Solar Thermal Panels for Small- Medium Scale Air Cleaners in Major Cities

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Air quality in major cities across the globe is taking a hit to alarming levels due to large scale industrialization without respecting emission norms and hence, the topic of air purification systems in cities is becoming a fertile field for research. Here, consideration is given to the use of small-medium scale air purification system for cities using a kind of solar thermal panels by inducing convective currents intended to be used in parks, housing estates or similar urban places providing a local improvement of the quality of the air. The main difficulty which arose when attempting to use convective currents is that the upward flow of hot air -which has been cleaned from contaminant particles during its upward travel, must be returned back to the ground. Therefore, in order to accomplish this task, the air not only must be heated in the ground but also must be cooled in its vertical travel in order to obtain an effective buoyancy and then a closed-convective cell. In previous works, the solution has been the use of dedicated cooling system as is the use of water spraying systems which could be attractive for large towers. However, for application at a small-medium scale such dedicated cooling systems are out of question either because the requirement of water flow or the high local humidity generate in the place uncomfortable for humans. One solution could be taking advantage of vertical panels in which a side of the panel is permanently irradiated and the other is permanently in the shadow, in this way, heating and cooling could be performed eliminating the need for specialized cooling systems, and albeit that effective buoyancy is reduced, nevertheless, the induced air flow can be worthy for local application and small-scale as is intended. Utilizing a simplified physical model, the effective buoyancy was calculated and the attainable air mass flow calculated. It is shown that for a small panel of 5 m-height, an air flow per unit of width around 0.4 kg/s is attainable; and for a 10 m-height panel up to 0.6 kg/s per unit width. Computational fluid dynamics simulations were performed and agree with the analytical results within a + 30 per cent.

Keywords. *Air pollution control technologies in cities, Filtration, Convection*

I. INTRODUCTION

Air quality in major cities across the globe is taking a hit to alarming levels due to large scale industrialization without respecting emission norms. Large scale purification of air in cities is becoming imperative and an active emerging research field is nowadays developed, as for example, water spray geoengineering [1], photocalisys [2], and building materials [3]-[5], just to name a few.

As regard the use of solar thermal energy to mitigate urban air pollution, large scale solar towers have been proposed recently. For example, Cao et.al [6], analyzed a large-scale solar collector with radius 2500 m and a chimney with the height of 500 m. More recently, Gong et al. [7] proposed a novel large-scale solar chimney with an inverted U-type cooling tower and a water spraying system at the top. The chimney considered was of 200 m-height cooling tower and a radius of 5 m. It was demonstrated that such a chimney could processes atmospheric air at a volume flow rate of 810 m^3/s.However, as was assessed for these authors, the amount of water required is an issue which must be considered from an economic point of view, and for a small-medium scale in places like parks or housing estates -as is depicted in Fig. 1, the possible use of water spraying system is out of question.

The idea we have in mind is a first assessment for closed-convective cell but in a small-medium scale which can be applied in parks, housing estates or similar urban places providing a local improvement of the quality of the air. Fig. 1 is pictorially sketching what we have in mind. The idea of such small-scale cleaners, creating a kind of local "bubbles" of clean air is relatively new. Recently, the Dutch artist Daan Roosegaarde and his team have proposed to erect a seven-meter-tall air purifier in cities, which they called as the *"Smog Free Tower"* and with the goal to treat 8.3 m^3 of air per second which authors claim able to collect more than 75% of the two kinds of pollutants, PM2.5 and PM10, that contribute to smog, [8], the *"Smog Free Tower"* is driven by a external electricity source supplying around 1.4 kW of power. Following a first launch in the chinese capital of Peking, the *Smog Free Tower* arrived to the town of Tianjin and more recently at the European city of Rotterdam. The

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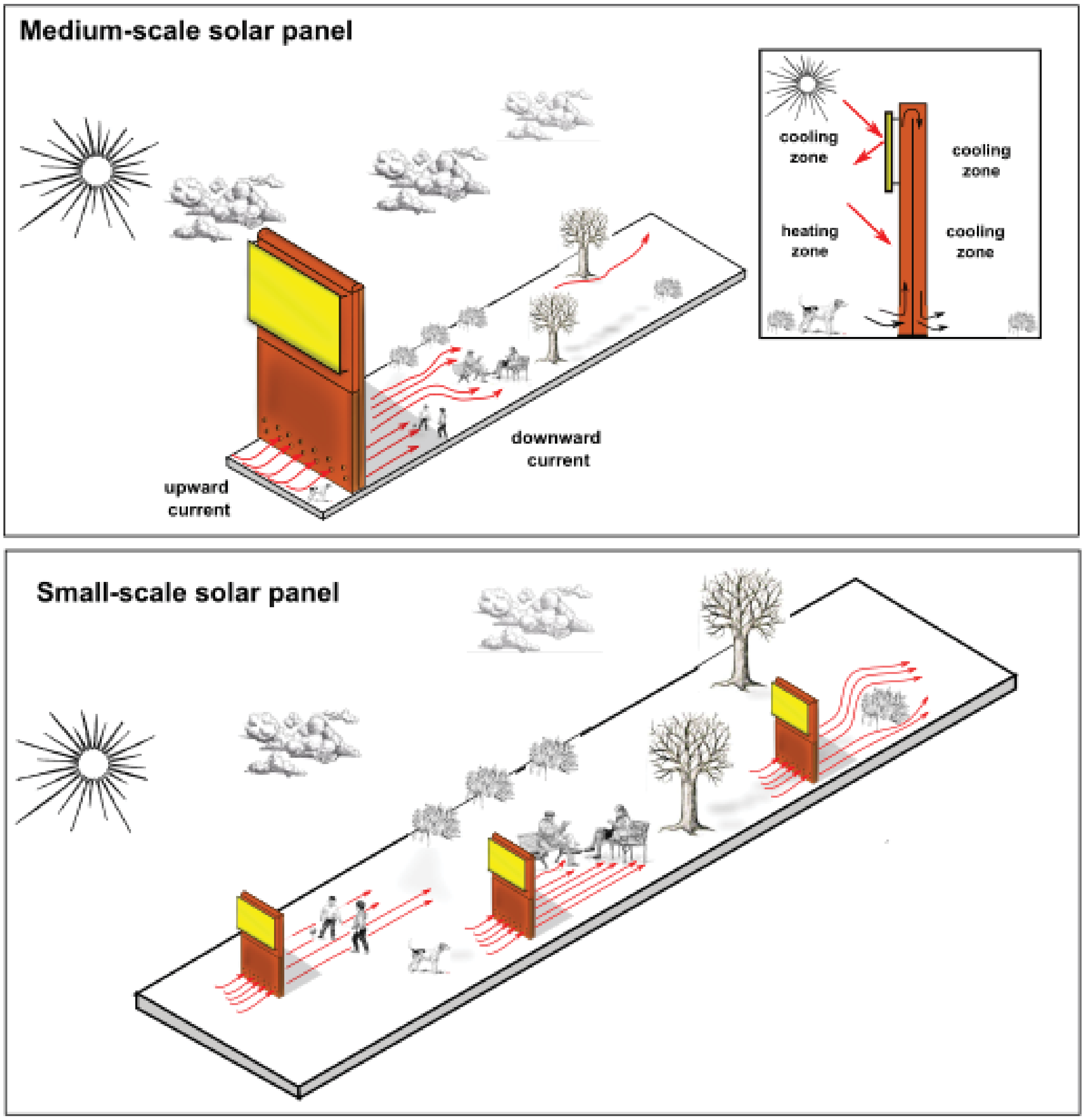

FIG. 1: A pictorial scheme of the idea discussed. At the top a 10 meters panel for local medium-scale air cleaning, at the bottom, a 2 m panel for local small-scale

"Smog Free Tower" uses a fan driven by wind energy to suck the contaminated air. The idea of such devices is worthy to be considered and the possible role of solar energy and particulary thermal solar energy.

As regard to wind energy it features some problems. For example, in order to get the claimed 8.3 m^3 of air per second, it will be necessary wind speeds around 5m/s to 6 m/s if a tower with diameter of 1 meter if a 3 meters tall-tower is considered. This velocity is difficult to get in a city and in any case, the use of fans increase the cost of installation as well as the need of mechanical parts which will require constant maintenance.

II. METHODS

A. statement of the core idea

The most general configuration we have in mind for this work is depicted in Fig. 2. This vertical solar plate keeps resemblance with classical U-tube solar towers, see for example, [9] and more recently [7]. However, definitive differences can be found not only in shape and scale considering that those solar towers with with heights around 200-m or thereabouts were intended for massive air purification in cities with air flow rates around 800 m^3 per second and requiring vigorous natural convection systems driven by auxiliary water evaporative cooling systems requiring up to 9 kg/s water into the airflow. In our case -as depicted in Fig. 1, we are interesting in a much

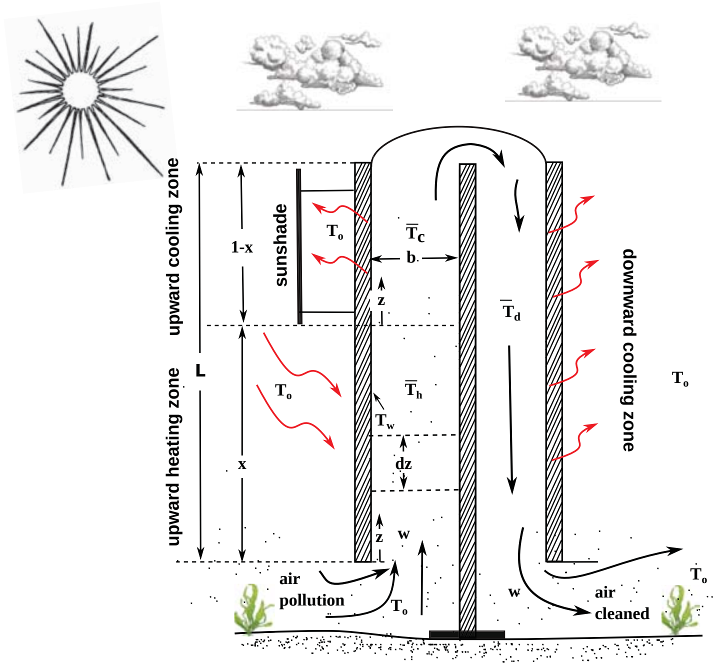

FIG. 2: Schematic of flow in the vertical solar plate.

more discreet scale, only enough to create very localized areas of clean air and spread through the city, say, in parks, etc... Therefore, the air flow would be at least 2 orders of magnitude lower and therefore evaporative cooling (driven by water spraying systems) is not required which in addition can be impractical if one considers the sharp local increase in the relative humidity which can be a problem for human comfort. For this work, there is not cooling system of the air but the panel have two sides one facing the Sun (where the air is heated) and the other in the shadow (where the air is cooled).

Returning to the Fig. 2, in this a vertical panel has two sides, namely: one side which is permanently facing the sun and the other at the shadow. For the sake of generality, a fraction $(1-x)$ of the irradiated side is deliberately also used for cooling by shadowing this, for example using a sunshade. This fraction (1-x) can or cannot be needed, but it is not known in advance, and then the most general analysis is considering this fraction in the general model.
The main problem with this scheme -to difference with traditional applications, is that air must be taken from the ground, transported inside a vertical structure and after the removal of contaminant particles, the cleaned air must be returned to the ground, and all of this driven by buoyancy and withouth need of any evaporative cooling at the top or similar. It is seen, that in order that such a convective closed cell works it would be necessary that the air during its vertical travel be firstly heated in a region and then cooled in other dedicated region.

With this simple scheme we can estimate the effective buoyancy of the full vertical pattern. To do this, first of all, we need an estimation of the respective densities, which can be calculated with the average temperature of the respective region, [10], thus

$$\bar{\rho}_h = \rho_o(1-\beta(\bar{T}_h - T_o)) \quad upward \quad heating$$

$$\bar{\rho}_c = \rho_o(1-\beta(\bar{T}_c - T_o)) \quad upward \quad cooling$$

$$\bar{\rho}_d = \rho_o(1-\beta(\bar{T}_d - T_o)) \quad downward \quad cooling \qquad (1)$$

where $\bar{\rho}_h$, $\bar{\rho}_c$ and $\bar{\rho}_d$ are the average density of hot, cool and downward region, respectively., and $\bar{T}_h$, $\bar{T}_c$, $\bar{T}_d$ are the average temperature of the hot, cool and downward region, respectively., β is the volumetric coefficient of thermal expansion of the air, ρ_o and T_o are the reference environment density and temperature, respectively. Because the hot and cool region can be represented as a column and the downward region as another, the total gradient of pressure or buoyancy is the difference of densities between both columns

$$\Delta P\Big|_{buoy} = gL\left(\bar{\rho}_h x + \bar{\rho}_c(1-x) - \bar{\rho}_d\right) \qquad (2)$$

where g is gravity, and L is the height of the panel (see Fig. 2). After some arrangements, Eq.(2) simplifies as

$$\Delta P\Big|_{buoy} = \rho_o \beta g L \left(\bar{T}_d - x(\bar{T}_c - \bar{T}_h)\right) \tag{3}$$

Therefore, the next step is to estimate the average temperature of each region. This can be inferred by considering a balance of energy and mass as follows:

- **Region I**. the upward heating region.

The heating region may be considered as an isothermal region in radiative equilibrium with a temperature T_w and then by equating the heat transferred from this isothermal wall with that absorbed in the flow, we have

$$w^* c_p dT = h_h (T_w - T_h) dz \tag{4}$$

where w^* is the air mass flow per unit width, c_p is the specific heat at constant pressure, h_h the heat transfer coefficient in this region.

By continuity, the mass flow w^* is constant, and assuming temperature-independent properties, by integrating Eq.(4) yields the local air temperature or bulk temperature at any vertical position in the heating zone as

$$T_h = T_w - (T_w - T_o) e^{-\Gamma_h z} \tag{5}$$

where

$$\Gamma_h = \frac{h_h}{w^* c_p} \tag{6}$$

The average air temperature $\bar{T}_f$ can be found by integrating Eq.(5) from $x = 0$ to the height of the heating zone Lx, and dividing by this length,

$$\bar{T}_h = \frac{1}{Lx} \int_0^{Lx} T_h dz$$

$$\bar{T}_h = T_o + (T_w - T_o) \left[1 - \frac{1 - e^{-Lx\Gamma_h}}{Lx\Gamma_h}\right] \tag{7}$$

The heat transfer coefficient h_h in this region can be estimated by considering the Nusselt number as

$$h_h = \frac{\kappa \mathbf{Nu}}{D_h} \tag{8}$$

where κ is the thermal conductivity of air, and D_h is the hydraulic diameter which for our case was calculated as $D_h = 2b$. Convection with uniform constant surface temperature for circular tubes (or plates working with its hydraulic diameter), give $\mathbf{Nu} = 3.66$, [11].

- **Region II**. the upward cooling region.

Once the air is heated in the heating region exit with a temperature $T_h(Lx)$ in the cooling region. In this region, the temperature of the wall may be assumed as first approximation as the environmental temperature T_o and then by considering continuity of the mass flow, a similar equation than Eq.(4) can be used. If a new origin of coordinated is defined for this region with origin $z = 0$ at the point where the cooling region starts, we have

$$w^* c_p dT = -h_a (T_h(Lx) - T_o) dz \tag{9}$$

where h_a is the convective heat transfer coefficient of air, and $T_h(Lx)$ is the exit temperature from the heating region. many semiempirical formulations for the convective heat transfer coefficient of air are available; but the following simplest expression, [12], seems preferable

$$h_a = 10.45 - u_w + 10 u_w^{\frac{1}{2}} \tag{10}$$

where u_w is the relative speed of the air (m/s), and h_a is in W/(m^{2}K). It is seen that for mild winds, as expected in cities, around 2-4 m/s, a conservative heat transfer coefficient of air around $h_a \approx 25$W/(m^{2}K).
Now, proceeding as before we obtain for the temperature at the cooling upward region

$$T_c = T_o - (T_h(Lx) - T_o) e^{-\Gamma_c z} \tag{11}$$

where

$$\Gamma_c = \frac{h_a}{w^* c_p} \tag{12}$$

and the average air temperature $\bar{T}_c$

$$\bar{T}_c = T_o + \frac{T_h(Lx) - T_o}{L(1-x)\Gamma_c} \left[1 - e^{-L(1-x)\Gamma_c}\right] \tag{13}$$

- **Region III**. the downward cooling region.

In the downward region is calculated exactly as the region II but with an inlet temperature $Tc(L(1-x))$ rather than $T_h(L(x))$, with the same heat transfer coefficient and integrated from $z = 0$ to $z = L$ and then we obtain

$$\bar{T}_d = T_o + \frac{T_c(L(1-x)) - T_o}{L\Gamma_c} \left[1 - e^{-L\Gamma_c}\right] \tag{14}$$

Finally, the ar mass flow in the system is calculated by momentum considerations

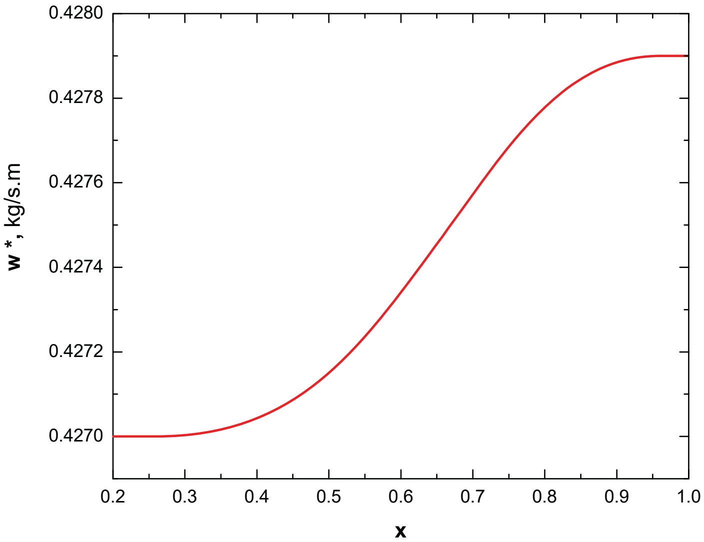

FIG. 3: The attainable air flow rate per unit width of the panel for a 5-meters height panel as function of the fraction x.

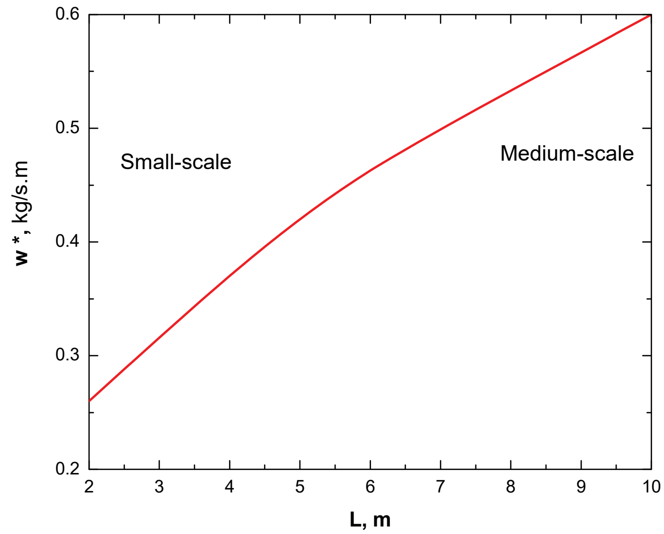

FIG. 4: The attainable air flow rate per unit width and as function of the height L and using $x = 1$.

B. Momentum considerations.

In turbulent, fully developed, two-dimensional flow between parallel plates with cross area sb, the pressure drop can be calculated by the Darcy-Weisbach equation,

$$\Delta P\Big|_{loss} \approx \frac{f L_t w^{*2}}{2 D_h b^2 \rho_o} \tag{15}$$

where L_t is the total length of the tower, and D_h is the hydraulic diameter. Considering that $b \ll z$ the hydraulic diameter is $D_h \approx 2b$ and also that the total length is $L_t = 2L$, then Eq.(15) becomes

$$\Delta P\Big|_{loss} \approx \frac{f L w^{*2}}{2 b^3 \rho_o} \tag{16}$$

Equating Eq.(16) and Eq.(3), the the air mass flow rate per unit width, in the channel, is found equal

$$w^* = \left[\frac{2b^3 \rho_o^2 \beta g}{f}\left(\bar{T}_d - x(\bar{T}_c - \bar{T}_h)\right)\right]^{\frac{1}{2}} \tag{17}$$

• **Discussion**

In order to obtain some idea of the air mass flow per unit width attainable predicted by Eq.(17), we assume some typical values and estimation of the parameters:

an environmental temperature $T_o = 293$ K; hot wall temperature $T_w = 380$ K (see Appendix); $c_p = 1.0 \times 10^3$ J/(kg K); $\kappa = 0.025$ W/(mK); $\beta = 3 \times 10^{-3}$/K; $\bar{\rho} = 1.15$ kg/m^3; $g = 9.8$ m/s^2 and a friction factor $f = 0.002$; $\mathbf{Nu} = 3.66$, [11]; and $h_a = 25$ W/m^{2}K corresponding to an air speed around 2-3 m/s (see Eq.(10). The resulting curves are shown in Fig. 3 and Fig. 4 Referring to Fig. 4. it was analyzed the need of a dedicated cooling zone in the upward region by evaluating the fraction x using a 5-meters height panel, where $x = 1$ represents that there is not such dedicated zone (see Fig. 2). It is seen, that this zone is not necessary and the changes are absolutely negligible. In Fig. 4 is plotted the curve for the attainable air flow rate per unit width of the panel as function of the height L of the panel. It is seen that, for a small-scale 5-m-height panel an air flow per unit width around 0.3 kg/sm, is attainable, and for a large panel with up to 10 m-height it is attainable up to a 0.6 kg/sm. Therefore, with a practical widths around 5 to 10 meters, air flows around 1.5 kg/s to 3 kg/s and 3kg/s to 6 kg/s.

III. COMPUTATIONAL SIMULATION

In this section a simplified 2-dimensional pressure-based CFD model was developed using the commercially available CFD software Fluent. The solution was calculated using the pressure-based solver and the Boussinesq assumption was used to model buoyancy, [13]. Fig. 5 and Fig 6 show for illustration the temperature and velocity profile for a 1-m-height column, respectively., and Fig. 7. is the comparison between the results using the analytical model (depicted previously in Fig. 4) and the CFD simulation. It is seen that the theoretical model overestimates the air flow rate around 30% in compari-

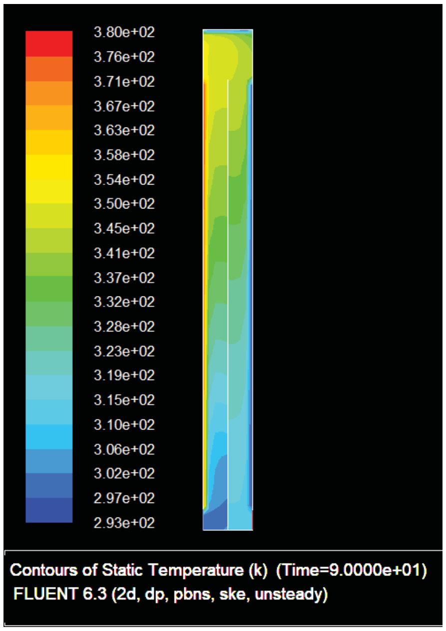

FIG. 5: Temperature profile from the CFD simulation for a 1-m-height column.

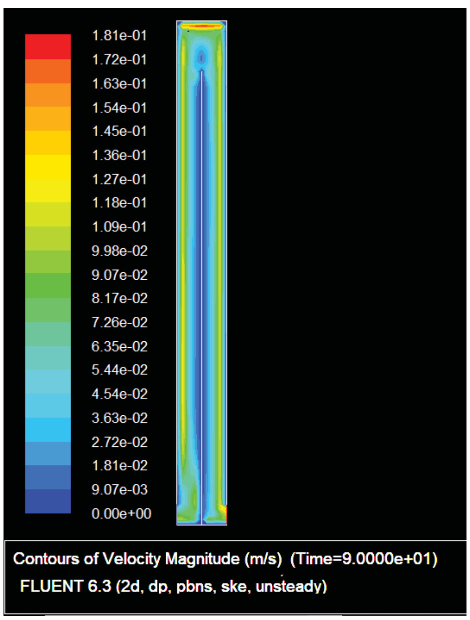

FIG. 6: Velocity profile from the CFD simulation for a 1-m-height column.

son with the CFD simulation. This result is acceptable in view of the several approximations taken in the theoretical model. Therefore, it can be concluded that the use of the analyzed solar panels for local urban air purification could provide betwenn 0.1 kg/s to 0.5 kg/s per width for 1 m to 10 meters-heights.

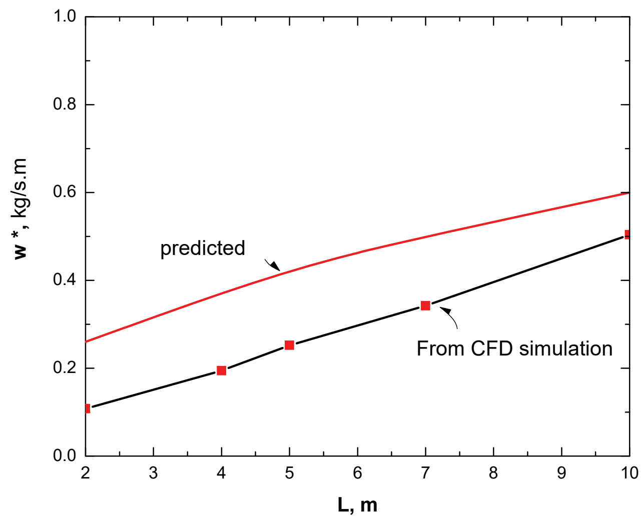

FIG. 7: Comparison between the predicted air flow rate per unit width and the CFD simulation.

IV. SUMMARY OF RESULTS AND CONCLUSIONS

Solar thermal panels for small-medium scale of air cleaners in major cities has been proposed and analytical expression for the attainable air flow per unit width attainable with such panel derived. Some interesting conclusions are raised by this preliminary work as follows:

(a) Flow rates per unit of width of the panel up to 0.5 kg/s per meter width are attainable.

(b) From **(a)**, it is quantitatively reasonable to take in consideration solar thermal panels for local small-medium scale air cleaners in major cities such as parks, housing estates or other local regions in cities which can provide a kind of "clean islands" where air be more breathable.

V. APPENDIX

The attainable wall temperature at the heating region may be, as first estimation, derived from considering an radiative equilibrium by using the Stefan-Boltzmann law as

$$q \approx \bar{\epsilon}\sigma T_w^4 \tag{18}$$

where q is the solar flux per unit of area, ϵ the effective thermal emissivity of the wall which included both absorbtion and emission, σ the Stefan-Boltzmann constant, and T_w the wall temperature. The correct estimation of

T_w implies the knowing of the effective emissivity of the material used, but it can be found as low as $\epsilon = 0.2$ for high-tech solar technology or as high as $\epsilon = 0.8$ for more coarse materials. On the other hand, solar insolation can be as high as 700 W/m^2 for a summer clear day and drop to 200 W/m^2 during a winter cloudy day.Therefore, taking somewhat medium values as $\epsilon = 0.4$ and $q = 400$ W/m^2, the attainable wall temperature could be as high as 380 K or thereabouts.

NOMENCLATURE

b = plate spacing
c_p = heat capacity at constant pressure
D_h = hydraulic diameter
f = friction factor
g = gravity
h = heat transfer coefficient
L = height of the panel
Nu = Nusselt number
P = pressure
q = solar flux per unit area

s = plate width
T_o = inlet temperature
T_w = wall temperature
u_w = speed of the air.
x = vertical coordinate
w^* = air mass flow per unit width

Greek symbols

ρ_o = density
β = volumetric coefficient expansion of air
κ = thermal conductivity of air
Γ = Thermal parameter
ϵ = emissivity
σ = Stefan-Boltzmann constant

subscripts symbols

c = cold upward region
d = cold downward region
h = hot upward region
o = inlet, initial
w = wall
p = pressure

ACKNOWLEDGEMENTS

This research was supported by the Spanish Ministry of Economy and Competitiveness under fellowship grant Ramon y Cajal: RYC-2013-13459.

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