STUDIE DAY ON 16 NOVEMBER 2023

Simulation of an IEC chiller

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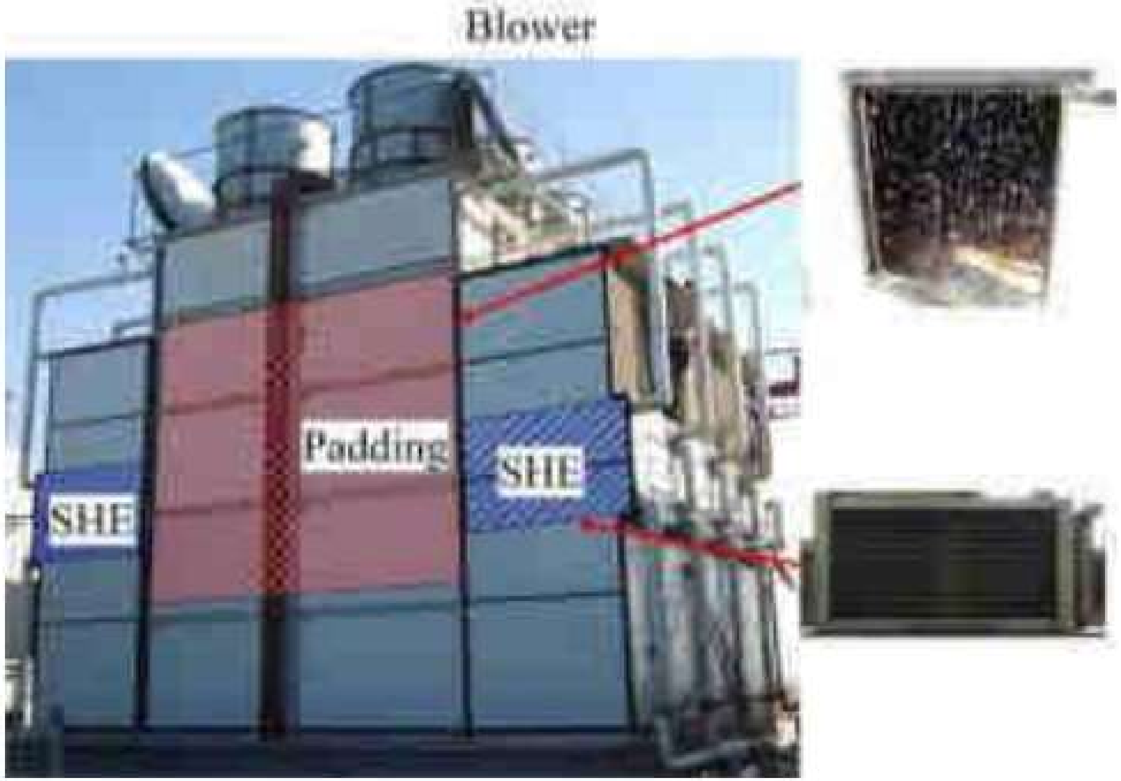
Fresh air flow rate (kg/s)
6.64

Exhaust air fan (kW)	By-pass water pump (kW)	Supply water pump (kW)
5.4	1.2	3.5



141410 m^3/h

Experimental results of pump and fans power consumption.

Component	Power (kW)
Pump	16.66
Fan 1	6.93
Fan 2	6.96
Fan 3	7.03
Fan 4	6.76
Fan 5	6.74
Fan 6	6.72
Total	57.80

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Heights of tested paddings (m)	Electricity consumption of fan(kW)	Total air flow rate (m3/h)
3	7.46	28840

Air velocity of air cooler (m/s)	Air velocity of paddings (m/s)
2.0	2.0

Total pressure drop of paddings (Pa)	53
Pressure drop of air coolers with 8 rows (Pa)	101
Other local resistance, like air turning, et al.(Pa)	39

Information contained in the ASHRAE "primary" toolkit

Nominal conditions

Wet bulb at tower supply:

$$t_{wb,CTjk,su,n} = 25.6 \text{ [C]}$$

$$Approach_{CTjk,n} = 3.8 \text{ [K]}$$

with

$$Approach_{CTjk,n} = t_{lw,CTjk,ex,n} - t_{wb,CTjk,su,n}$$

$$Range_{CTjk,n} = 5.6 \text{ [K]}$$

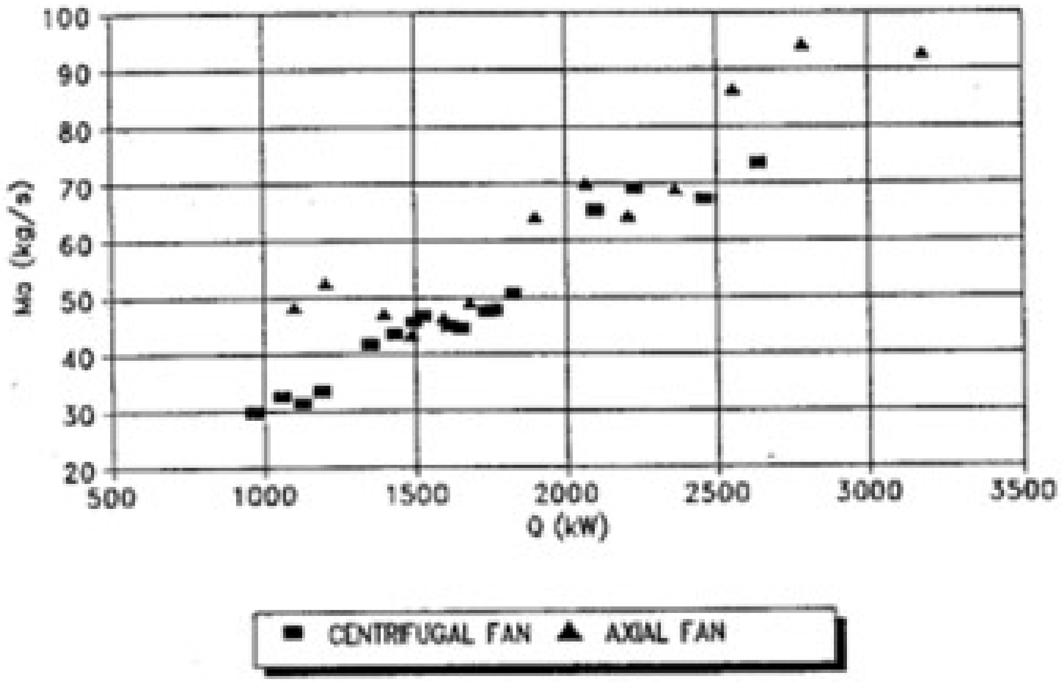
with

$$Range_{CTjk,n} = t_{lw,CTjk,su,n} - t_{lw,CTjk,ex,n}$$

Nominal liquid water flow rate

$$\dot{M}_{lw,CTjk,n} = \frac{\dot{Q}_{CTjk,n}}{c_{lw} \cdot Range_{CTjk,n}}$$

Nominal dry air flow rate



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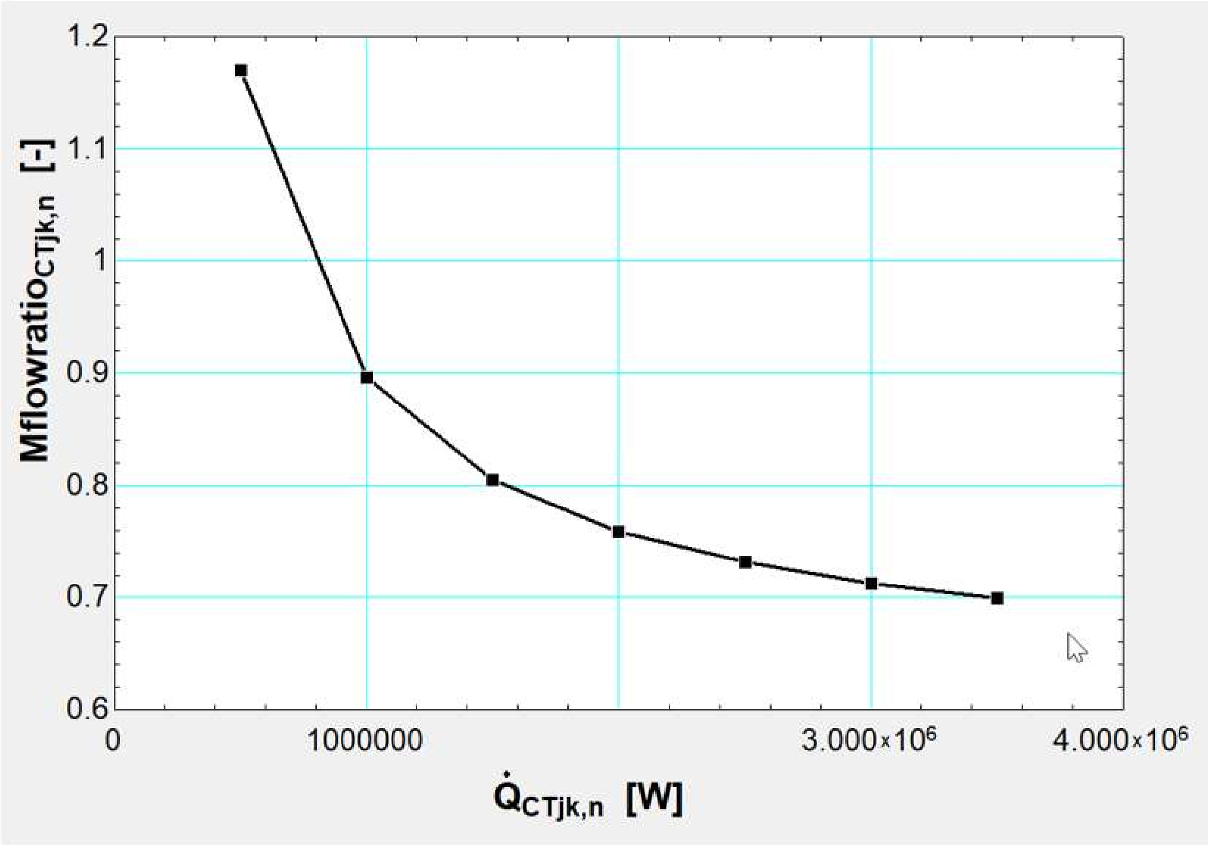
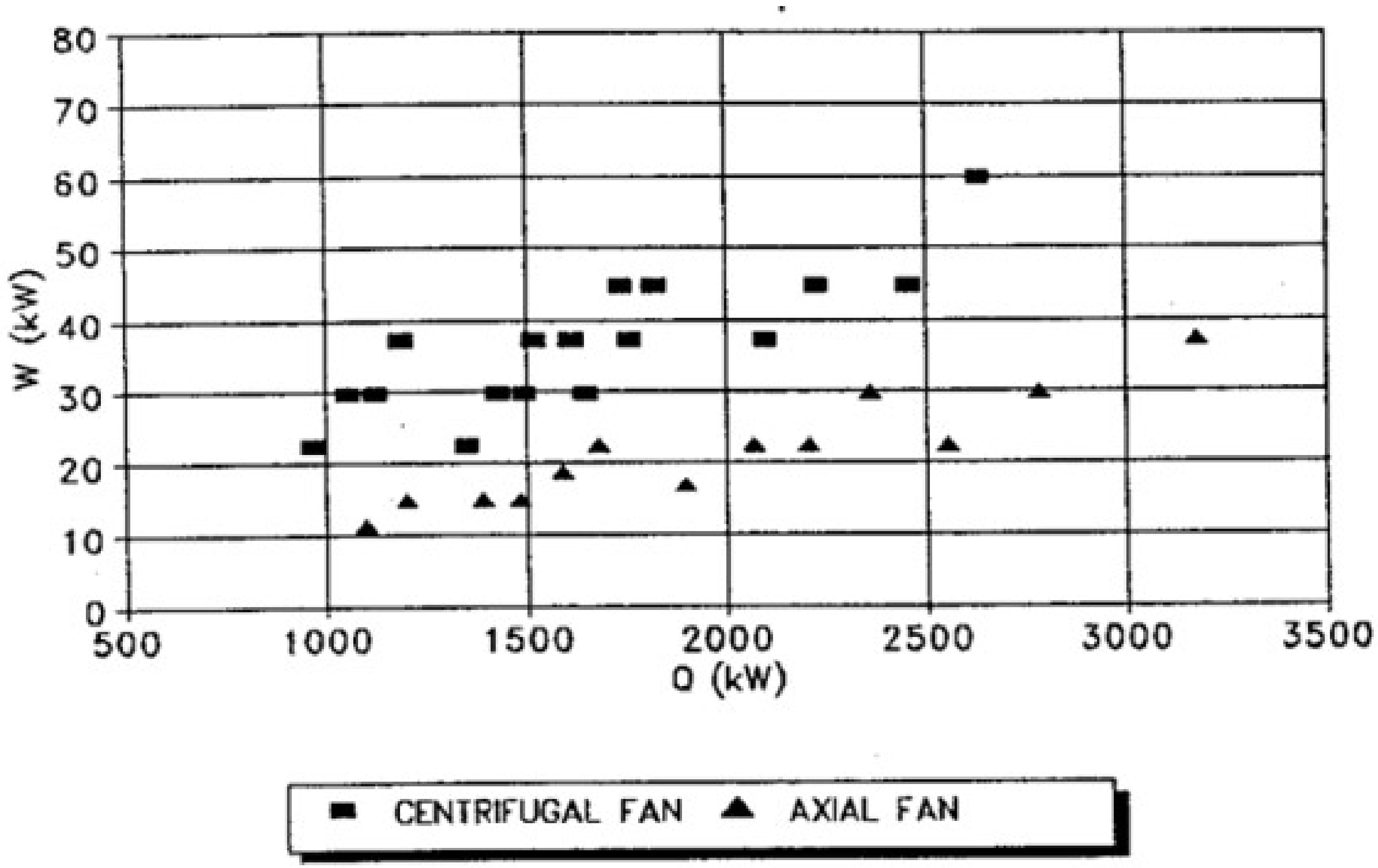
Variation of the air/water mass flow rate ratio suggested by the ASHRAE toolkit with **axial fan**

$$Mflowratio_{CTjk,n} = \frac{\dot{M}_{a,CTjk,n}}{\dot{M}_{lw,CTjk,n}}$$



Fan power as function of thermal power in nominal conditions



Air side pressure drop as function of *nominal cooling power*, according to the ASHRAE toolkit with **axial** fan

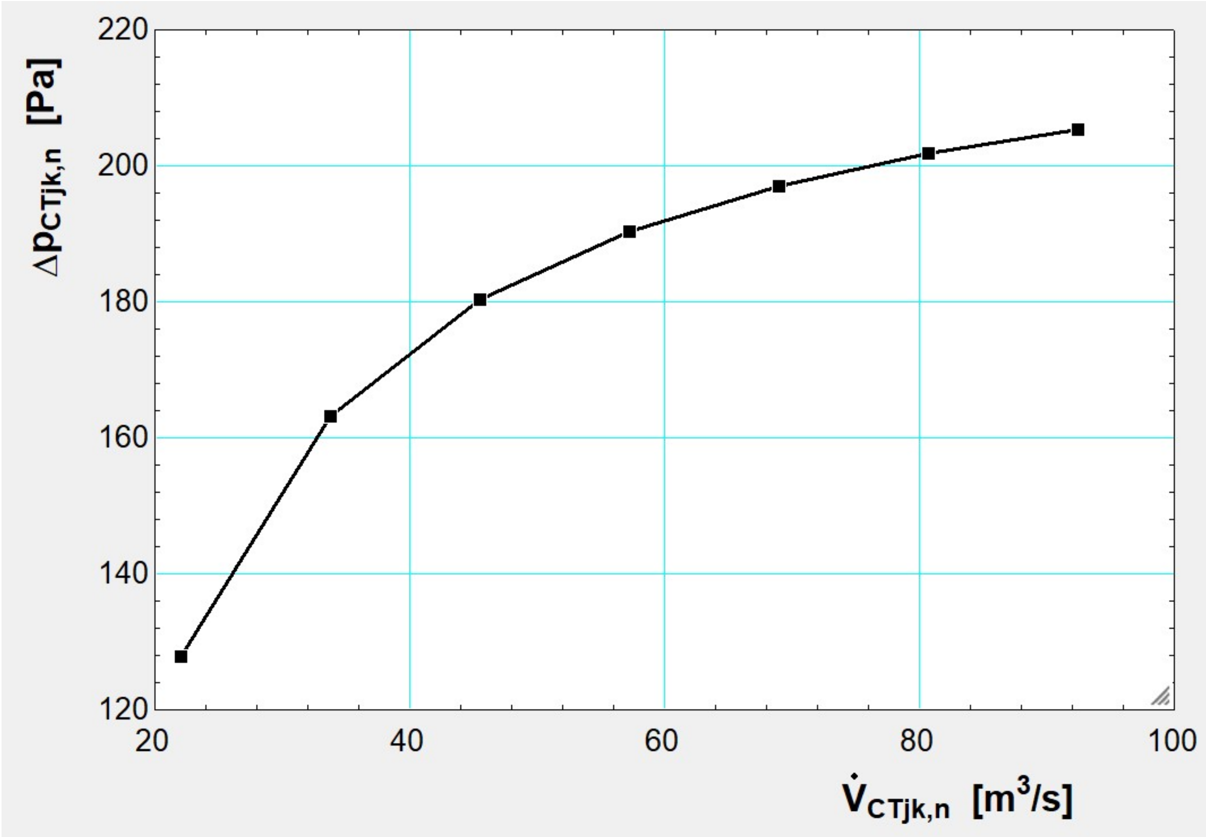
$$\dot{W}_{CTjk,n} = \dot{V}_{CTjk,n} \cdot \frac{\Delta p_{CTjk,n}}{\eta_{fan,CTjk,n}}$$

with

$$\dot{V}_{CTjk,n} = \dot{M}_{a,CTjk,n} \cdot v_{CTjk,su,n}$$

$$\eta_{fan,CTjk,n} = 0.5 \text{ [-]}$$

(hypothetical!)



If this pressure drop is supposed to occur *through the padding* of the cooling tower only, it can be defined *as if* produced through a *fictitious tube*:

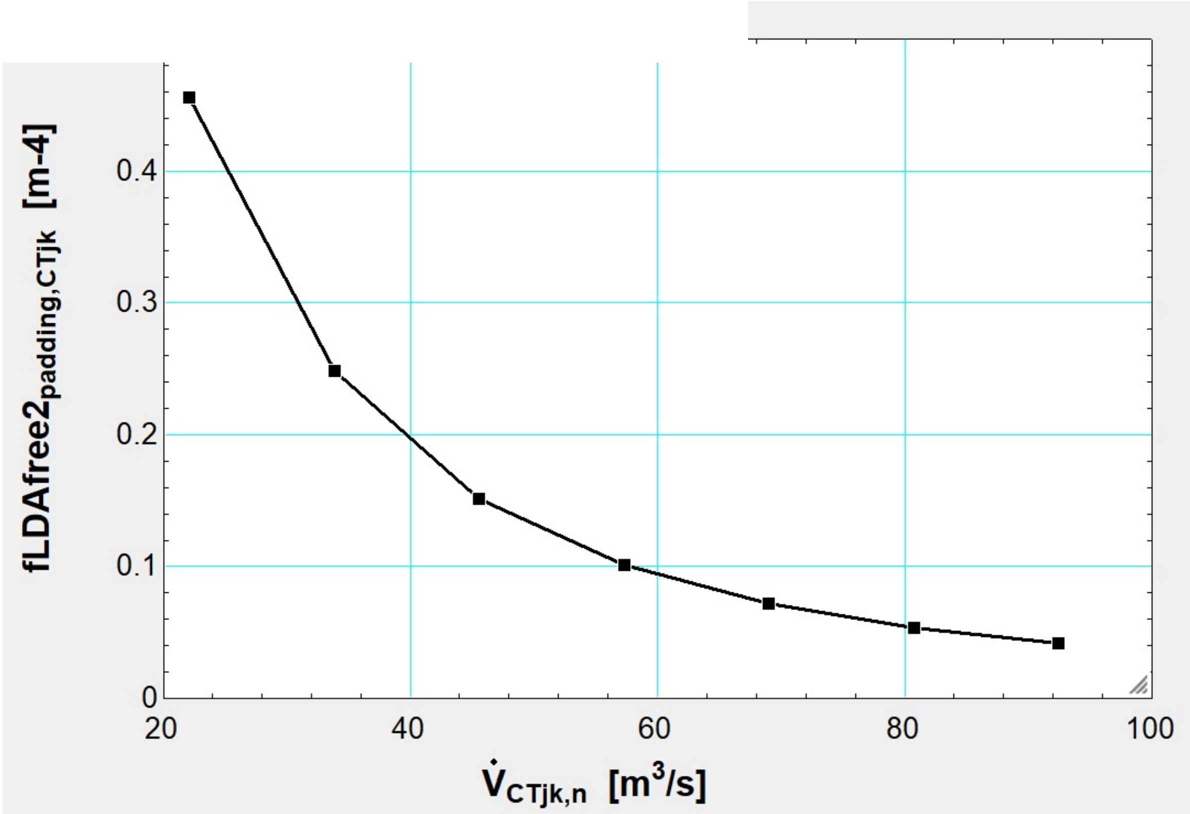
$$\Delta p = f \cdot \frac{L}{D_h} \cdot \rho \cdot \frac{vel^2}{2}$$

In the ASHRAE primary toolkit, no information is given about any typical geometry of the cooling tower.

Therefore and at this stage, one can only identify a global "**friction coefficient**":

$$fL\backslash DAfree2 = f \cdot \frac{L}{D_h \cdot Afree^2} \quad [1/m^4]$$

$$\Delta p = fL\backslash DAfree2 \cdot \rho \cdot \frac{\dot{V}^2}{2}$$



Nominal heat transfer coefficient

Fictitious heat transfer coefficient:

$$AU_{f,CTjk,n} = \frac{\dot{Q}_{CTjk,n}}{\Delta t_{ln,CTjk,n}}$$

$$\Delta t_{ln,CTjk,n} = \frac{\Delta t0_{CTjk,n} - \Delta tL_{CTjk,n}}{\ln\left[\frac{\Delta t0_{CTjk,n}}{\Delta tL_{CTjk,n}}\right]}$$

with

$$\Delta t0_{CTjk,n} = t_{lw,CTjk,su,n} - T_{wb,CTjk,ex,n}$$

$$\Delta tL_{CTjk,n} = t_{lw,CTjk,ex,n} - t_{wb,CTjk,su,n}$$

Air fictitious specific heat:

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

Actual (sensible) heat transfer coefficient:

$$AU_{CTjk,n} = AU_{f,CTjk,n} \cdot \frac{c_{p,CTjk,n}}{c_{p,f,CTjk,n}}$$

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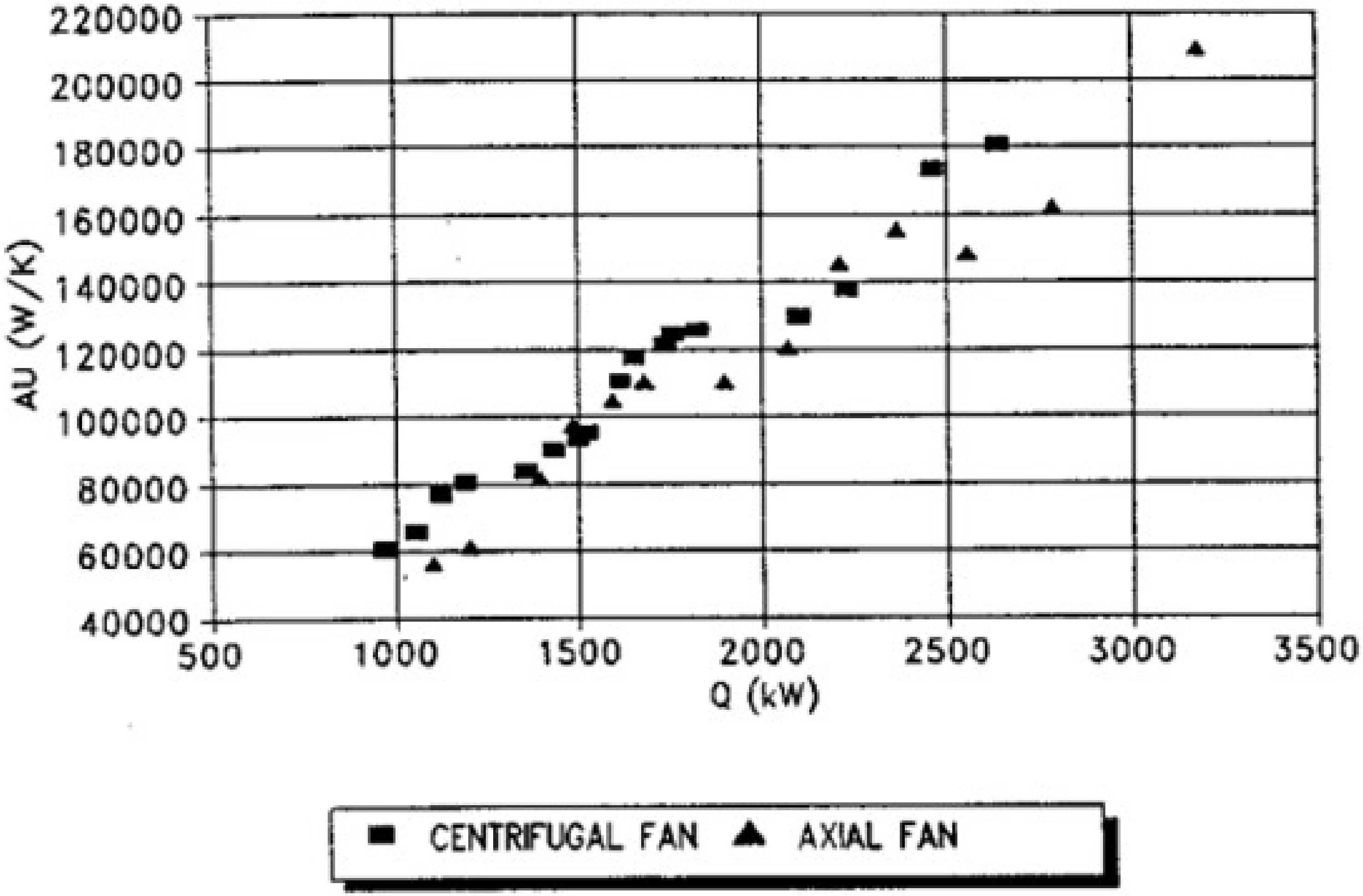
Heat transfer coefficient as function of thermal power in nominal conditions



For a given cooling tower, the actual global heat transfer coefficient can be described as follows:

AU=A_wet *U [W/K]
A_wet=epsilon_spreading*A_dry
A_dry=alpha*Volume
alpha: “padding compactness” [1/m]
U=h_c [W/m^2K]

h_c=Nusselt*k/D_h
Nusselt=j*Reynolds*Prandtl^(1/3) [-]
vel: “average velocity” **inside** the channels [m/s]
nu: “cinematic viscosity” (currently around 0.15E-4 [m^2/s])
This gives:

$$AU_{CTjk,n} = jPrkAwet\backslash nuAfree_{CTjk,n} \cdot \dot{V}_{CTjk,n}$$

$$jPrkAwet\backslash nuAfree = j \cdot Prandtl \cdot \frac{k}{\nu} \cdot \alpha \cdot Volume \cdot \frac{\varepsilon_{spray}}{A_{free}}$$

$$\text{Prandtl} \cdot \frac{k}{\nu}$$

Combination of *humid air properties* (*almost constant* in most current conditions, but with a slight effect of atmospheric pressure)

$$\alpha \cdot \text{Volume}$$

Constant combination of geometrical characteristics of **this** cooling tower

$$\frac{\varepsilon_{spray}}{A_{free}}$$

Combination of other geometrical characteristics, *increasing function of the water flow rate*…

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Very limited information available to identify most of these variables.

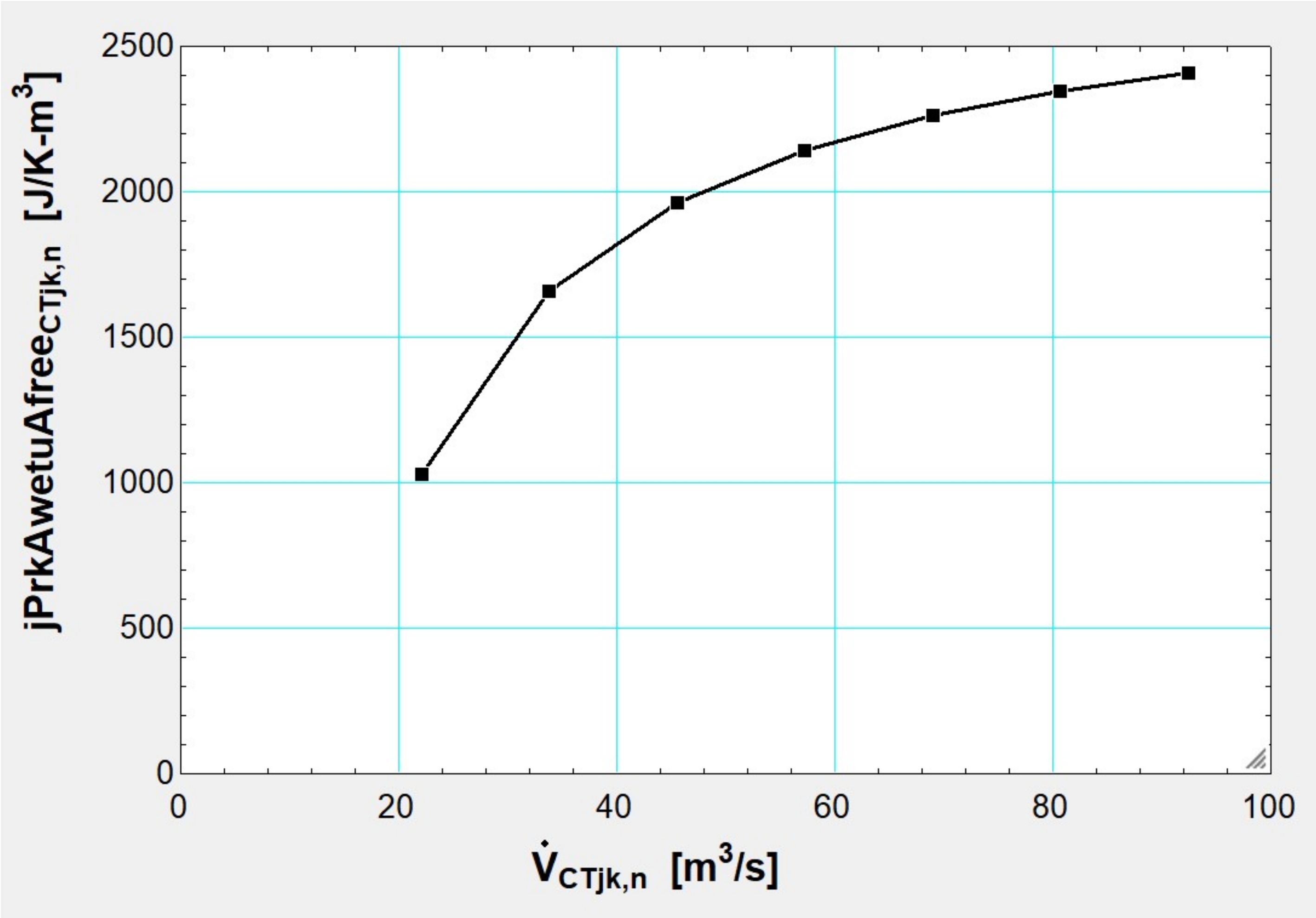
One may only expect the following tendencies:

1) AU proportional to the product "Alpha*volume*j*V_dot"

With Colburn number "j" (slowly) decreasing function of V_dot and, therefore,

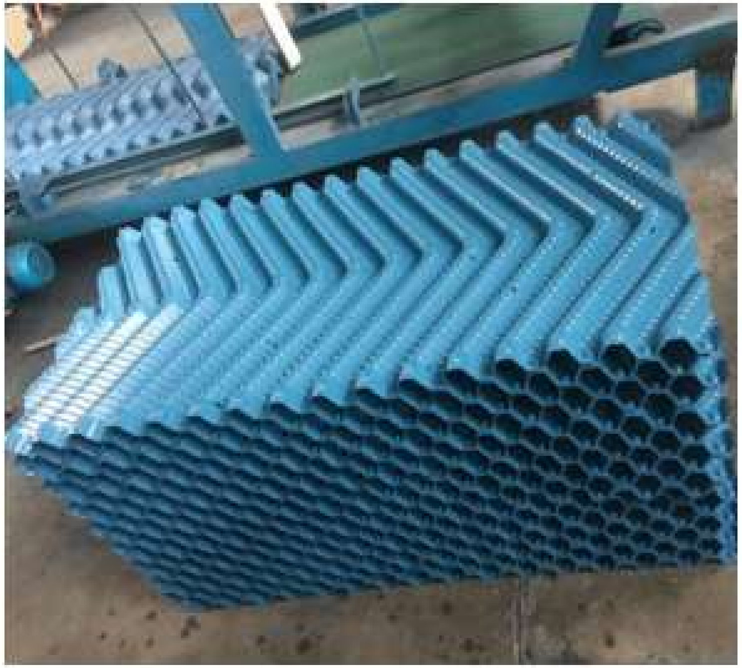
1) AU proportional to "V_dot^n" with the exponant "n" (slightly) lower than 1.
2) AU (slowly) increasing function of the water flow rate (thanks to the growing spraying effectiveness)…

Nominal Colburn coefficient as function of the nominal air flow rate with axial fan



Information produced by a manufacturer

Padding 1 (type S)



Padding 2 (oblique refraction)

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Pressure drop of Paddings:

$$\Delta P = 9.81 * a * va^m$$

With va= "approach air velocity" [m/s]

$$m = 0.0023 * q^2 - 0.0522 * q + 2.0273$$

$$a = -0.0017 * q^2 + 0.0652 * q + 0.6124$$

q= "approach (and fictitious) water velocity" [m/h]

should correspond to the obstruction produced by the liquid water falling through the padding:

$$\frac{\left[\frac{A}{A_{free}}\right]^2}{D_h}$$

Mass transfer coefficient

$Ka = C * g^{a1} q^{b1}$

With g= approach air "mass velocity" [kg/sm2]

C=3713, a1=0.59,b1=0.39; (height=1.5m)

The constant "C" appears as a (slow) decreasing function of the padding height.

The exponents of both velocities are not affected by the padding height.

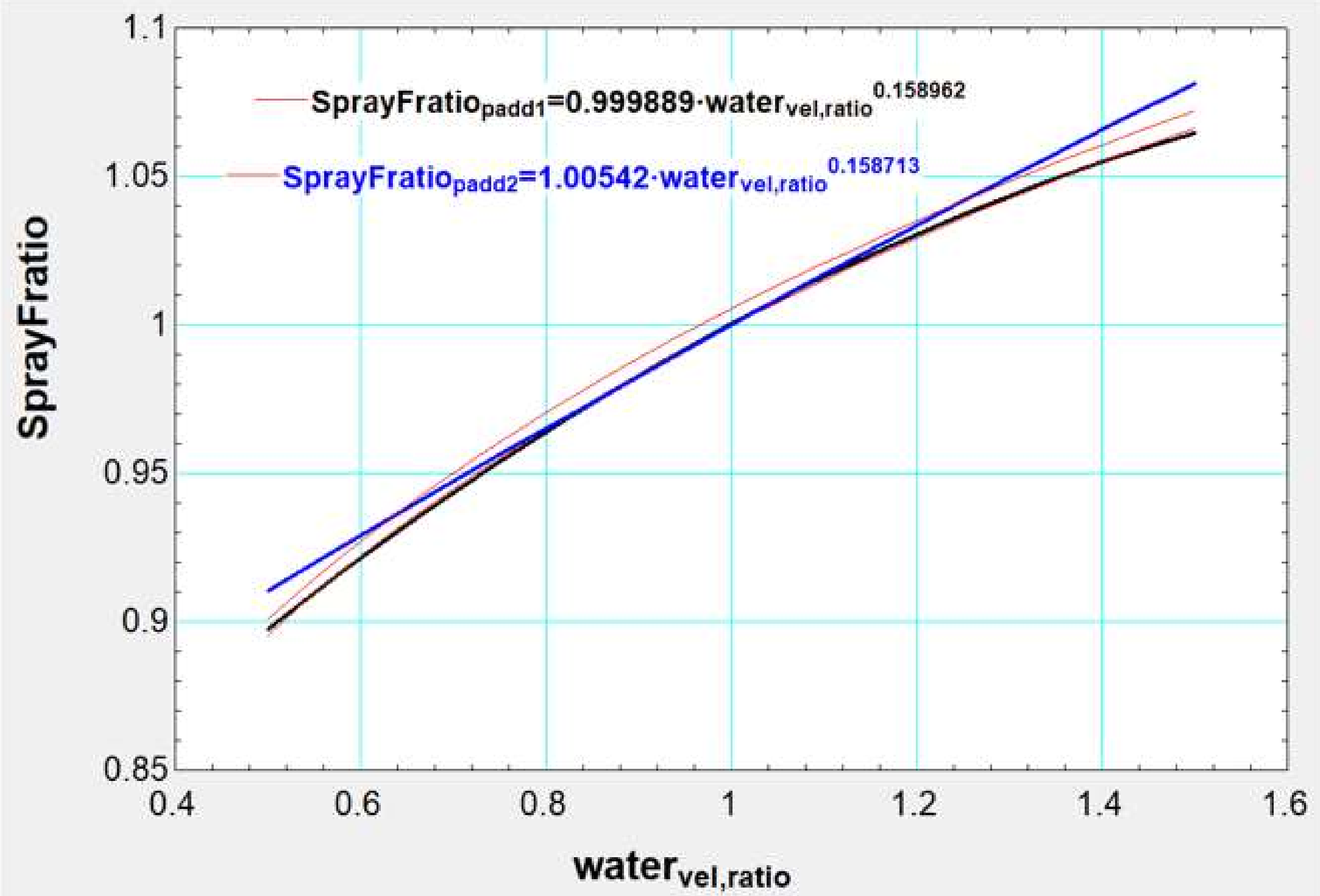
The positive effect of liquid water flow rate probably corresponds to an increase of the factor

$$\frac{A}{A_{free}} \cdot \varepsilon_{spray}$$

Contained in the Colburn coefficient.

The variation of friction coefficient due to the water spray can be expressed in relative value:

$$SprayFratio_{padd1} = \frac{fL\backslash HA\backslash A_{free2\backslash D,padd1}}{fL\backslash HA\backslash A_{free2\backslash D,padd1,ref}}$$



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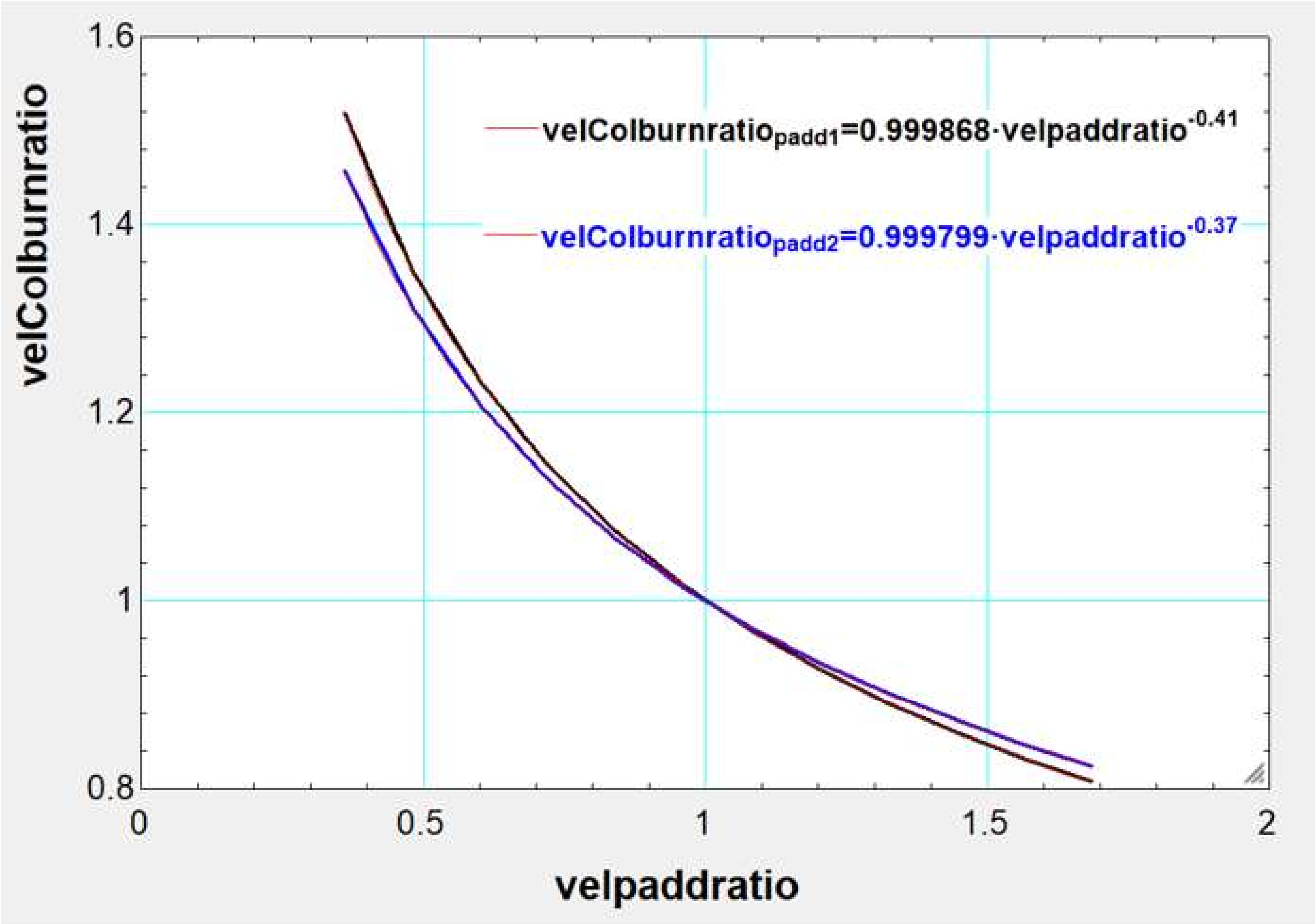
Relative effect of **air** velocity on the Colburn Coefficient



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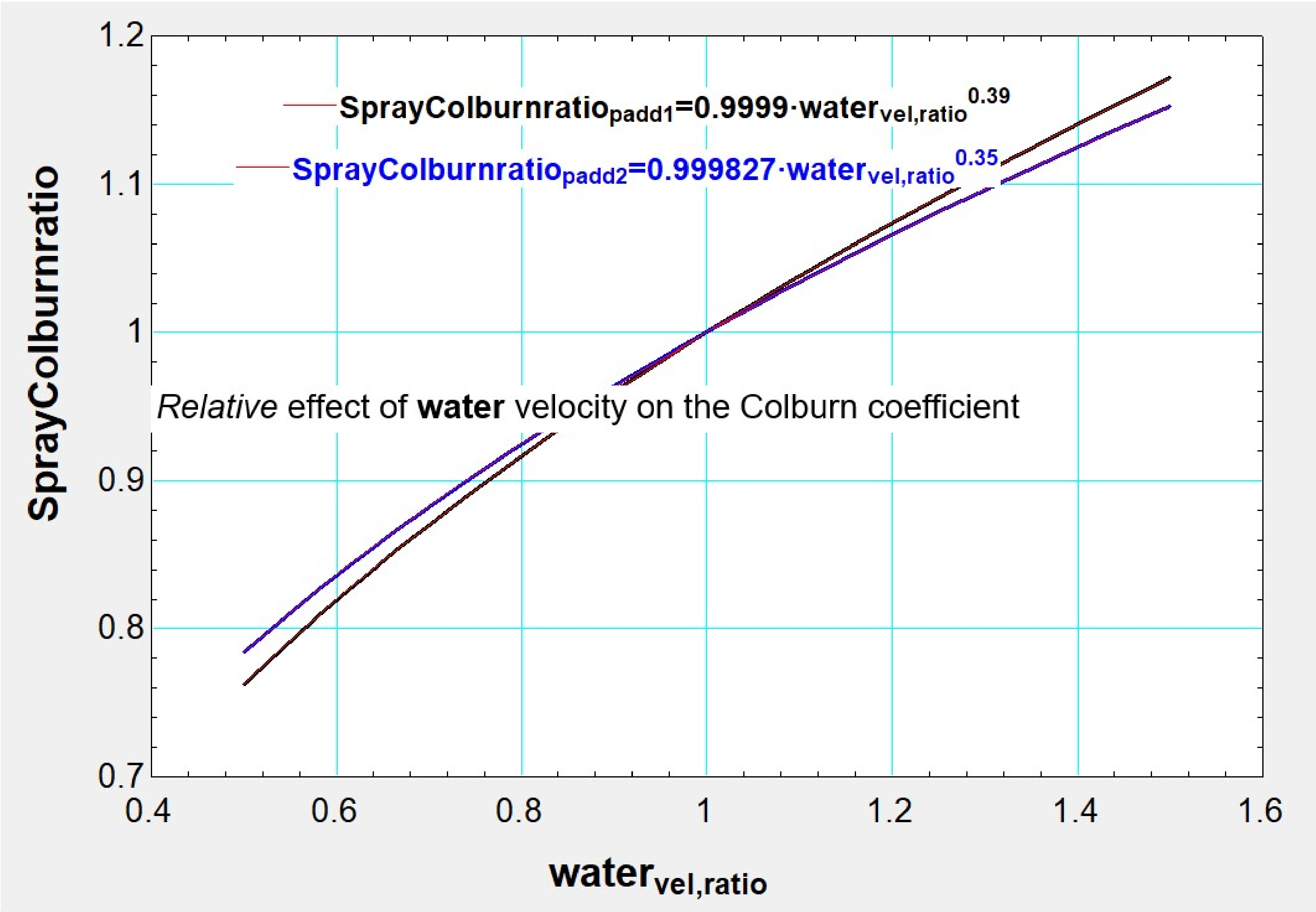
Relative effect of **water** velocity on the Colburn coefficient



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The effects both (air and water) flow rates are not independent:

increasing the air flow rate with *constant* water flow rate produces a decrease of spray effectiveness.

This seems to be suggested by the "complementarities" of the exponents of both power regressions

The water velocity effect is better observed with the flow mass ratio as independent variable.

At constant flow mass ratio, the Colburn coefficient would appear as almost constant, as to be expected in fully turbulent regime...

Conclusions

No conclusion yet…

This work should be continued and more experimental results would be very welcome!