SIMPLIFIED SIMULATION OF IEC SYSTEMS

April 22, 2021

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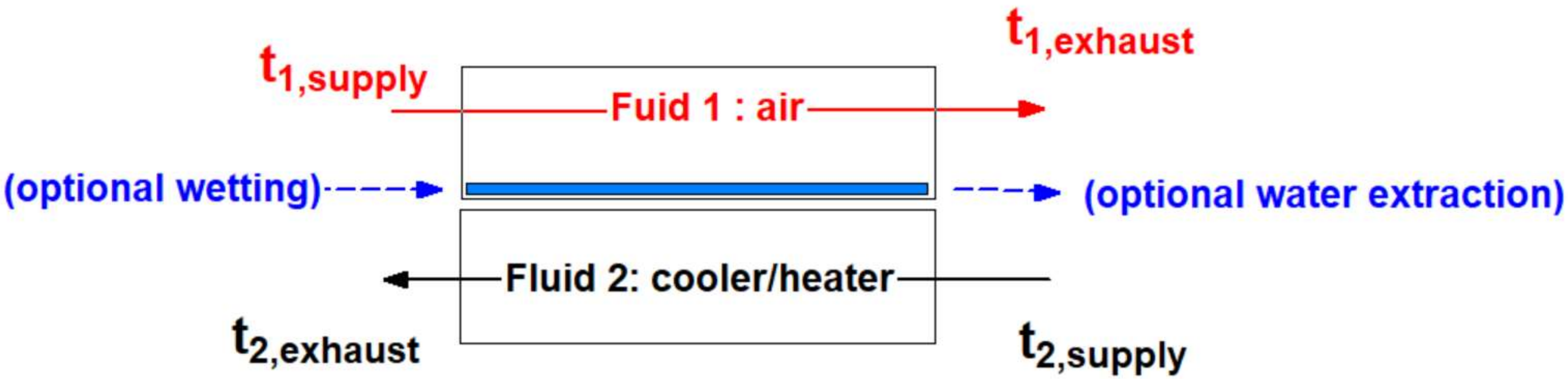
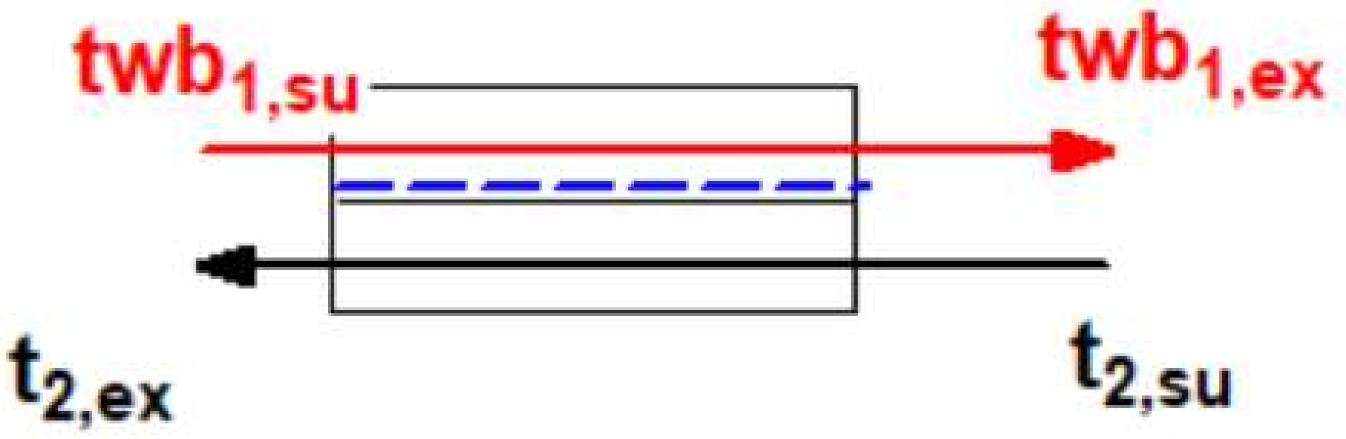
Air cooling/heating through a heat exchanger



A *wet* heat exchanger can be simulated by substituing a fictitious, pure and perfect gas to the *humid* air:



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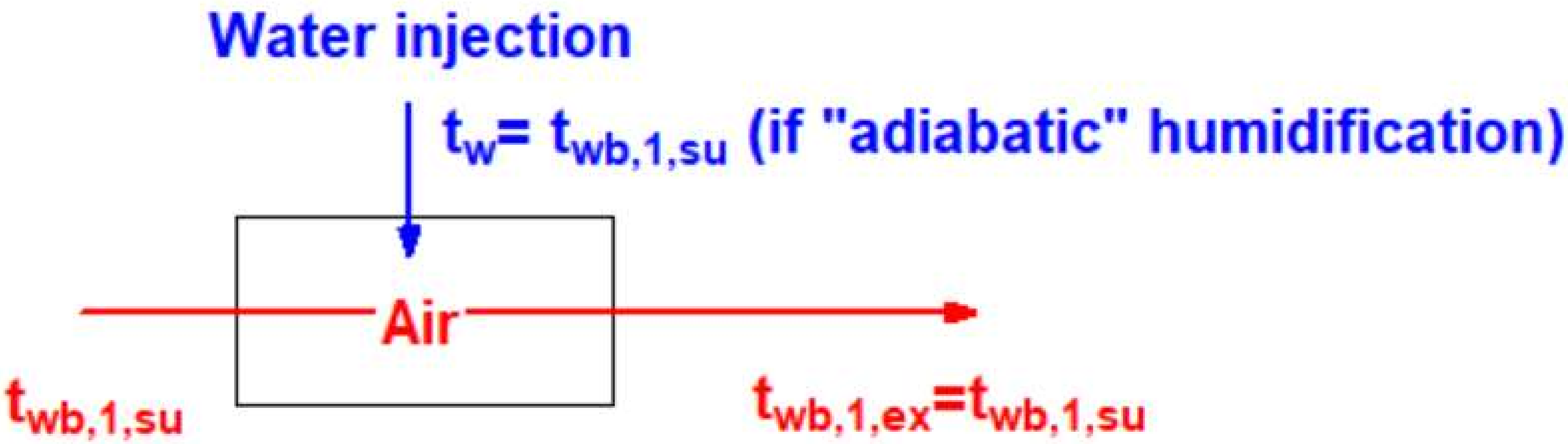
The « adiabatic » humidifyer is no more than a limit case:



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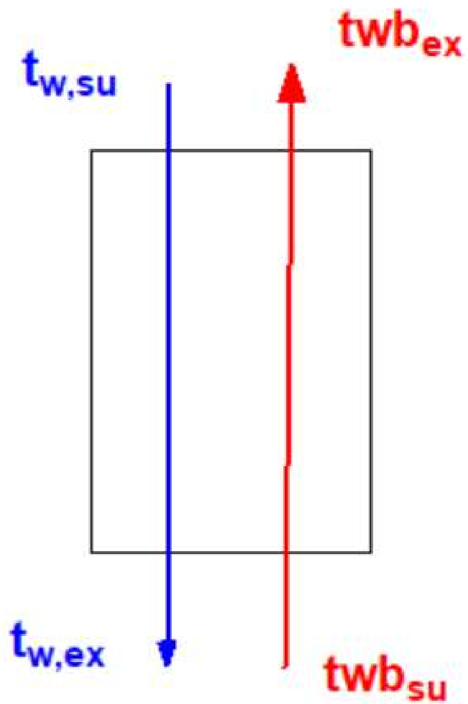
The « direct contact » cooling tower is another limit case:



Fictitious specific heat of the perfect gas substituted to the humid air:

$$c_{p,f,ct} = \frac{h_{ct,ex} - h_{ct,su}}{t_{wb,ct,ex} - t_{wb,ct,su}}$$

« Capacity » flow rates:

$$\dot{C}_{f,ct} = \dot{M}_{a,ct} \cdot c_{p,f,ct}$$

$$\dot{C}_{w,ct} = \dot{M}_{w,ct} \cdot c_{w}$$

$$\dot{C}_{min,f,ct} = \mathbf{Min}\,(\dot{C}_{w,ct}, \dot{C}_{f,ct})$$

$$\dot{C}_{max,f,ct} = \mathbf{Max}\,(\dot{C}_{w,ct}, \dot{C}_{f,ct})$$

$$\omega_{f,ct} = \frac{\dot{C}_{min,f,ct}}{\dot{C}_{max,f,ct}}$$

Fictitious heat transfer coefficient:

According to the Merckel's theory, the fictitious 'wet' heat transfer coefficient can be related to its 'dry' value as follows:

$$AU_{f,ct} = AU_{dry,ct} \cdot \frac{c_{p,f,ct}}{c_{p,ct}}$$

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$$NTU_{f,ct} = \frac{AU_{f,ct}}{\dot{C}_{min,f,ct}}$$

and, in the case of a counter flow arrangement,

$$\varepsilon_{f,ct} = \frac{1 - \exp(-NTU_{f,ct} \cdot (1 - \omega_{f,ct}))}{1 - \omega_{f,ct} \cdot \exp(-NTU_{f,ct} \cdot (1 - \omega_{f,ct}))}$$

$$\dot{Q}_{ct} = \varepsilon_{f,ct} \cdot \dot{C}_{min,f,ct} \, (t_{w,ct,su} - t_{wb,ct,su})$$

Water side energy balance:

$$\dot{Q}_{ct} = \dot{C}_{w,ct} \cdot (t_{w,ct,su} - t_{w,ct,ex})$$

(if neglecting here the water consumption, in fair approximation)

Reference variables used by practitioners :

$$approach = t_{w,ct,ex} - t_{wb,ct,su}$$

$$range = t_{w,ct,su} - t_{w,ct,ex}$$

Combined influences of both water and air flow rates on the global heat transfer coefficient:

$$AU_{dry,ct} = AU_{dry,ct,n} \cdot \left[\frac{\dot{M}_{w,ct}}{\dot{M}_{w,ct,n}}\right]^{m_{ct}} \cdot \left[\frac{\dot{M}_{a,ct}}{\dot{M}_{a,ct,n}}\right]^{n_{ct}}$$

Exponents of the heat transfer law:

m_{ct} = 0.05 [-]

n_{ct} = 0.6 [-]

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Air state at cooling tower exhaust:

$$h_{ct,ex} = h_{ct,su} + \frac{\dot{Q}_{ct}}{\dot{M}_{a,ct}}$$

Water consumption:

$$\dot{M}_{w,loss} = \dot{M}_{a,ct} \cdot \left(\omega_{ct,ex} - \omega_{ct,su}\right)$$

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Example of cooling tower

Nominal conditions:

$$\dot{M}_{a,ct,n} = 8 \quad [kg/s]$$

$$\dot{M}_{w,ct,n} = 12 \quad [kg/s]$$

$$AU_{dry,ct,n} = 15000 \quad [W/K]$$

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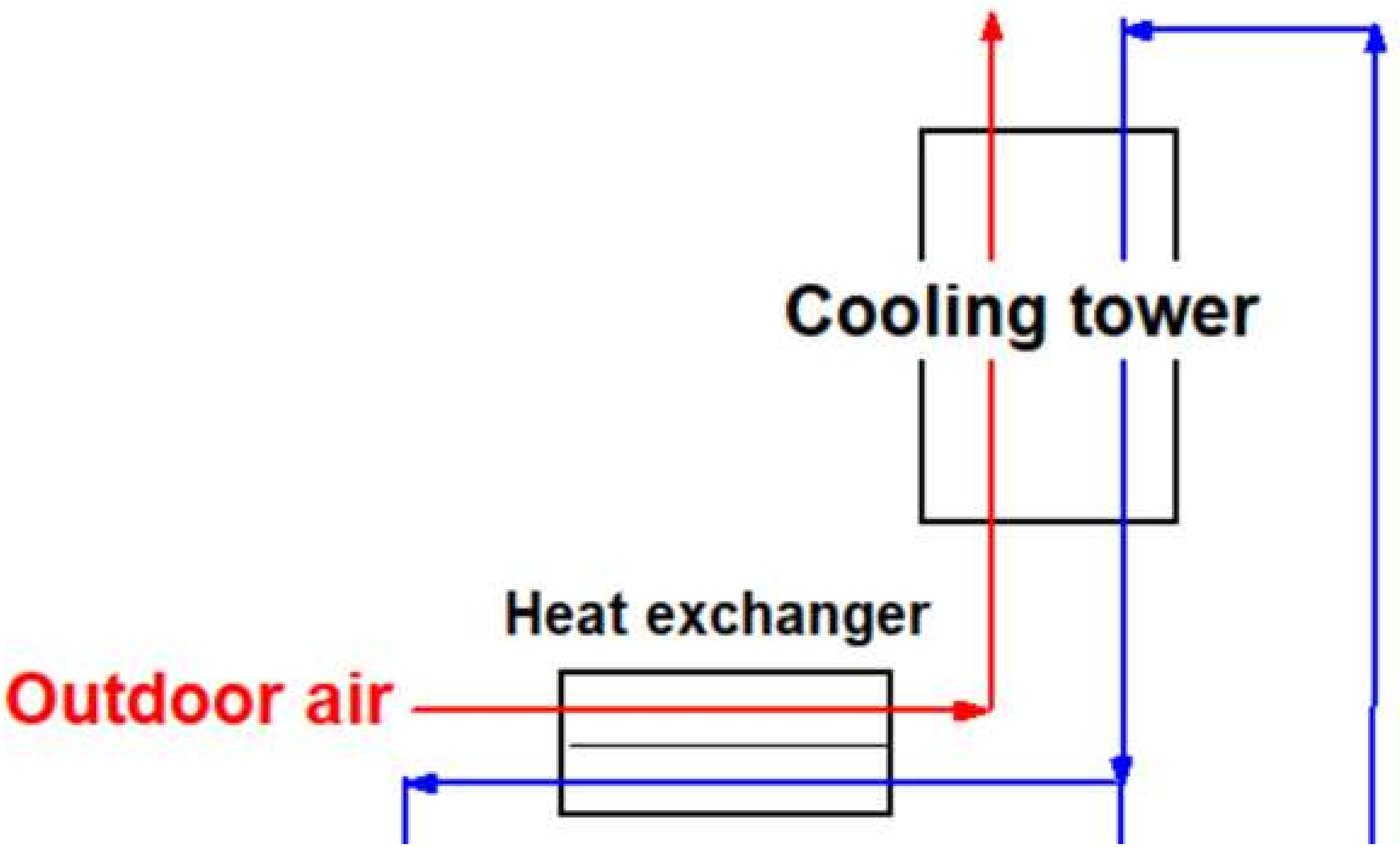
A Water-air heat exchanger can be used to pre-cool the air:



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Thermal resistances on air and water sides, global resistance and corresponding transfer coefficient:

$$R_{a,hx} = R_{a,hx,n} \cdot \left[\frac{\dot{M}_{a,hx,n}}{\dot{M}_{a,hx}}\right]^{m_{hx}}$$

$$R_{w,hx} = R_{w,hx,n} \cdot \left[\frac{\dot{M}_{w,hx,n}}{\dot{M}_{w,hx}}\right]^{n_{hx}}$$

$$R_{hx} = R_{a,hx} + R_{w,hx}$$

$$AU_{hx} = \frac{1}{R_{hx}}$$

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Parameters:

Nominal conditions:

$\dot{M}_{a,hx,n} = 20$ [kg/s]

$\dot{M}_{w,hx,n} = 5$ [kg/s]

$AU_{hx,n} = 50000$ [W/K]

$$R_{hx,n} = \frac{1}{AU_{hx,n}}$$

$$R_{a,hx,n} = \frac{R_{hx,n}}{2}$$

$$R_{w,hx,n} = \frac{R_{hx,n}}{2}$$

$m_{hx} = 0.8$ [-]

$n_{hx} = 0.8$ [-]

Inputs:

$\dot{M}_{a,hx} = 5$ [kg/s]

$\dot{M}_{w,hx} = 2$ [kg/s]

$t_{a,hx,su} = T_{outdoor}$

$\omega_{hx,su} = \omega_{outdoor}$

$t_{w,hx,su} = t_{w,ct,ex}$

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Capacity flow rates and NTU:

$$\dot{C}_{a,hx} = \dot{M}_{a,hx} \cdot c_p$$

$$\dot{C}_{w,hx} = \dot{M}_{w,hx} \cdot c_w$$

$$\dot{C}_{min,hx} = \textbf{Min}\,(\dot{C}_{w,hx}, \dot{C}_{a,hx})$$

$$\dot{C}_{max,hx} = \textbf{Max}\,(\dot{C}_{w,hx}, \dot{C}_{a,hx})$$

$$\omega_{hx} = \frac{\dot{C}_{min,hx}}{\dot{C}_{max,hx}}$$

$$NTU_{hx} = \frac{AU_{hx}}{\dot{C}_{min,hx}}$$

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Effectiveness (if counter flow) and water –air heat transfer:

$$\varepsilon_{hx} = \frac{1 - \exp(-NTU_{hx} \cdot (1 - \omega_{hx}))}{1 - \omega_{hx} \cdot \exp(-NTU_{hx} \cdot (1 - \omega_{hx}))}$$

$$\dot{Q}_{hx} = \varepsilon_{hx} \cdot \dot{C}_{min,hx} \cdot (t_{a,hx,su} - t_{w,hx,su})$$

Water side energy balance (gives exhaust water temperature):

$$\dot{Q}_{hx} = \dot{C}_{w,hx} \cdot (t_{w,hx,ex} - t_{w,hx,su})$$

Air side heat balance (gives exhaust air temperature):

$$\dot{Q}_{hx} = \dot{C}_{a,hx} \cdot (t_{a,hx,su} - t_{a,hx,ex})$$

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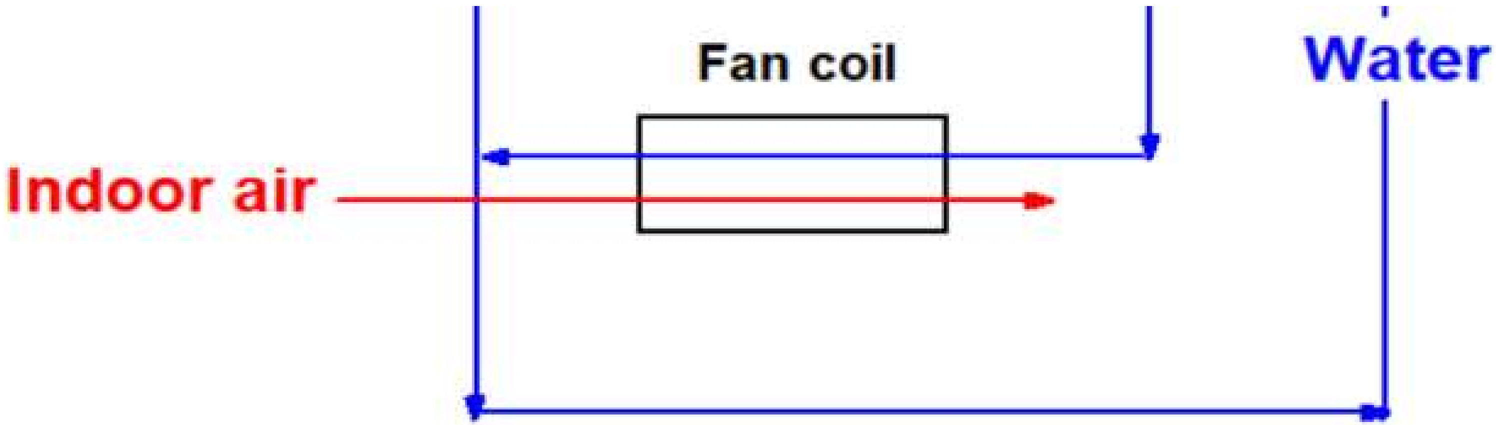
Let's assume that one (or several) fan coil(s) is (or are) used to cool the indoor air:



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$$\dot{Q}_{fc} = \varepsilon_{fc} \cdot \dot{C}_{min,fc} \cdot (t_{a,fc,su} - t_{w,fc,su})$$

Water side energy balance (gives exhaust water temperature):

$$\dot{Q}_{fc} = \dot{C}_{w,fc} \cdot (t_{w,fc,ex} - t_{w,fc,su})$$

Air balance (gives the exhaust air remperature):

$$\dot{Q}_{fc} = \dot{C}_{a,fc} \cdot (t_{a,fc,su} - t_{a,fc,ex})$$

Parameter:

$\varepsilon_{fc} = 0.7$ [-]

Inputs:

$\dot{M}_{w,fc} = 1$ [kg/s]

$t_{a,fc,su} = 24$ [C]

$t_{w,fc,su} = t_{w,ct,ex}$

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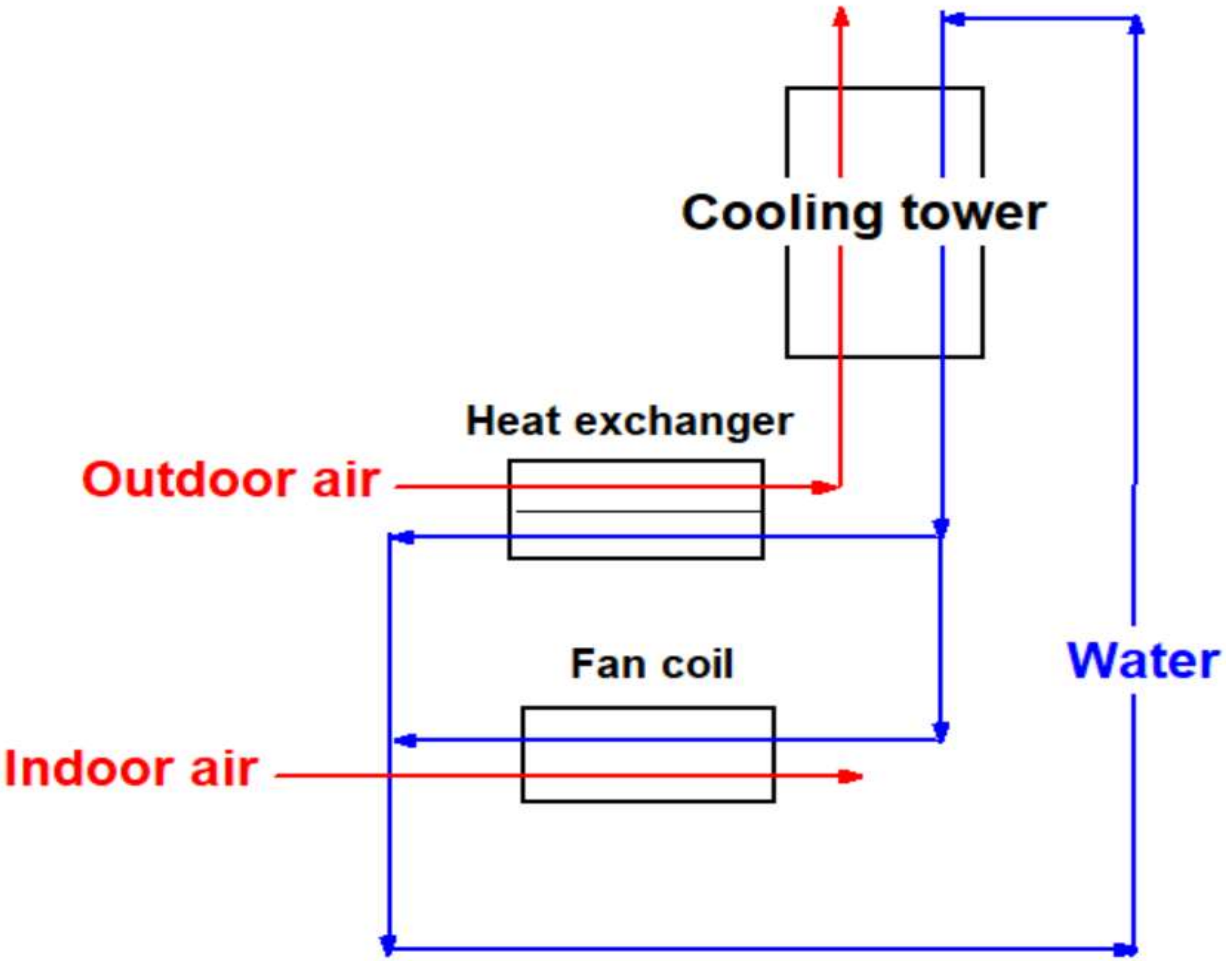
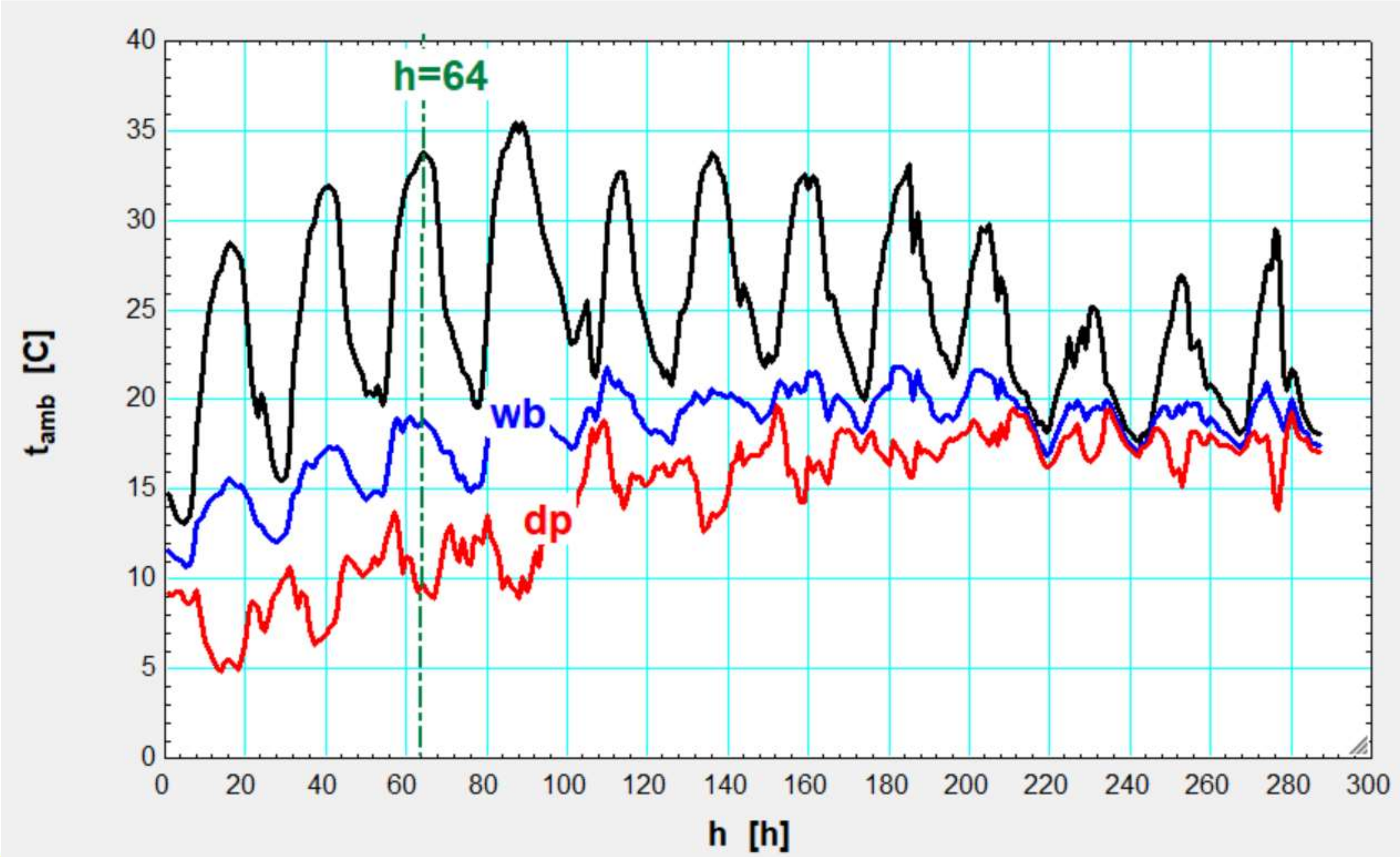
Global system considered:



Uccle hot wave of 2020:



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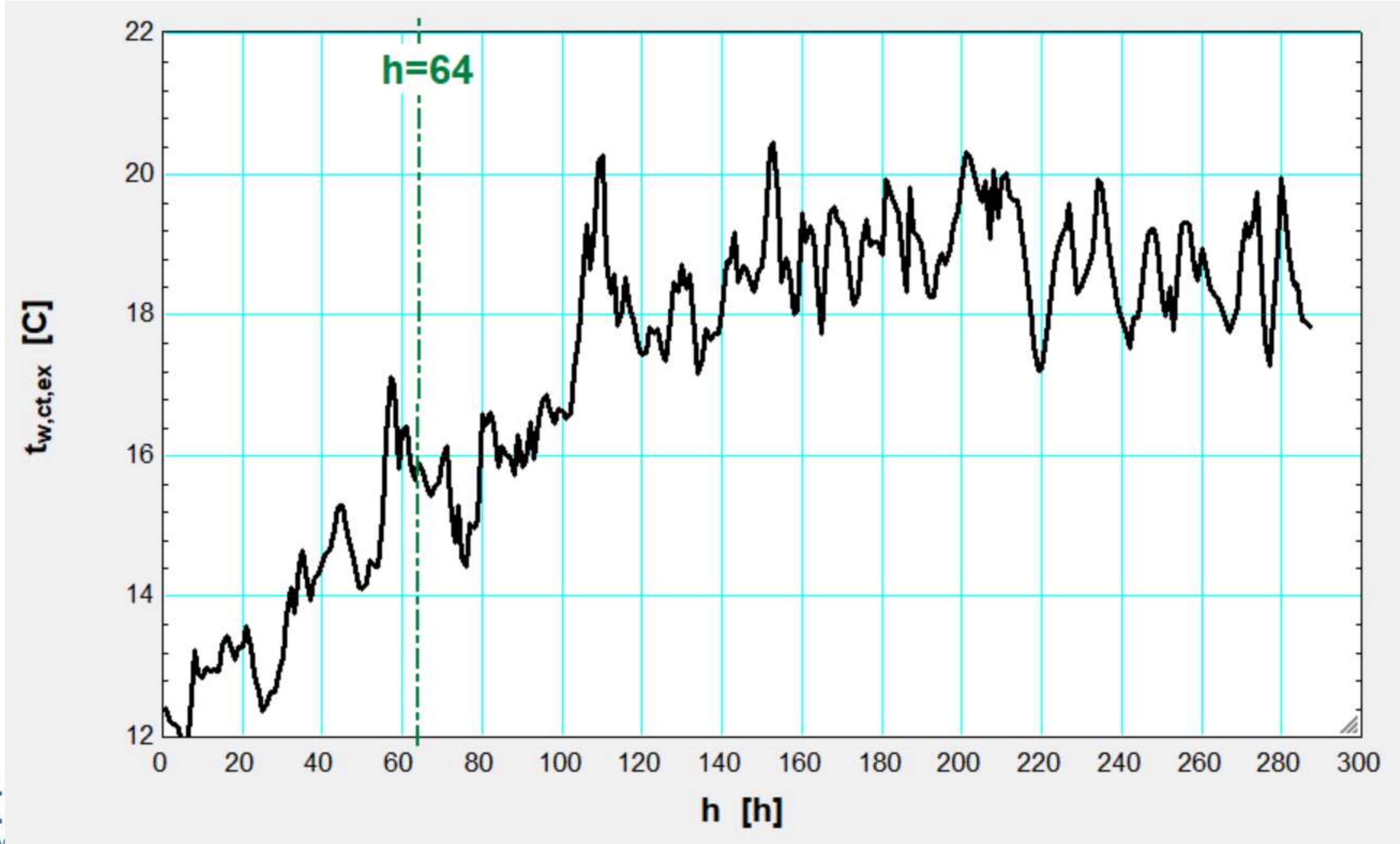
Water temperature at cooling tower exhaust:



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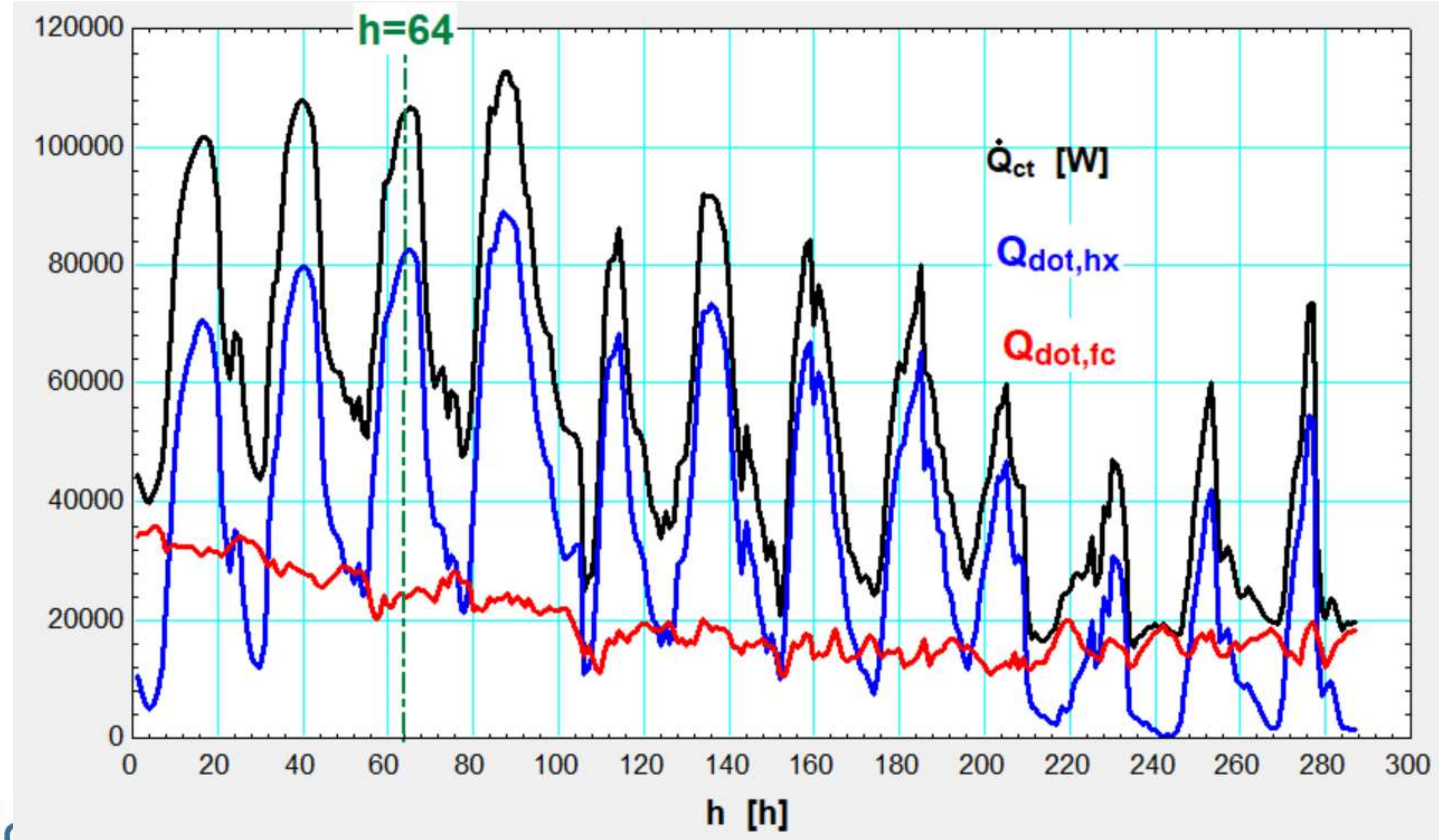
Thermal powers delivered by cooling tower, heat exchanger and fan coil(s)



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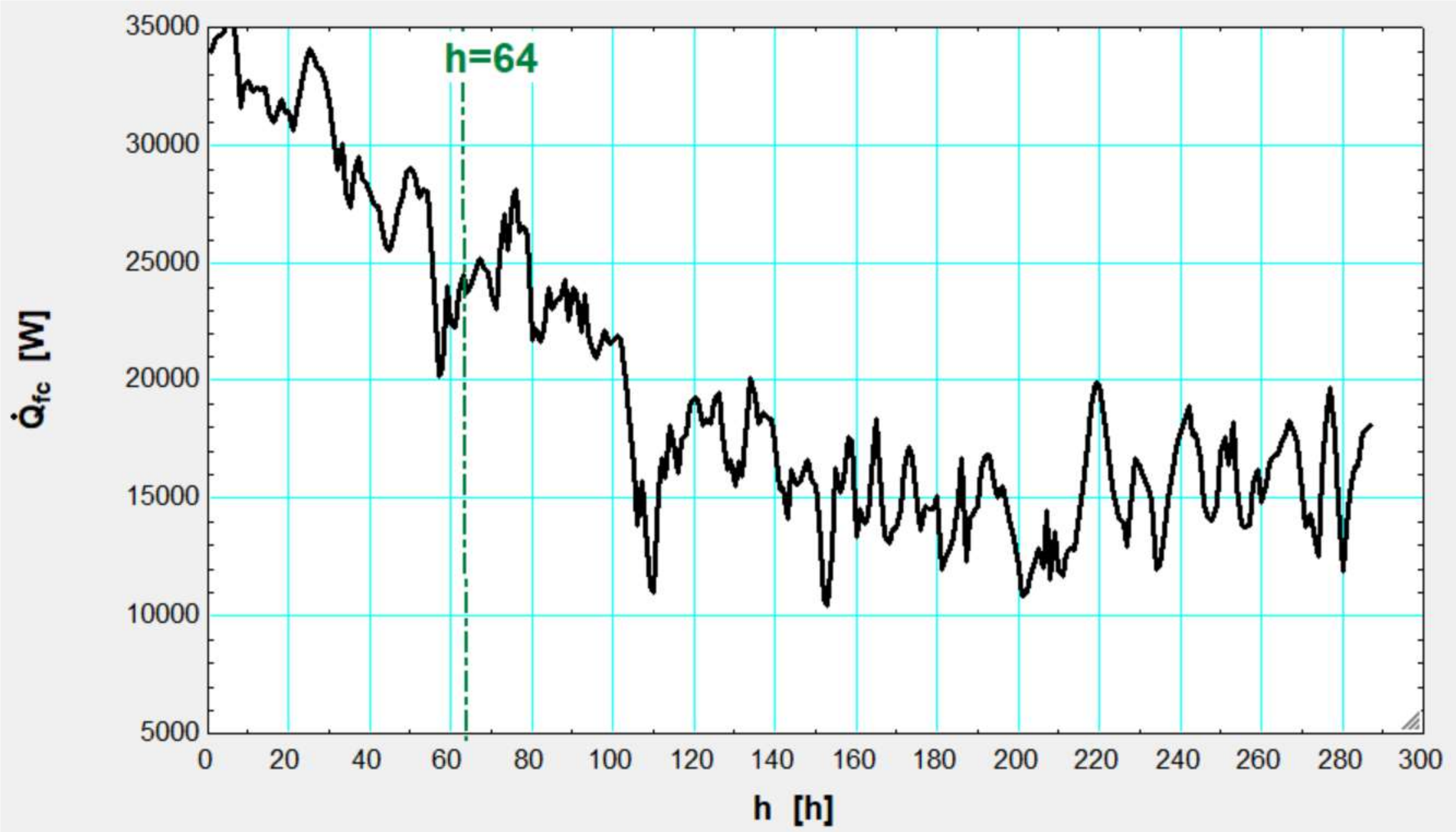
Useful cooling power:



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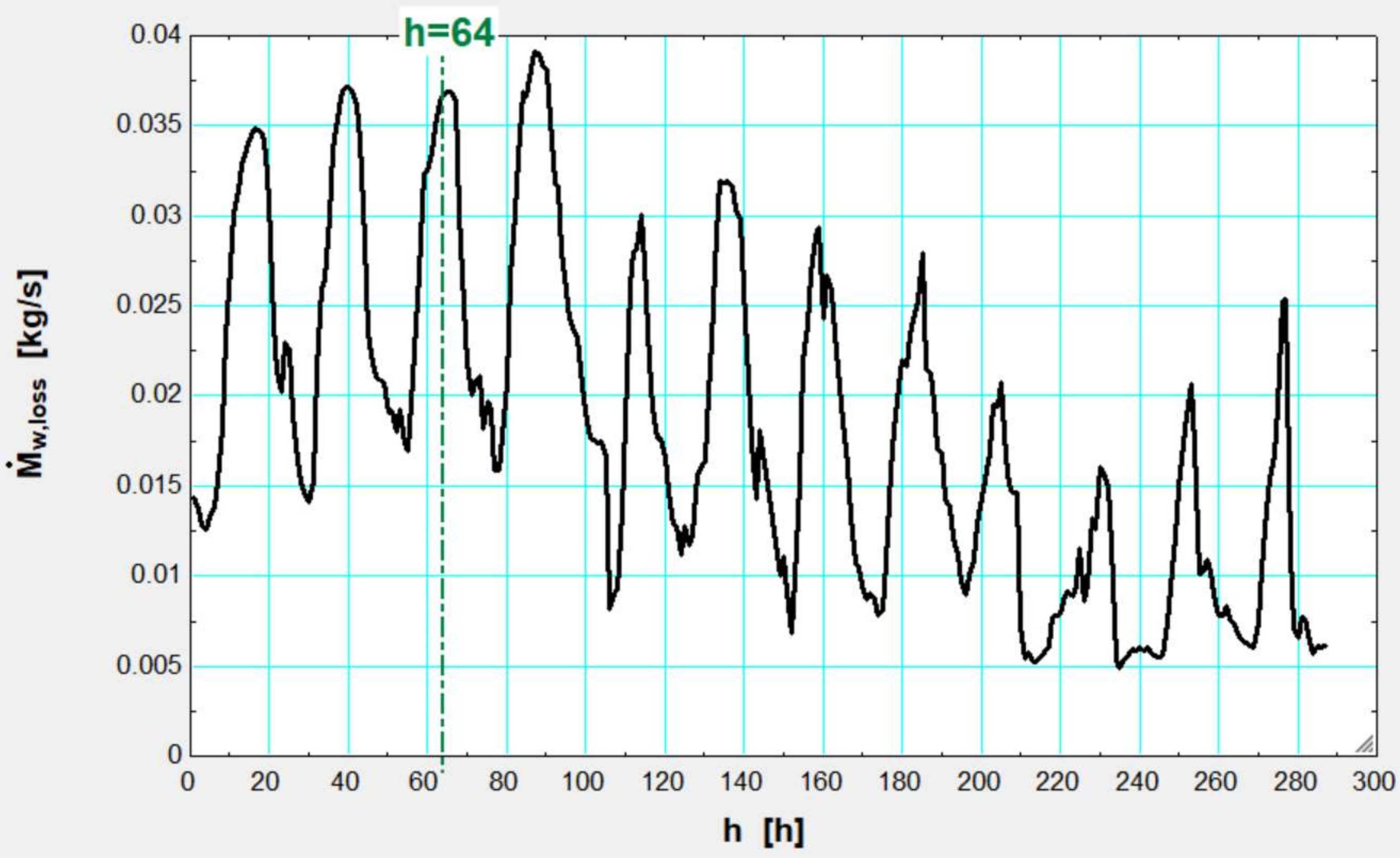
Water consumption:



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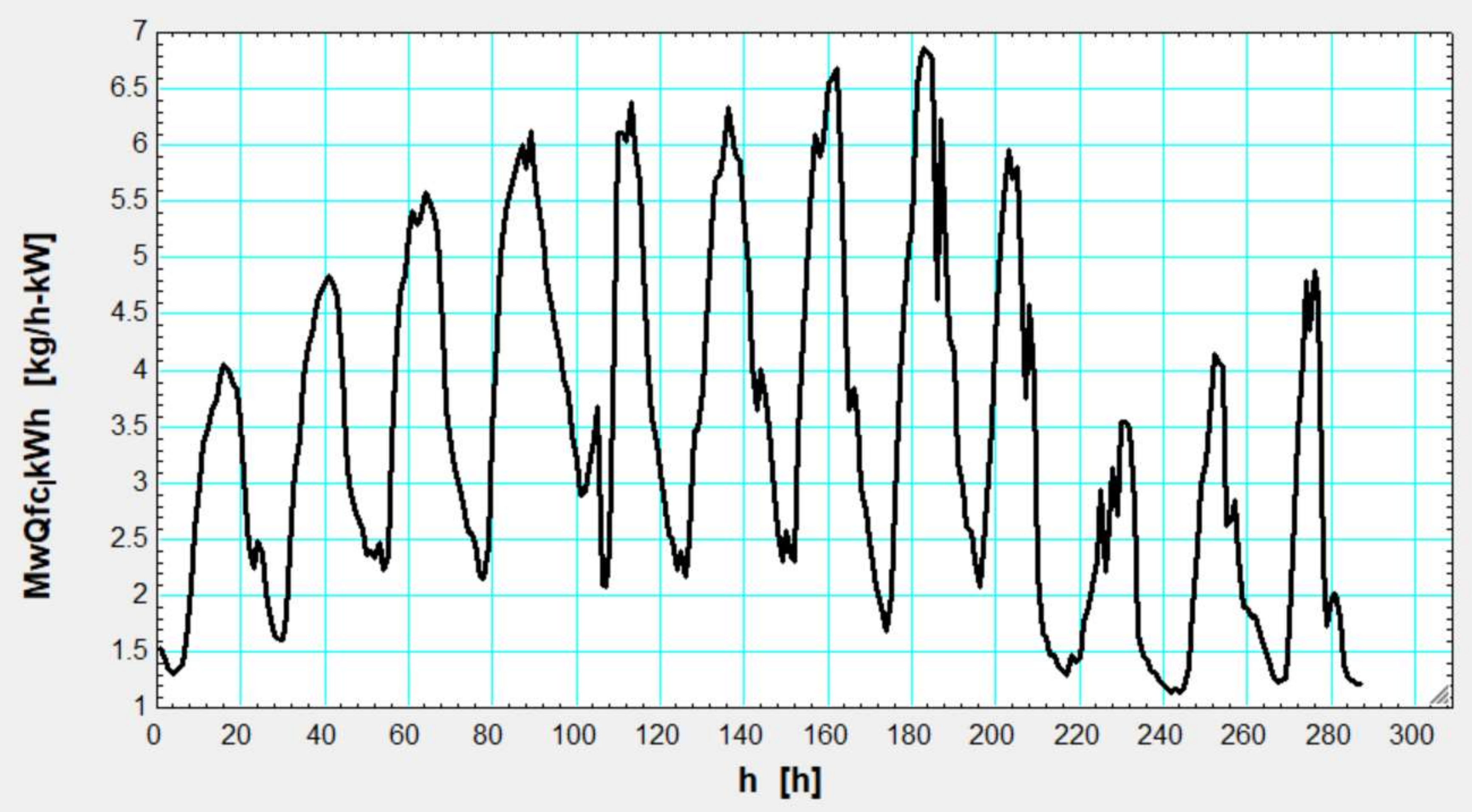
Specific water consumption in kg/kWh



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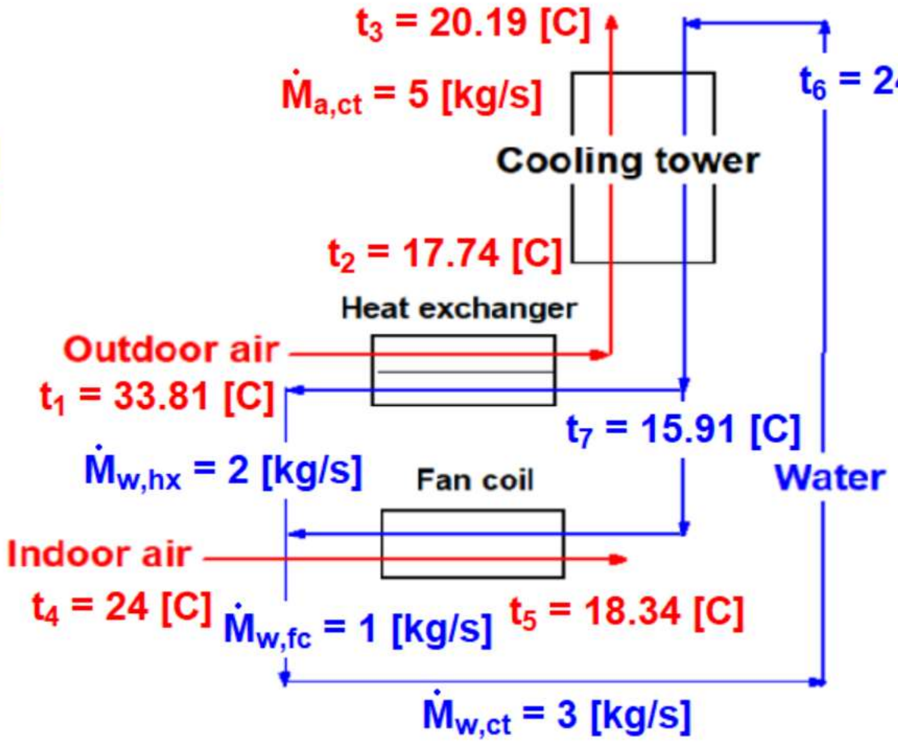
Main results at h=64



$\dot{Q}_{ct}$ = 105925 [W]

$\dot{Q}_{hx}$ = 82219 [J-C/s-K]

$\dot{Q}_{fc}$ = 23707 [W]

$\dot{M}_{w,loss}$ = 0.03672 [kg/s]

$Mw\backslash Qfc_{l\backslash kWh}$ = 5.577 [kg/kW-h]

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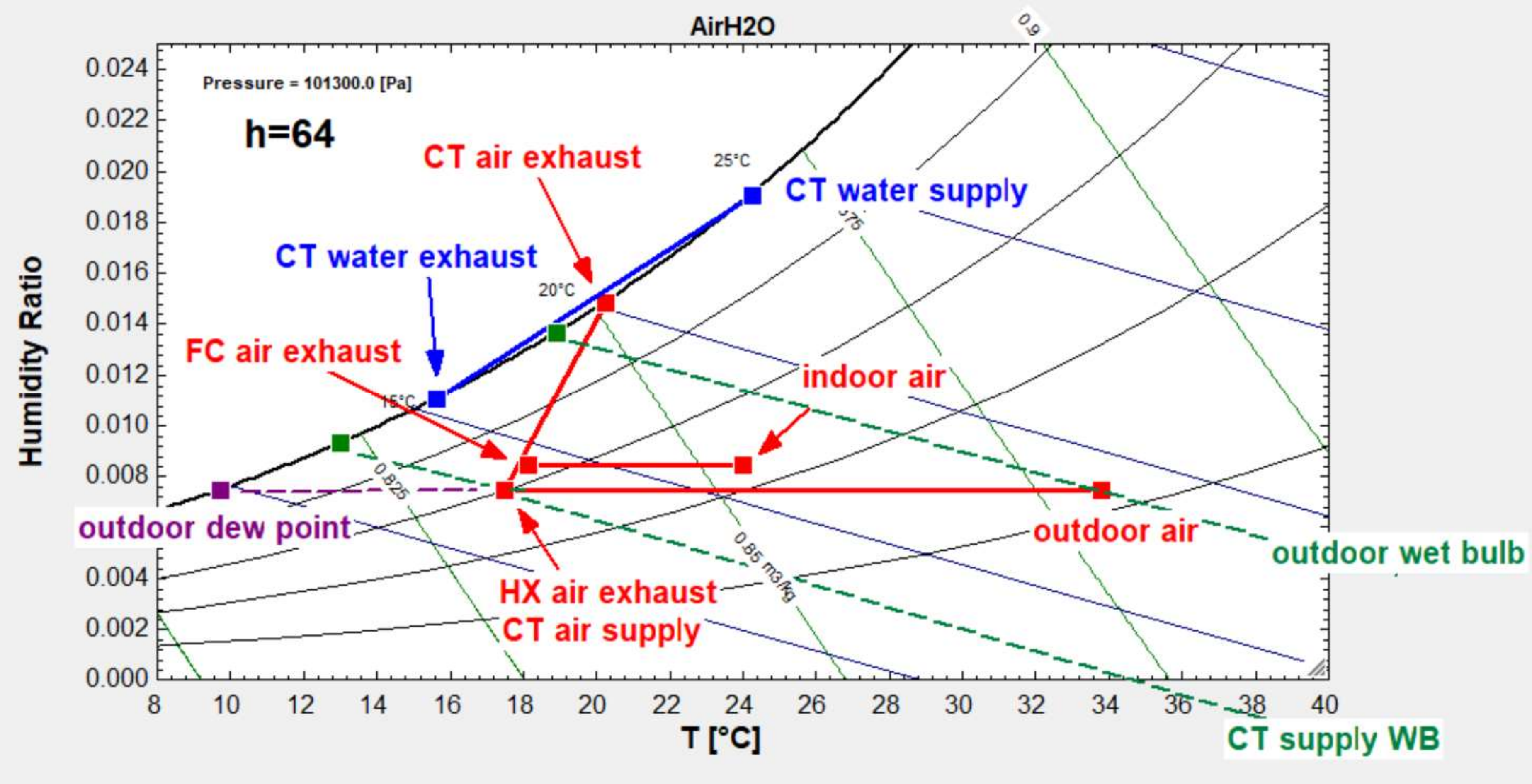
Air and water states at h =64



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Conclusions

This is no more than a very preliminary analysis!

It seems working, even (and mainly) during hot waves.

But all optimization has still to be performed with consideration to:

- actual sizings;
- « auxiliary » consumptions (fans and pumps!)
- Water consumption
- Etc.

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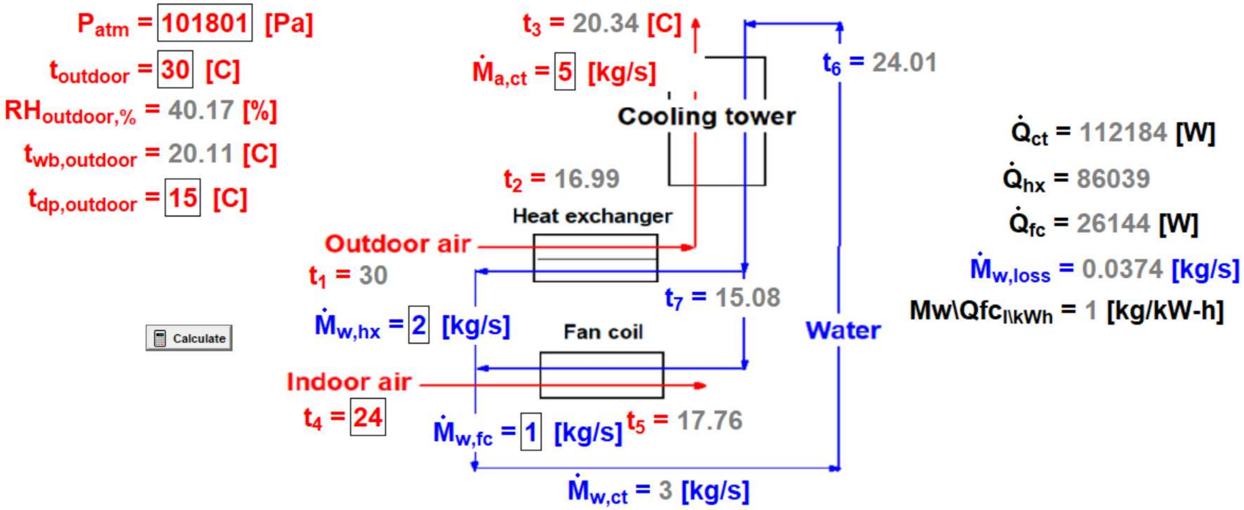
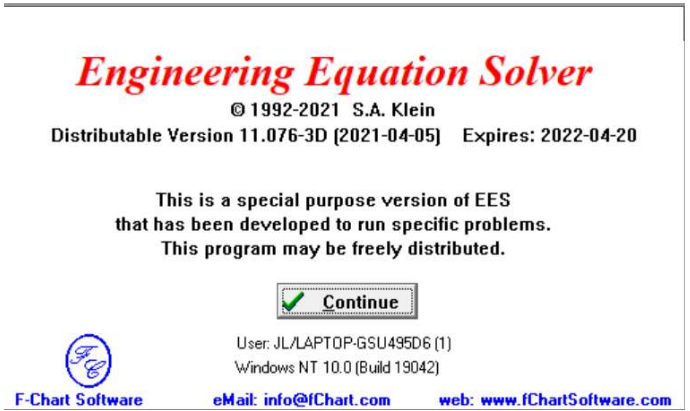
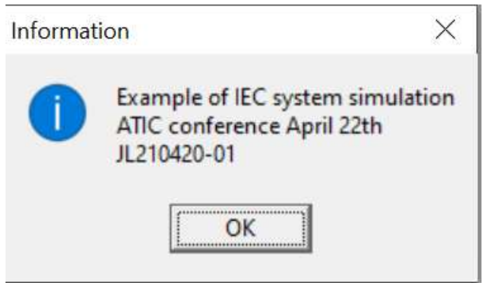
Thank you for your attention
And, if you are interested,…







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