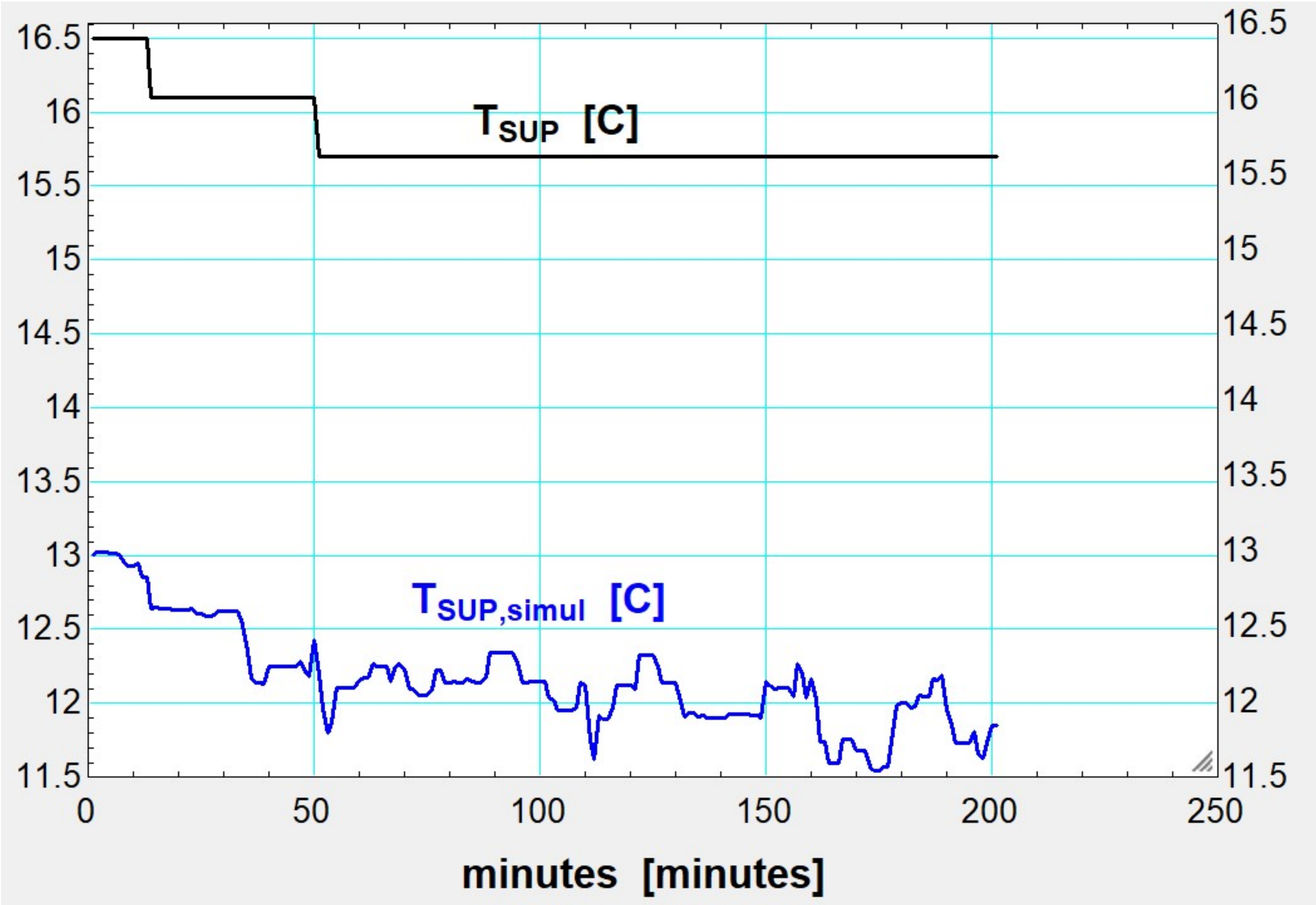
Other periods and other modes are less reliable because of:

1) strong disagreement among climate measured on site considered and on meteorological stations of the surroundings;
2) too short and unstable conditions in mode 2 (begin of the day);
3) unexplained control action around 16h.

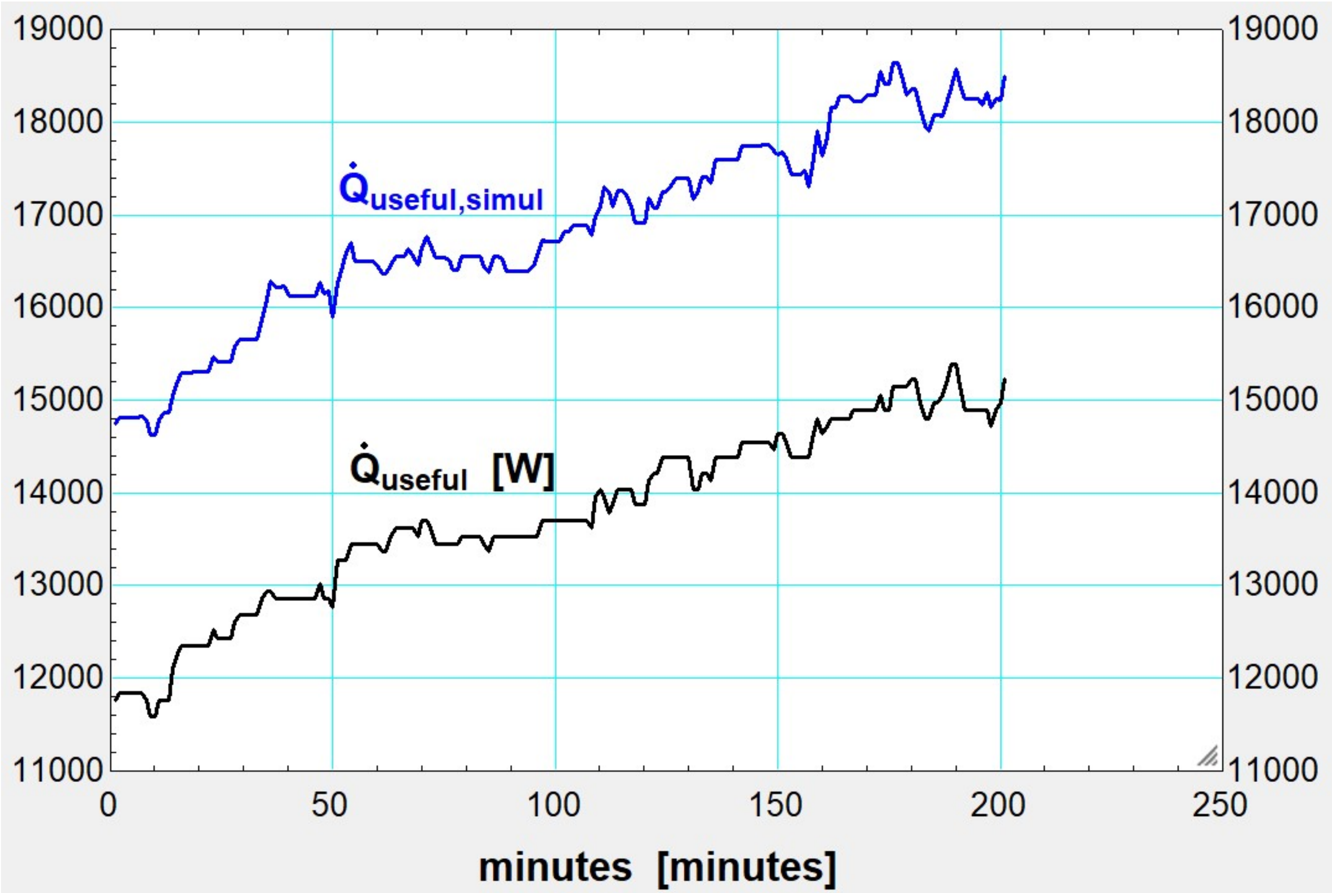
Simulation:

The parameters are kept the same as in “mode 3” already analysed.

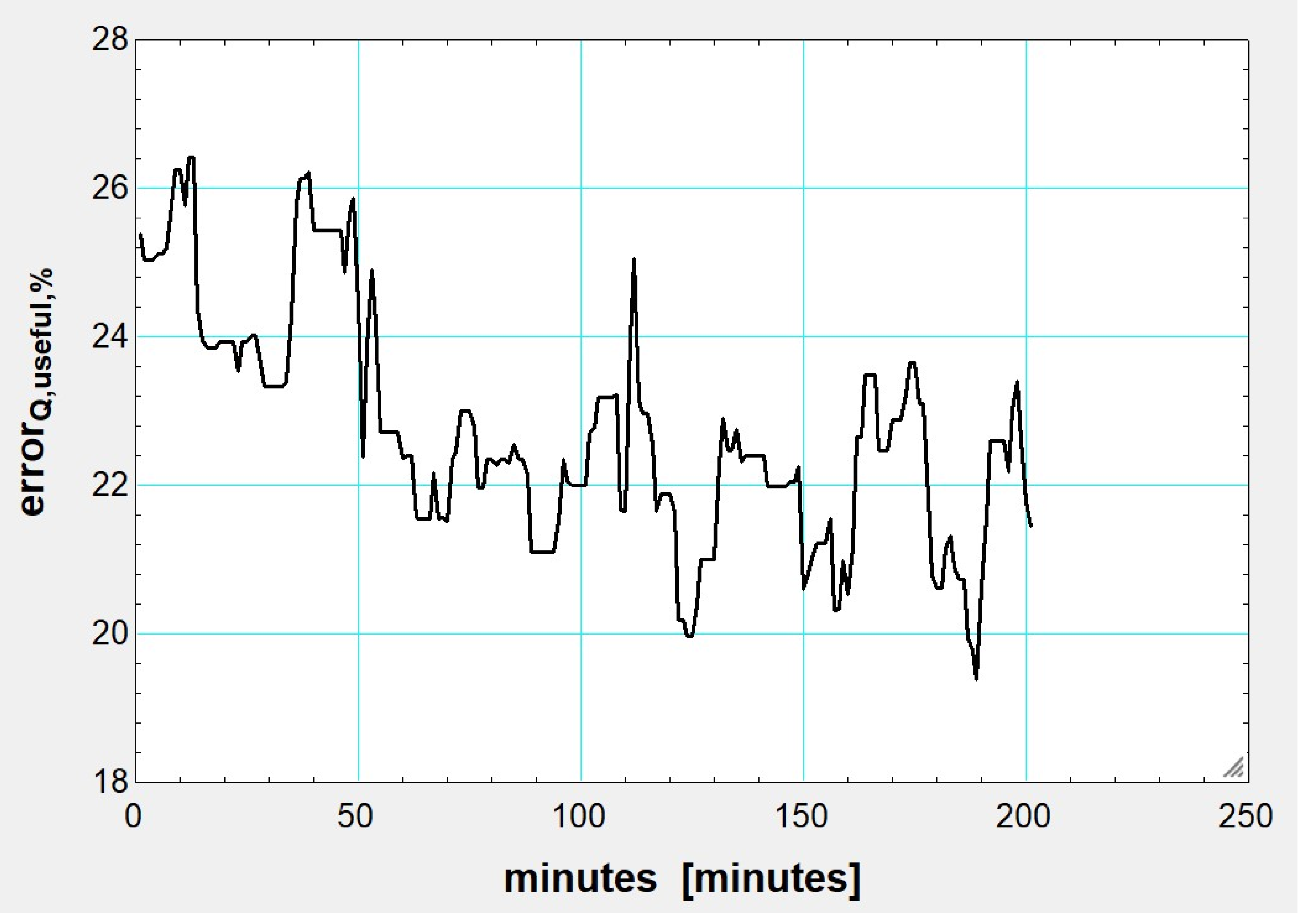
Measured and simulated building supply temperature (“TSUP”)



Measured and simulated useful thermal power



Relative error committed on the useful thermal power



It appears that the relative error committed on the useful thermal power is of the order of 20 %.
From other part, the actual fans consumption is fairly well simulated:

$$\dot{W}_{fans} = 1712 \ [W] \quad \dot{W}_{fans,simul} = 1943 \ [W]$$

Provisory conclusions

This is no more than a very first step.

Much more experimental results will be welcome in order to validate and tune such simulation model…