IEA-ECB Annex 85

Modelling of an indirect evaporative air cooling system

(In relation with subtask D)

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1. Introduction

This is the continuation of the work already presented [1].

The system considered includes two heat exchangers and two adiabatic humidifiers. Different "modes" of use will be hereafter considered.

New approximations are introduced into each heat exchanger simulation model, in order to make it more robust and to guaranty a shorter calculation time, mainly when the IEC model has to be connected to a large building and HVAC system.

2. Simplifying the simulation model (continued)

Previous simplifications consisted in keeping common reference values for all air properties, which are not significantly varying inside the system considered. Other variables were already calculated through simple polynomial expressions. Such approach is extended here, but most options are kept in reserve, inside the set of equations, in such a way to make easier any verification of actual accuracy.

Main equations and (arbitrary) reference values are listed hereafter.

Most of polynomial regressions are related to the *atmospheric pressure* taken as reference.

This means that this atmospheric pressure is considered as a "*parameter*" (i.e. a particular input, which is not significantly varying during the simulation).

Reference conditions:

Static pressure of reference:

p_{ref} = 101325 [Pa]

Temperature of reference:

t_{ref} = 22 [C]

Air humidity ratio:

ω_{ref} = 0.01 [-]

$p_{w,sat}$/Pascal=exp((A*(t/Celsius)/(B+(t/Celsius)))+C)

A = 17.438 [-]

B = 239.78 [-]

C = 6.4147 [-]

$p_{w,sat,ref}$ = 2644 [Pa]

(This polynomial expression is *not* depending on the actual atmospheric pressure).

saturation Humidity ratio:

$omega_{sat}$=0.622*$p_{w,sat}$/(p-$p_{w,sat}$)

or, at reference pressure,

$omega_{sat}$=D0+D1*T/Celsius + D2*(T/Celsius)^2

D0 = 0.005382

D1 = −0.000005526

D2 = 0.000023594148

$\omega_{sat,ref}$ = 0.01667 [-]

(The factors D0, D1 and D2 *are related* to the actual atmospheric pressure).

saturation enthalpy:

$h_{sat,ref}$=$c_{p,a}$*t_{ref}+$omega_{sat,ref}$*($c_{p,g*t,ref+h,fg,0}$)

or, at reference pressure,

h_{sat}/J\kg=F0+F1*t/Celsius+F2*(t/Celsius)^2+F3*(t/Celsius)^3

F0 = 9151.158

F1 = 1823.94

F2 = 8.90684

F3 = 1.02708

$h_{sat,ref}$ = 64469 [J/kg]

(The factors F0, F1, F2 and F3 *are related* to the actual atmospheric pressure).

wet bulb temperature:

$p_{w,sat,twb}$/Pascal=exp((A*($t_{wb/Celsius)/(B+(t,wb/Celsius)))+C}$)
$omega_{sat,twb}$=0.622*$p_{w,sat,twb}$/($p_{ref-p,w,sat,twb}$)
h_{twb}=$c_{p,a}$*t_{wb}+$omega_{sat,twb}$*($c_{p,g*t,wb+h,fg,0}$)
h_{twb}=h+($omega_{sat,twb-omega}$)*c_f*t_{wb}

Or

T_{wb}/Celsius=C0 + C1*h/J\kg +C2*(h/J\kg)^2 + C3*(h/J\kg)^3

$C0 = -4.994$

$C1 = 0.0006111$

$C2 = -3.676 \times 10^{-9}$

$C3 = 1.072 \times 10^{-14}$

$T_{wb,ref}$= 16.9 [C]

(The factors C0, C1, C2 and C3 *are related* to the actual atmospheric pressure).

fictitious specific heat:

dTwb\Celsius\dh\J\kg=C1 +2*C2*(h/J\kg) + 3*C3*(h/J\kg)^2
$c_{p,f}$=(1/dTwb\Celsius\dh\J\kg)*J\kg/Celsius

$c_{p,f,ref}$= 2991 [J/kg-C]

(*This is the most expedient and robust simplification!*)

partial pressure:

p_w=p*omega/(0.622+omega)

$p_{w,ref}$= 1603 [Pa]

relative humidity:

RH=p_w/$p_{w,sat}$

RH_{ref} = 0.6063 [-]

dew point:

t_{dp}/Celsius=E1 + E2*omega +E3*omega^2

$E1 = -5.103$

$E2 = 2322$

$E3 = -41384$

$t_{dp,ref} = 13.98$ [C]

(The factors E1, E2 and E3 *are related* to the actual atmospheric pressure).

3. The IEC system

The IEC air system considered here is very similar to a real system tested in Belgium and which can be used in three different modes:

- In mode "1", the system is reduced to classical heat exchanger which is useful almost only in *heating* season [2].
- The two *cooling* modes "2" and "3", hereafter roughly simulated, are represented in **Figures 1** and **2**.

The real system is supposed to be sized for a useful air flow rate of 4086 m3/h.

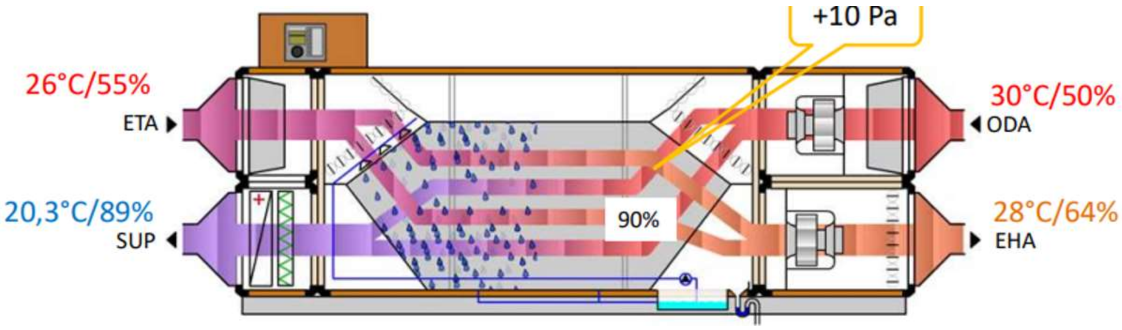


Figure 1: Use of the real ICE in "mode 2"

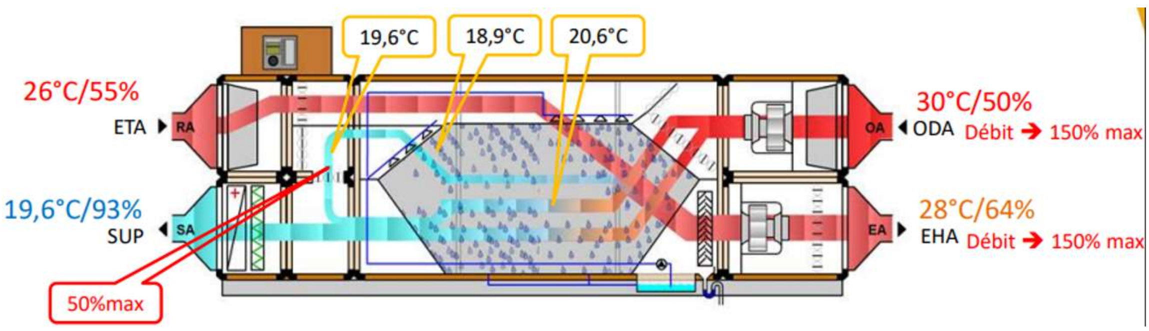


Figure 2: Use of the real CE in "mode 3"

In the waiting of more measuring results, the rough model of this system is based on the association in series of two counter flow heat exchangers.

The characteristics of these two heat exchangers are the same as previously considered [1], except for doubling two main sizes:

$$L_{HE01,1} = 2 \cdot 0.47 \quad [m]$$

$$L_{HE01,2} = 2 \cdot 0.47 \quad [m]$$

The air heating-up (less than 1 K) through each fan is not (yet) included in this modelling.

Typical results, obtained in modes "1", "2a", "2b", "2c" and "3" in same reference conditions are presented in **Figures 3** to **7**.

Mode 1: Dry heat exchanger

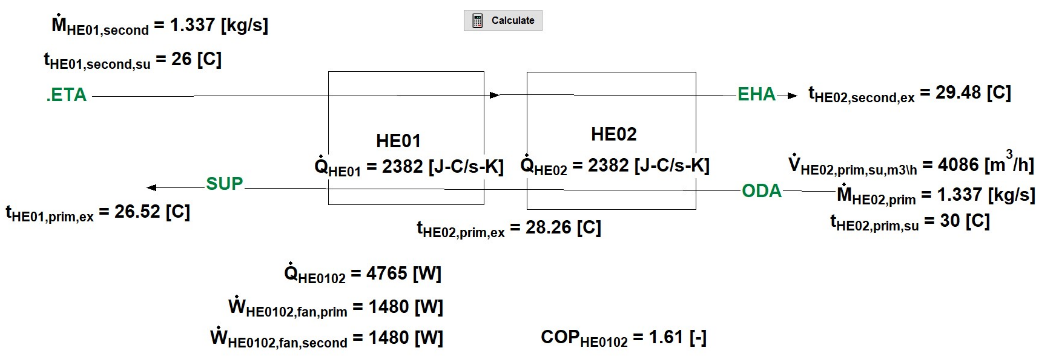


Figure 3: Mode 1: *dry heat exchange* only (EES file: JL220326-03 JL220322-03 HE01 and HE02 mode 1)

As to be expected, this cooling mode is of no interest in the reference conditions considered: because of too small difference between inside and outside temperature and because of significant fans consumptions, the global COP (1.6) doesn't appear as satisfactory…

Mode 2a: Dry heat exchanger, but with (perfect) adiabatic humidification of the air at secondary supply.

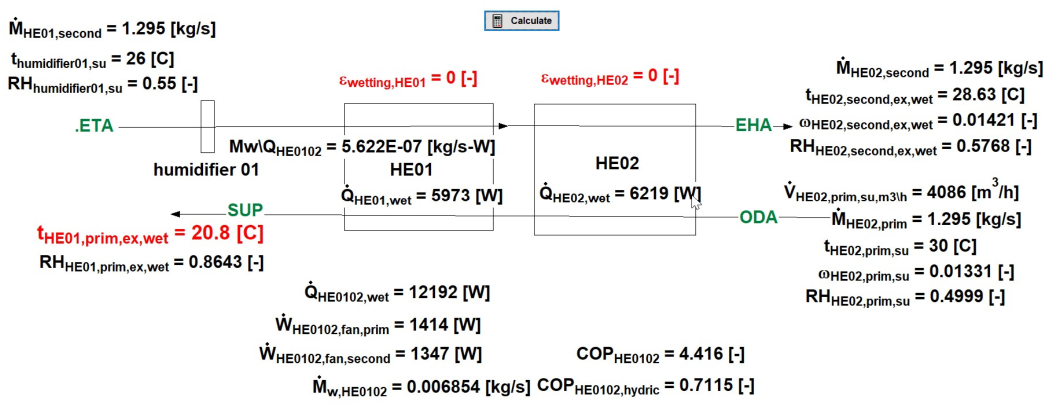


Figure 4: Mode2a: *with humidifier*, *but no surface wetting* (EES file: JL220327-01 JL220326-02 HE01 and HE02 mode 2a)

This mode appears as much more interesting than the previous one, but such attractive results are obtained with, at least, three *questionable hypotheses*:

- *Perfect adiabatic humidification* of the air (i.e. until saturation and at constant wet bulb temperature);
- *Negligible effect of this humidification on pressure drop* in secondary circuit;
- *Negligible consumption of the humidifier pump…*

Mode 2b: same as mode 2a, but with wetting of the secondary surface of heat exchanger 01.

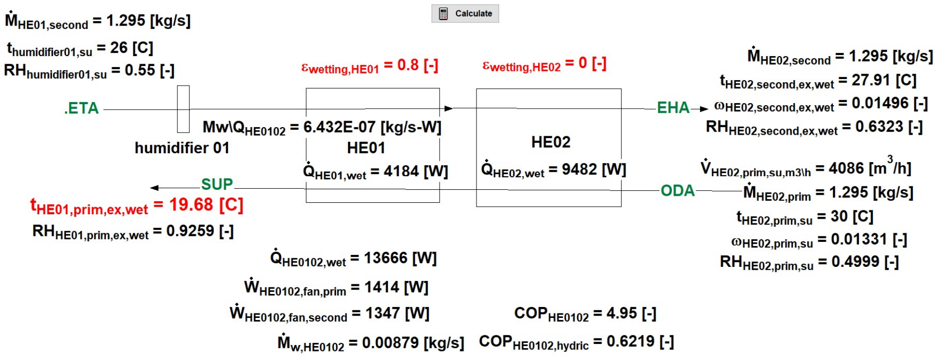


Figure 5: Mode 2b: *with wetting of secondary side of heat exchanger 01*

(EES file: JL220326-02 JL220322-03 HE01 and HE02 mode 2b)

The wetting of the secondary surface of heat exchanger 01 seems producing a small increase of cooling power and of corresponding coefficient of performance. But this apparent progress is related to *one more questionable hypothesis*: the surface wetting is supposed to be obtained *without significant increase* of air pressure drop.

Unfortunately, the surface wetting also generates a significant increase of the (minimal) water consumption and, therefore, a significant decrease of the hydric COP…

Mode 2c: same as mode 2b, but with, also, wetting of the secondary surface of heat exchanger 02

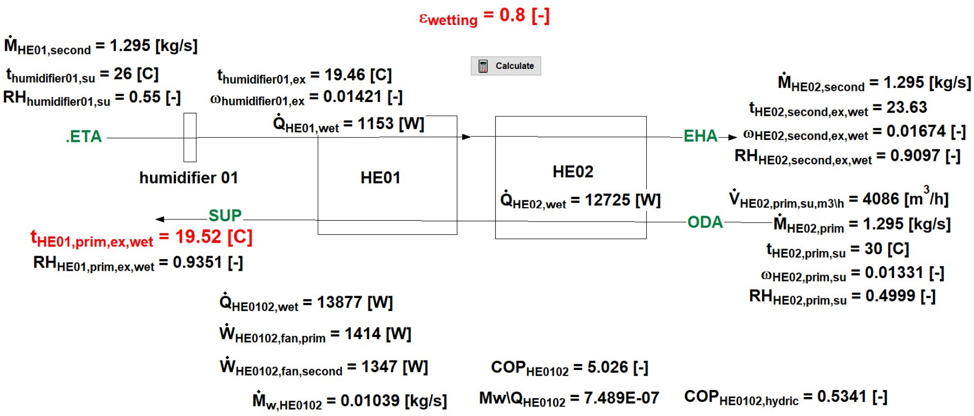


Figure 6: Mode 2c: *with wetting of secondary side of both heat exchangers*

(EES file: JL220322-03 HE01 and HE02 mode 2c)

It appears that the wetting of the secondary surface of heat exchanger 02 (which might need a *second humidifier*) doesn't generate a significant increase of global performances: the cooling power and the corresponding thermal COP appear as only very slightly increased, at the cost of a significant increase of water consumption. *Even less attractive results* would probably be obtained, if pressure drops and pumps consumptions were correctly identified…

Mode 3: with partial recirculation of primary air on the secondary side of heat exchanger 01

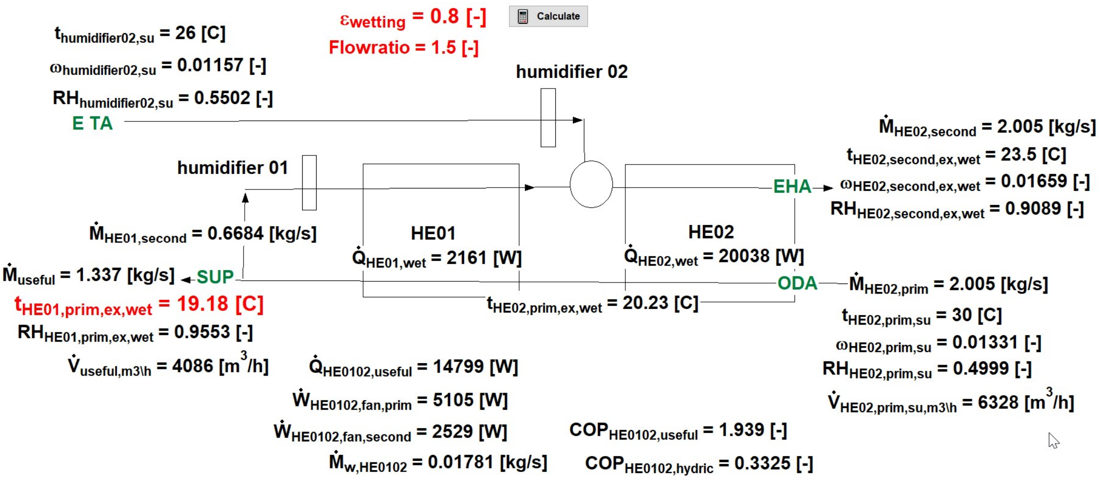


Figure 7: Mode 3 (EES file: JL220320-01 JL220218-02 IEC corrected).

In *present sizing*, *present reference conditions* and *present hypotheses*, this mode appears as producing disappointing results: too low thermal and hydric COP's.

Both COP's are strongly *decreasing* functions of the "air flow ratio" (ratio between the flow rate on primary side of both heat exchanger and the "useful" air flow rate) as shown in **Table 1**.

Flowratio [-]	$t_{HE01,prim,ex,wet}$ [C]	$Q_{HE0102,useful}$ [W]	$W_{HE0102,fans}$ [W]	$COP_{HE0102,useful}$ [-]	$M_{w,HE0102}$ [kg/s]	$COP_{HE0102,hydric}$ [-]
1.01	20.11	13528	2361	5.731	0.006896	0.7847
1.064	19.97	13712	2754	4.979	0.008121	0.6754
1.119	19.84	13896	3190	4.357	0.009346	0.5948
1.173	19.71	14073	3670	3.835	0.01057	0.5326
1.228	19.59	14237	4197	3.392	0.01179	0.4831
1.282	19.48	14384	4774	3.013	0.013	0.4425
1.337	19.39	14512	5404	2.686	0.01421	0.4085
1.391	19.31	14623	6088	2.402	0.01541	0.3795
1.446	19.24	14718	6831	2.155	0.01661	0.3544
1.5	19.18	14799	7634	1.939	0.01781	0.3325

Table1: Effect of flow ratio (EES file: JL220326-01 JL220320-01 JL220218-02 IEC corrected new parametric)

As to be expected, the thermal power and the fans consumption are both *increasing* functions of the air flow ratio, but, also as to be expected , the increase of fans consumption is much quicker. This means that the lowering of supply temperature and the corresponding increase of cooling effect have to be “paid” by quick decreases of both “thermal” and “hydric” COP’s.

This inconvenience is obviously reinforced by the fact that the regime is *turbulent* inside both heat exchangers (except on secondary side of heat exchanger 01).

In order to make a better use of this “mode 3”, one should *decrease* the useful air flow or *increase* the frontal sizes of both heat exchangers.

With present *sizing*, present *useful air flow rate*, present *reference air states* inside and outside the building and present (questionable) *hypotheses*, the all three modes “2” (“a”, “b” and “c”)appear as more interesting than the mode “3”.

4. Resizing

In order to make this IEC system more “competitive” in mode 3, one possibility consists in enlarging its front area:

Number of plates:

$Npl_{HE01} = 2 \cdot 119$ [-] $Npl_{HE02} = 2 \cdot 119$ [-]

New results, obtained in reference conditions, are presented in **Figures 8** to **11** and in **Table 2**.

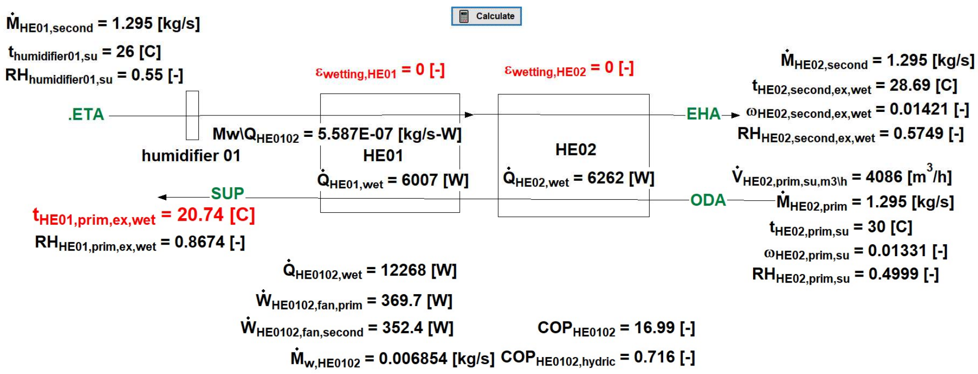


Figure 8: Mode2a *after resizing* (to be compared to **Figure 4**) (EES file: JL220404-03 JL220327-01 JL220326-02 HE01 and HE02 mode 2a resized)

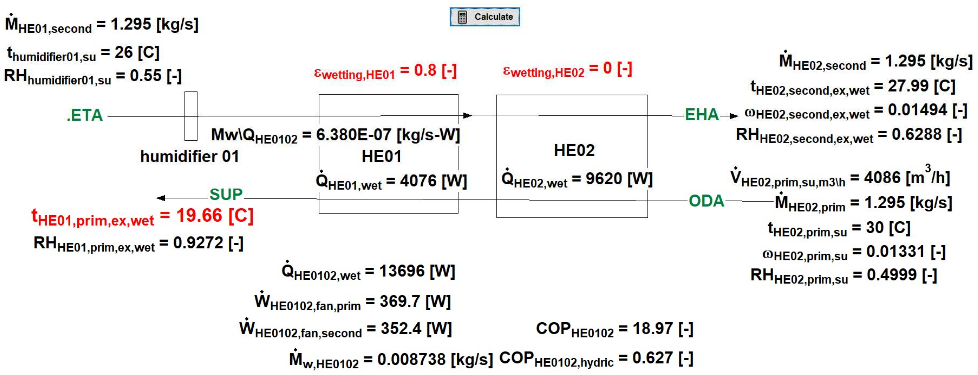


Figure 9: Mode 2b after resizing (to be compared to **Figure 5**) (EES file: JL220326-02 JL220322-03 HE01 and HE02 mode 2b)

The wetting of the secondary side of heat exchanger 01 is still producing some useful effect.

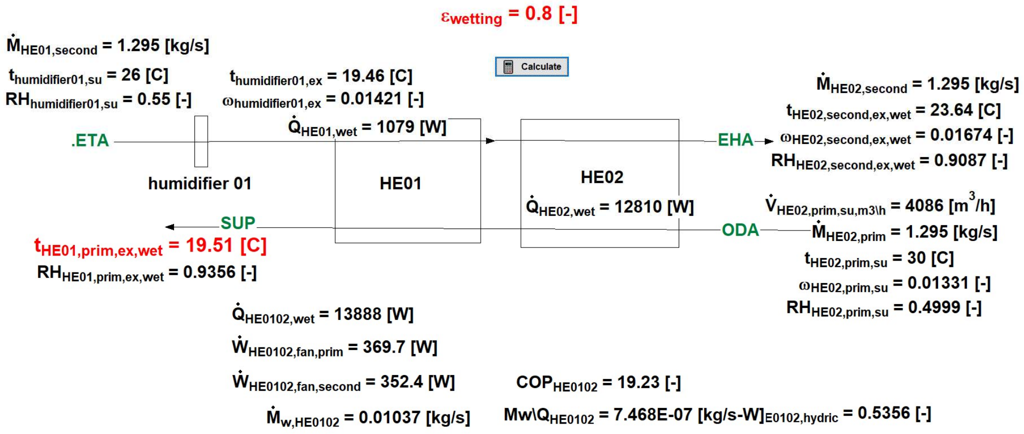


Figure 10: Mode 2c: *with wetting of secondary side of both heat exchangers* (to be compared with **Figure 6)** (EES file: JL220404-04 JL220322-03 HE01 and HE02 mode 2c resized)

Extending the surface wetting to heat exchanger 02 is no more producing any significant progress (both heat exchangers are already oversized)…

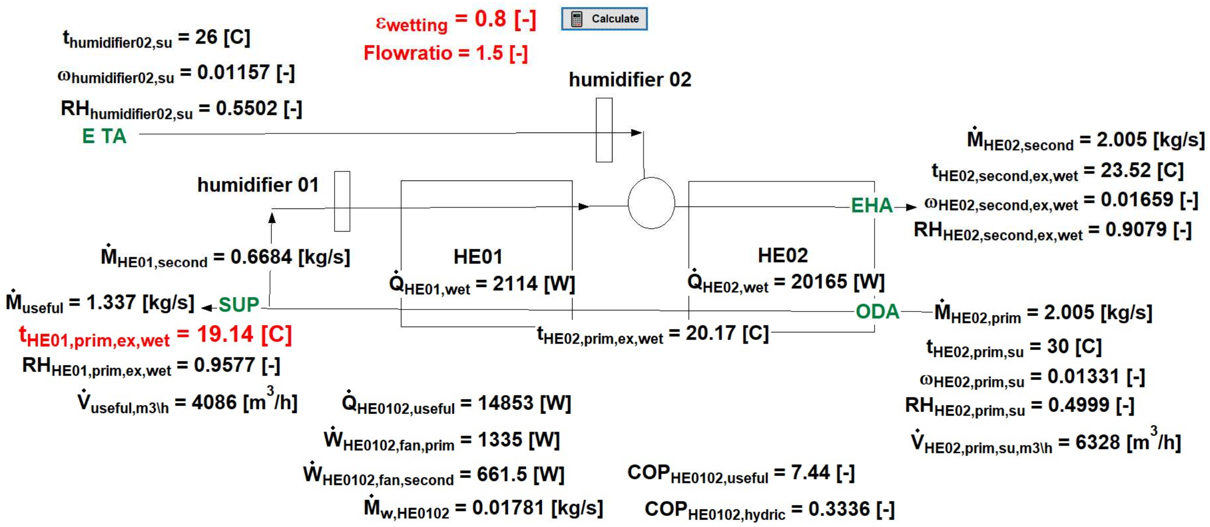


Figure 11: Mode 3 after resizing (to be compared to **Figure 7**)

(EES file: JL220404-01 JL220320-01 JL220218-02 IEC corrected resized).

Thanks to the last oversizing, this mode is now becoming “practicable”, but not much more efficient than the three modes “2”.

At this stage, it stays difficult to identify an optimal flow ratio: the cooling power, on one side, and both COP’s, on the other side, are still increasing and decreasing functions of this variable (**Table 2**).

Flowratio [-]	$t_{HE01,prim,ex,wet}$ [C]	$\dot{Q}_{HE0102,useful}$ [W]	$\dot{W}_{HE0102,fans}$ [W]	COP_{HE01} [-]	$\dot{M}_{w,HE0102}$ [kg/s]	COP_{HE01} [-]
1.01	20.06	13594	617.4	22.02	0.006907	0.7873
1.064	19.93	13775	720.3	19.12	0.008131	0.6777
1.119	19.8	13956	834.2	16.73	0.009354	0.5968
1.173	19.67	14131	959.7	14.72	0.01058	0.5344
1.228	19.55	14293	1098	13.02	0.01179	0.4848
1.282	19.44	14439	1249	11.57	0.01301	0.444
1.337	19.35	14567	1413	10.31	0.01421	0.4099
1.391	19.27	14678	1592	9.218	0.01542	0.3808
1.446	19.2	14772	1786	8.269	0.01662	0.3556
1.5	19.14	14853	1996	7.44	0.01781	0.3336

Table2: Effect of flow ratio after resizing (to be compared to **Table 1**) (EES file: JL220404-02 JL220326-01 IEC corrected resized parametric)

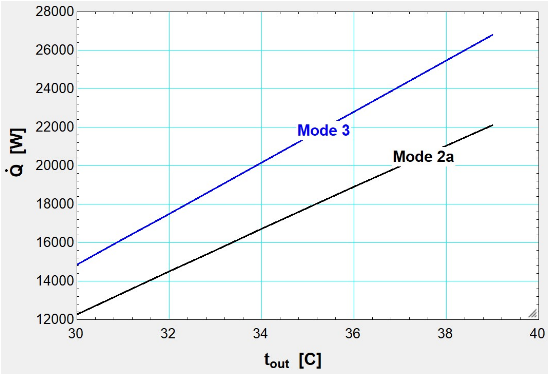
The effect of outdoor dry bulb (for same humidity ratio) in modes 2a and 3 is shown in **Tables 3** and **4** and in **Figures 12** to **14**.

$t_{HE02,prim,su}$ [C]	$RH_{HE02,prim,su}$ [-]	$t_{HE01,prim,ex,we}$ [C]	$\dot{Q}_{HE0102,wet}$ [W]	COP_{HE0102} [-]	$\dot{M}_{w,HE0102}$ [kg/s]	COP_{HE01} [-]
30	0.4999	20.74	12268	16.99	0.006854	0.716
31	0.4721	20.86	13390	18.67	0.006831	0.784
32	0.446	20.98	14505	20.37	0.006809	0.8521
33	0.4216	21.1	15612	22.08	0.006787	0.9202
34	0.3986	21.22	16713	23.8	0.006765	0.9883
35	0.3771	21.33	17807	25.54	0.006743	1.056
36	0.3568	21.45	18894	27.28	0.006721	1.124
37	0.3378	21.57	19974	29.04	0.006699	1.193
38	0.3199	21.69	21048	30.81	0.006678	1.261
39	0.3031	21.81	22114	32.59	0.006656	1.329

Table 3: Effect of outside dry bulb (with constant humidity ratio) in *mode 2a* (EES file: Mode 2a JL220404-06 JL220404-03 mode 2a effect of tout)

$t_{HE02,prim,su}$ [C]	$RH_{HE02,prim,su}$ [-]	$t_{HE01,prim,ex,we}$ [C]	$\dot{Q}_{HE0102,useful}$ [W]	$COP_{HE0102,use}$ [-]	$\dot{M}_{w,HE0102}$ [kg/s]	$COP_{HE0102,hyd}$ [-]
30	0.4999	19.14	14853	7.44	0.01781	0.3336
31	0.4721	19.17	16178	8.076	0.01828	0.354
32	0.446	19.2	17504	8.709	0.01876	0.3733
33	0.4216	19.23	18830	9.336	0.01923	0.3917
34	0.3986	19.26	20156	9.96	0.0197	0.4092
35	0.3771	19.29	21483	10.58	0.02018	0.4259
36	0.3568	19.32	22810	11.19	0.02065	0.4418
37	0.3378	19.35	24137	11.8	0.02112	0.4571
38	0.3199	19.38	25464	12.41	0.0216	0.4717
39	0.3031	19.41	26792	13.01	0.02207	0.4856

Table 4: Effect of outside dry bulb (with constant humidity ratio) in *mode 3* (EES file: JL220404-05 JL220404-01 mode 3 effect of tout)



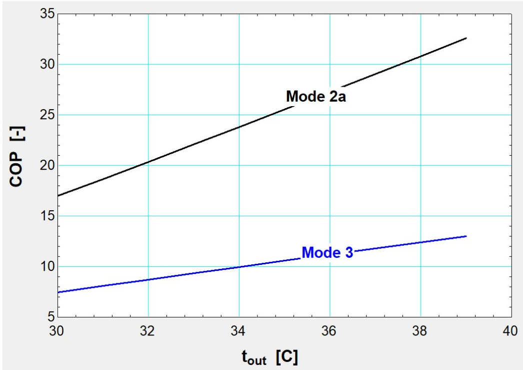


Figure 12 : Cooling power

Figure 13: Thermal COP

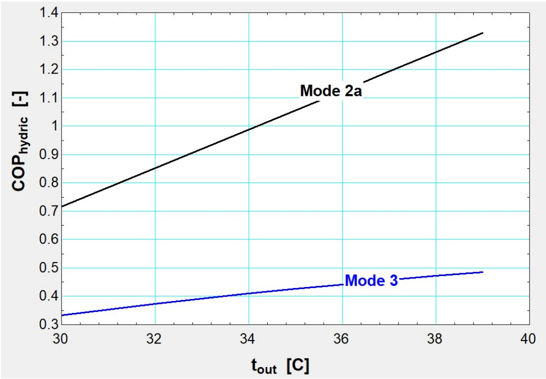


Figure 14: Hydric COP

It appears that the benefits of both modes are increasing functions of the outdoor temperature, but that these effects are stronger in mode 2a than in mode 3. The only advantage of mode 3 is an increase of cooling power, which has to be paid by some decreases of both COP's.

Such comparison could be pushed further by associating the IEC system to a building…

5. **The reference building of IEA-ECB Annex 59** [3], [4],[5],[6].

This is a 10 floors building (**Figure 15**) with large glazing areas and also heavy internal (occupancy, lighting and appliances) thermal loads.

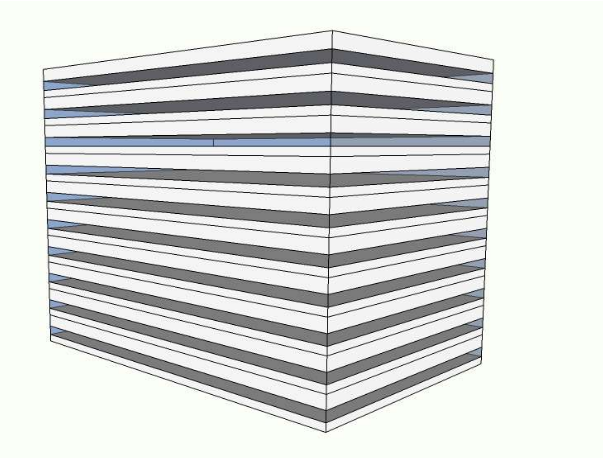


Figure 15: Axonometric view of the 10 floors medium size office building [3]

Each floor is subdivided into a set of 6 zones (4 peripheral, 1 central and 1 ceiling void). A detailed air flow rates analysis is established in each zone as indicated in **Figure 16**.

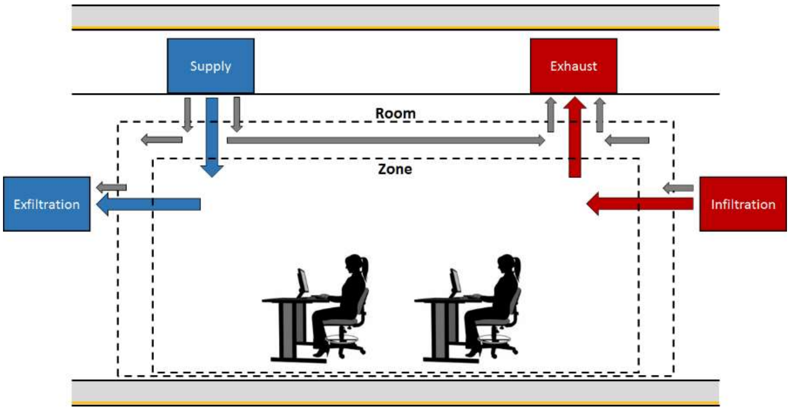


Figure 16: Room and zone air mass balance [4]

Simple R-C-R schemas are used to describe the thermal behaviour of all (internal and external) walls (**Figure 17**).

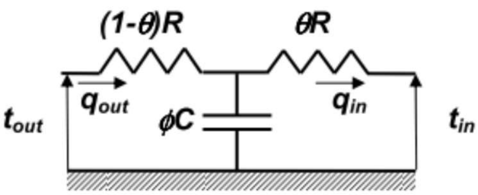


Figure 17: Capacitive wall representation, 2R-1C network [4]

All the windows are supposed to have a solar factor of 0.5 (which could be very much reduced!).

In nominal conditions, the building has a total of 971 occupants on a total of 16607 m2 (i.e. 0.058 occ/m2). Each occupant is suppoed to produce 70 W of sensible heat, 40 g/h of water and 40 g/h of CO2.

Also in nominal conditions, the building is supposed to be submitted to 215800 W (about 13 W/m2) of lighting and to 77760 W (about 7.8 W/m2) of appliances heat gains.

The total ventilation air flow rate in nominal conditions is fixed to 45360 m3/h (about 45 m3/h occ in average).

As shown in **Figure 18**, a complete HVAC system was supposed to be originally installed in this building.

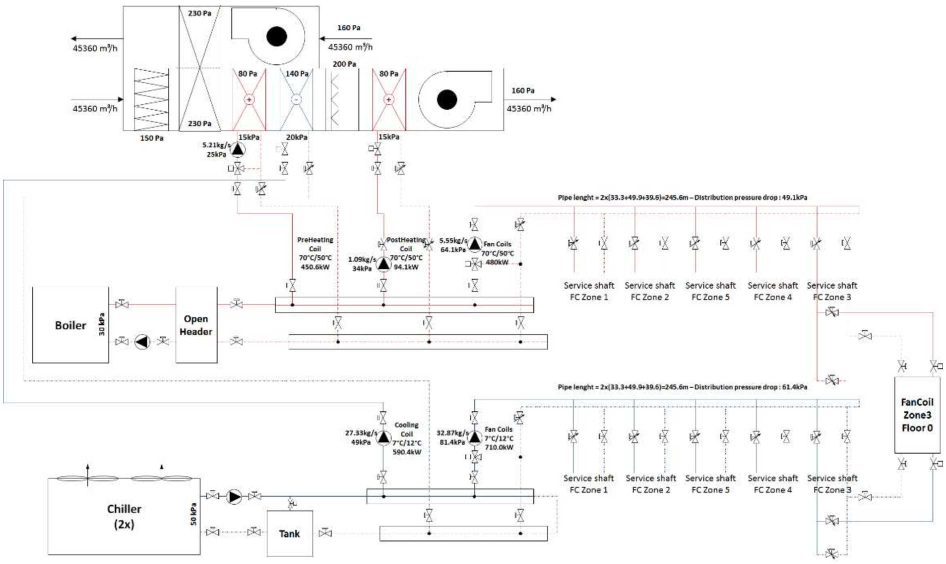


Figure 18: The original HVAC system [6]

A significant part of the cooling is required in order to control the relative humidity inside the building. This control is performed in the AHU. The post heating is also used to keep the air supply temperature to a comfortable level.

In the scope of the present study (and before adding an IEC to this system), three components are removed from the air handling unit: the recovery heat exchanger, the cooling coil and the post heating coil.

6. First examples of simulation results obtained with annex 59 building

The building is supposed to be located in Brussels and submitted to the weather of a reference year (**Figures 19** and **20**).

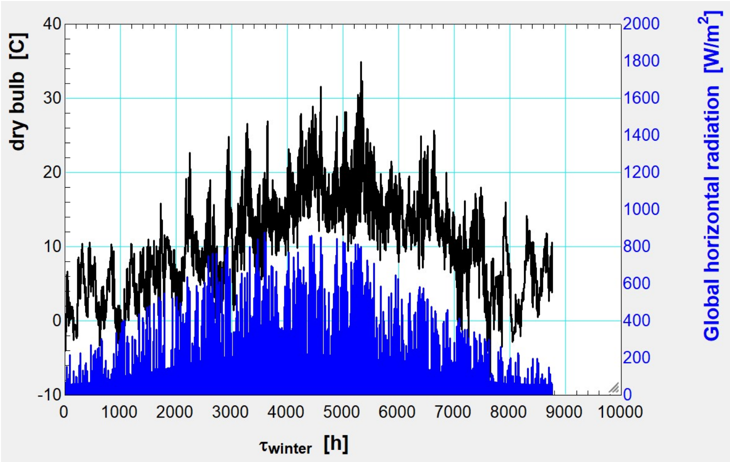


Figure 19: the Brussels reference year

A “zoom” is done on a “hot” 500 hours period (**Figure 20**).

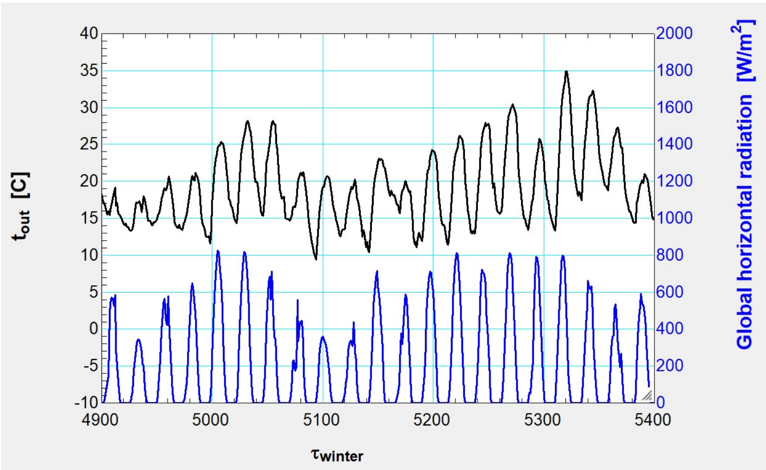


Figure 20: The 500 hours selected period

An typical example of occupancy profile is plotted in **Figures 21**.

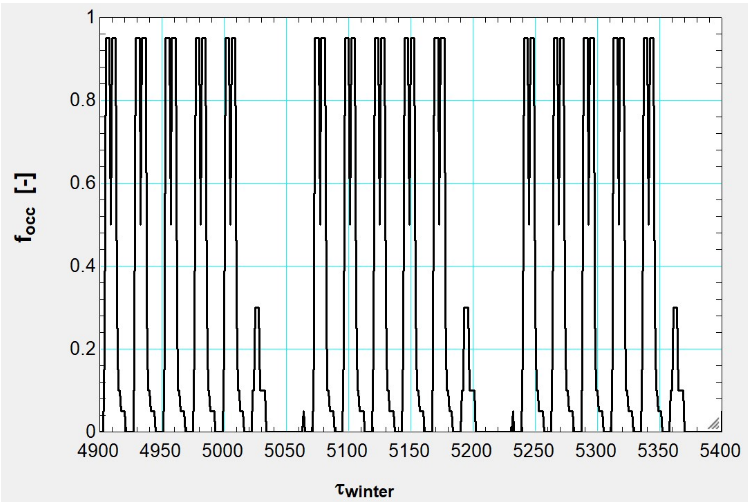


Figure 21: Occupancy factor in zone 1

Daily variations and weekends are taken into account.

Corresponding CO2 concentration, dry bulb temperature and relative humidity are plotted in **Figures 22** to **24**.

(EEES file: JL220308-01 JL220307-03 JL220304-02 without cooling and heating in AHU and reduced dp)

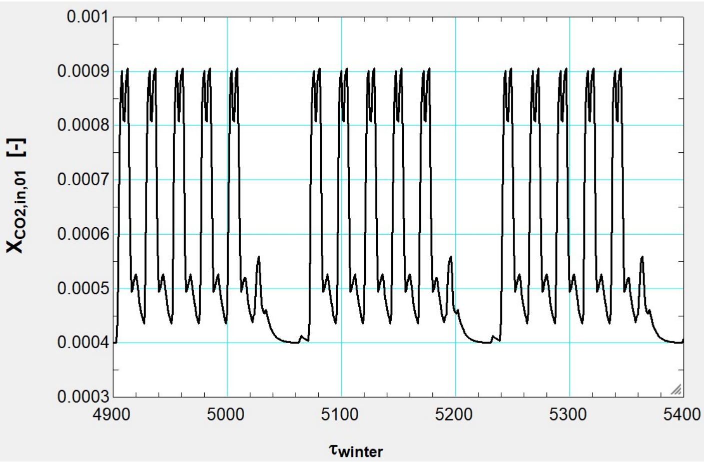


Figure 22: CO2 concentration in zone 01

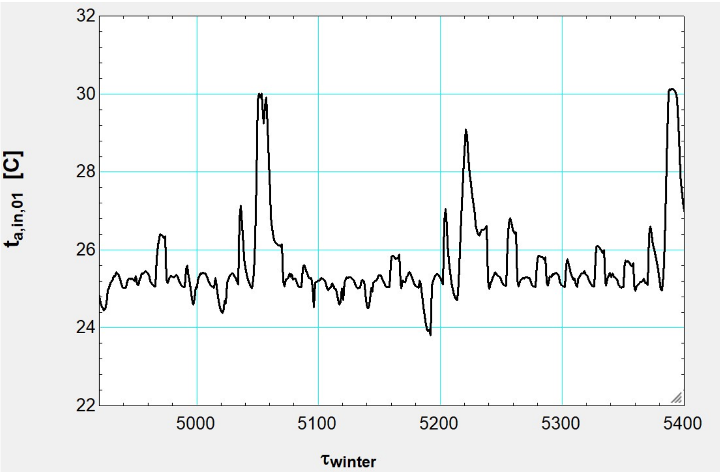


Figure 23: dry bulb temperature in zone 01

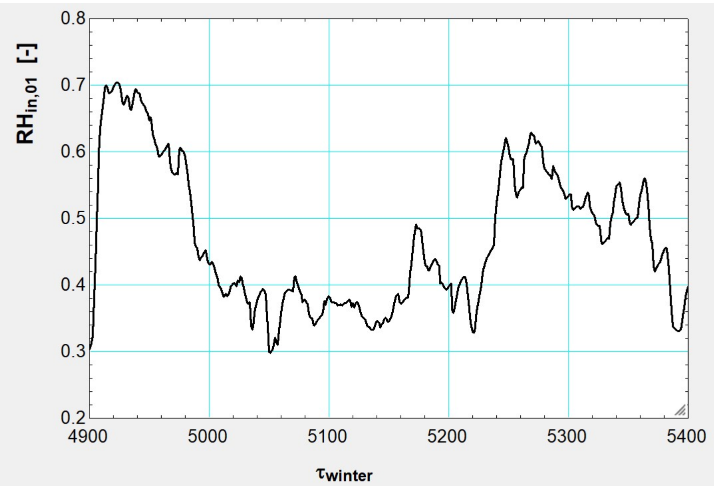


Figure 24: Relative humidity in zone 01

These results confirm that:

- the actual ventilation rate is sufficient to keep the CO2 concentration below 1000 ppm;
- the cooling power of the terminal unit is sufficient to keep the dry bulb temperature around 25 C during occupancy periods;
- the elimination of the cooling coil from the air handling unit makes that *there is no more any control of relative humidity*, which might generate some small inconvenience…

Total electrical consumption and chiller consumption are plotted in **Figure 25**.

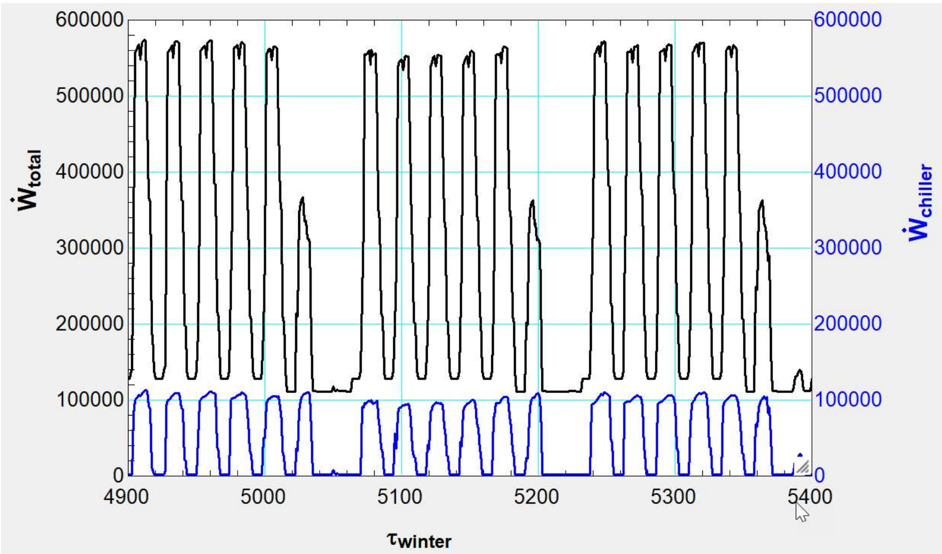


Figure 25: Total and chiller electrical consumptions

It appears that, during occupancy time, *the electrical power of the chiller represents less than 20 % of the total*. This is because of the very high lighting and appliances contributions, and also because of very significant “auxiliary” (fans and pumps) consumptions.

Main terms of the energy balance are presented in **Figure 26**.

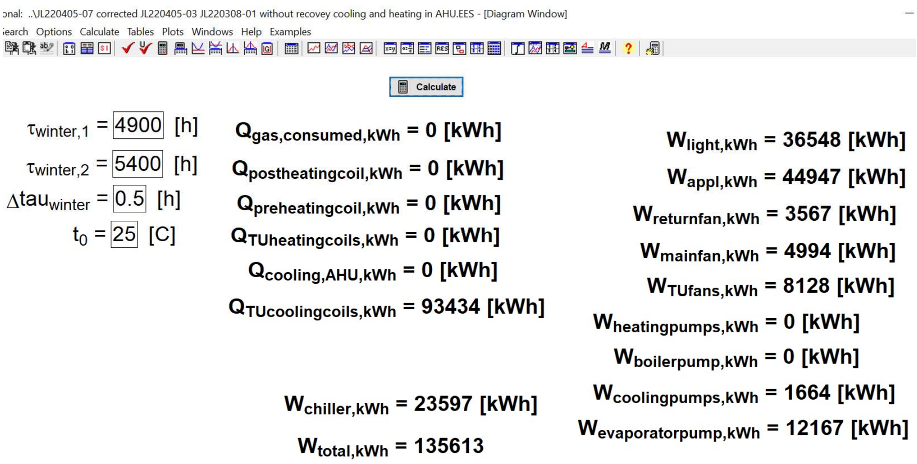


Figure 26: Energy balance established on the 500 h period without ICE (JL220405-07 corrected JL220405-03 JL220308-01 without recovery cooling and heating in AHU)

Results obtained with a set of *10 resized ICE units, used in mode 2a and connected to the existing AHU,* are presented in **Figure 27**.

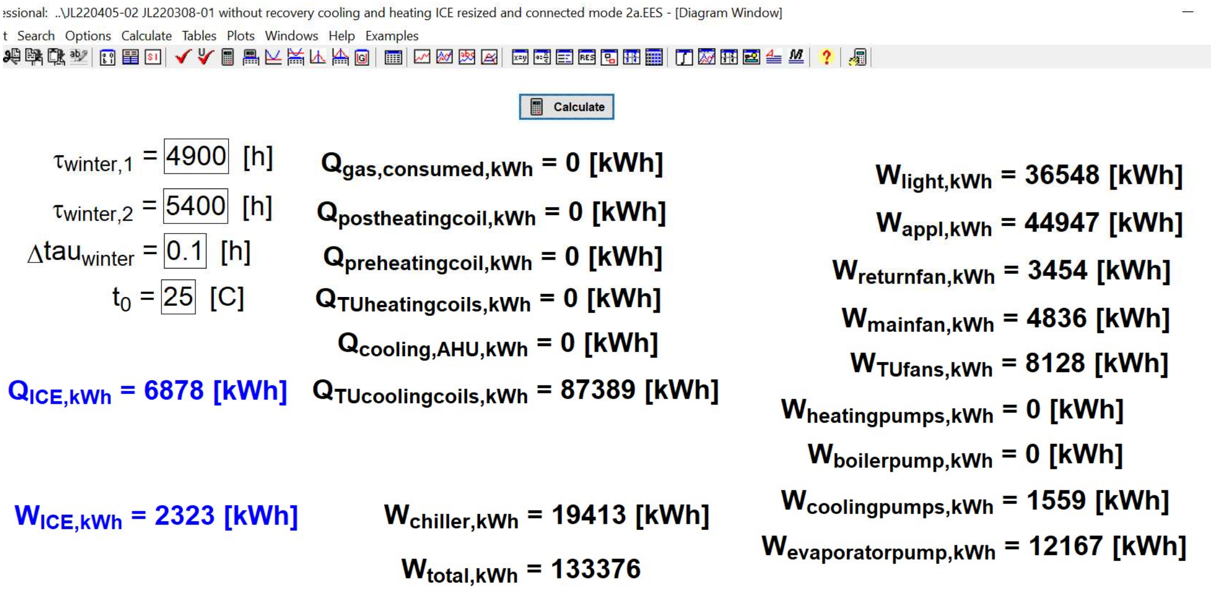


Figure 27: Energy balance established on the 500 h period with ICE in mode 2a

It appears that the addition of the 10 ICE units has not a spectacular effect on the electrical consumption of this building. The chiller consumption is actually reduced, but this effect is very modest in comparison with other terms of the energy balance (lighting, appliances, fans and pumps).

7. Conclusion

Such results are to be considered as no more than a very first provocation: not only because all building thermal loads could (and should) be very much reduced, but, even more, because all so-called “auxiliary” consumptions (fans and pumps mainly the evaporator pump) should be minimized, thanks to optimal sizing and optimal control, before going further…

6. Reference

[1] JL220119-01 JL211227-01 annex 85 step 3

[2] Study report CEREF Annex 85

[3] Annex59_ReferenceBuilding_POLITO_ULG_JCJ_140507

[4] Annex59_ReferenceBuildingModel_ULG_JCJ_140507

[5] Annex59_ReferenceBuildingLoads_ULG_JCJ_1400707

[6] ReferenceHVAC_ULG_JCJ_140708
