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Computational Fluid Dynamics study of a corrugated hollow cone for enhanced dew yield

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28 exchange is similar for both smooth and corrugated surfaces, and corrugation increases cooling. At
29 higher air flow velocities, convective heat exchange is larger for the W-cone but cooling is only
30 slightly smaller than found on the S-cone. Corrugated W-cone should thus give larger yield than the
31 corresponding smooth S-cone.

32

33 **Keywords**

34 Water Harvesting; Dew condensation; CFD simulation; hollow cone; corrugated hollow cone

35

36 **Highlights**

- 37 • Air flow and surface temperature of smooth and corrugated hollow cones are compared by
38 CFD
- 39 • Corrugations enhance dew collection by slope and edge effects
- 40 • Corrugations only marginally increases heat losses

41

42 **1. Introduction and background**

43

44 In the hydrological cycle, natural dew has the same origin as fog and rain precipitations, namely
45 condensation of the water vapor contained in the atmosphere. The latter results from water evaporated
46 from sea, river, lakes, vegetation and soil. Recent isotopic studies (Kaseke et al., 2017) indeed show
47 that fog and dew have multiple origins and that groundwater in drylands can be recycled via
48 evapotranspiration and redistributed to the upper soil profile as non-rainfall water.

49 In contrast to precipitations, which condense in the cold regions of the atmosphere, dew, alike
50 radiative fog, forms under radiation cooling of a substrate. Dew water is thus a surface phenomenon,
51 in contrast to precipitations which is a volume process. Natural dew is a common and sometimes
52 significant water source in many ecosystems, especially those in arid and semi-arid areas of the world
53 (see e.g. Monteith, 1963; Baier, 1966; Malek et al., 1999; Agam and Berliner, 2006; Liu et al., 2020).

54 Dew as a source of water for humans has been long ignored, although it is generally a water of good
55 quality (Kaseke and Wang, 2018). Since the last 20 years, many studies aimed to increase the dew

56 yield up to its theoretical limit on order of $1\text{L}/\text{m}^2$ (see e.g. the reviews by Tomaszekiewicz et al. (2015)
57 and Beysens (2018)). The dew yield essentially depends on the balance between the surface and
58 atmosphere emissions. Dew yield is limited by the available cooling energy, on order of $60\text{ W}\cdot\text{m}^{-2}$ in
59 the humid air conditions where dew can form (Bliss, 1961; Beysens, 2018), the latent heat of
60 condensation and the convective heat flux between the substrate and the surrounding air. The latter
61 increases with wind speed but can be greatly reduced in hollow structures like hollow cones.

62 Increasing the dew yield needs to augment the cooling energy but also the efficiency of water drops
63 collection. The latter involves the efficiency of droplets shedding by gravity. The effect of gravity
64 increases with the angle of the structure with horizontal; however, radiative cooling diminishes with
65 this angle. A good compromise is a tilt angle of 30° with horizontal (Beysens et al., 2013). Hollow
66 cones with 60° half-angle have thus provided good results for dew condensation and dew collection in
67 outdoor conditions (Clus et al., 2009).

68 Dew drop collection efficiency can be enhanced by the effect of convex edges where drops grow faster
69 thanks to a solid angle of vapor collection larger than in the middle of the substrate (Medici et al.,
70 2014). Edge drops reaches the critical size where the gravity force overcomes the pinning force sooner
71 and slide down, incorporating the drops on their pathway, acting thus as efficient natural wipers. In
72 addition, corrugation increases the local tilt angle with horizontal (Beysens, 2018), thus increasing
73 locally the gravity forces acting on the drops (Fig. 1). Origami structures that partially combine the
74 above positive effects on dew yield (partly hollow structures, edge effects) have been seen to increase
75 the yield by a factor as large as 400% for small yields (Beysens et al., 2013).

76

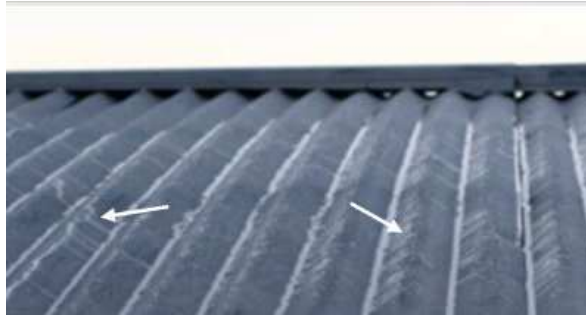


Fig. 1. Drops sliding from the top of corrugations (arrows) on a galvanized iron roof with small tilt angle ($\approx 15^\circ$).

(Combarbala, Chile; Photo J.-G. Minonzio).

Corrugation, however, increases the convective heat exchange with surrounding air and in this aspect may substantially reduce the substrate cooling. In order to estimate the effect of corrugation on a hollow cone, one reports in the following the comparison by Computational Fluid Dynamics (CFD) of a corrugated, W-shaped hollow cone with the same smooth structure, both showing condensing surfaces with nearly 30° angle with horizontal (cone half-angle of nearly 60°). Two softwares were used: Ansys CFX for detailed aerodynamics where the computational domain is modeled to obtain a fully developed wind profile and COMSOL Multiphysics aiming to detail the heat fluxes, including radiative exchange surface-to-sky and surface-to-surface. Since neither Ansys CFX nor COMSOL simulation models include dew formation, an evaluation of dew yield was performed by calculating with COMSOL the local temperatures, which can be related to the dew yield per surface area within some assumptions that will be discussed.

The Ansys CFX was utilised to analyse the proposed design of the cones and test the aerodynamic effects on the airflow velocity over the inner surfaces of the condenser. The metrics used for the aerodynamic performance assessment were based on minimising the air speed as well as maintaining uniform levels of air velocity on the condensing surface. The CFD analysis allowed the cones schema to be optimized by testing various architectural solutions, which resulted into the improvement of the design's aerodynamic performance. A total of five cases were studied from which two candidates were finally selected (Fig. 2). The thermal effects were studied by COMSOL on the two finalists, W- and S-cones.

101 2. Methods

102

103 2.1. The cones

104

105 In the pre-study by CFX simulation the cone dimensions are 7.25 m (S-cone upper radius), 5.5 m (W-
106 cone upper radius) with upper part at 10 m above the ground (Fig. 2). The reason for the upper radius
107 difference between S- and W-cones is due to the structural and architectural design of the cones. As
108 the angles of the cones are the same and the height also the same, then the mouth of the S-cone is
109 wider (the W-cone turns up at the edges to create a uniform top edge). The condensing surface area in
110 the W cone is $S_{cw} = 100.9\text{m}^2$, while the one in the S cone is $S_{cs} = 187.3\text{m}^2$. The fact that the W-cone
111 has smaller surface area is due to the fact that the W-cone is inscribed in the S-cone surface, the S cone
112 having a larger diameter than the W-cone.

113 The COMSOL simulation uses a schematic representation of both W- and S cones where the
114 supporting column has been omitted and the external shape has been schematized (see Fig. 6 below).
115 The goal is indeed to look only to the useful hydrodynamics around the cone and focusing on the air
116 flows and heat exchange inside the funnels. The dimensions of the cones are made identical to better
117 compare their thermal characteristics. The cone dimensions are (S cone) 4.735 m radius and (W cone)
118 4.635 external radius and 3.944 m internal radius, with both 0.21 m lower radius at 2.554 m below the
119 top. The S-cone half-angle is 60.5° and the W-cone is inscribed in two cones with half-angles 60.0°
120 and 55.6° . The upper part is at 8 m above the ground (Fig. 2). The condensing surface area in the W
121 cone is $S_{cw} = 66.8\text{ m}^2$, while the one in the S cone is $S_{cs} = 80.8\text{ m}^2$. As for the CFX simulation, the W-
122 cone exhibits a smaller surface area because it has a smaller external radius than the S-cone.

123 Air temperature is set at 15°C and wind speed is varied by steps from 0.5 to 5m/s (CFX) or at two
124 typical values, 10^{-4} and 2 m/s (COMSOL). The CFX simulation deals with the exact shape of the W-
125 cone.

126

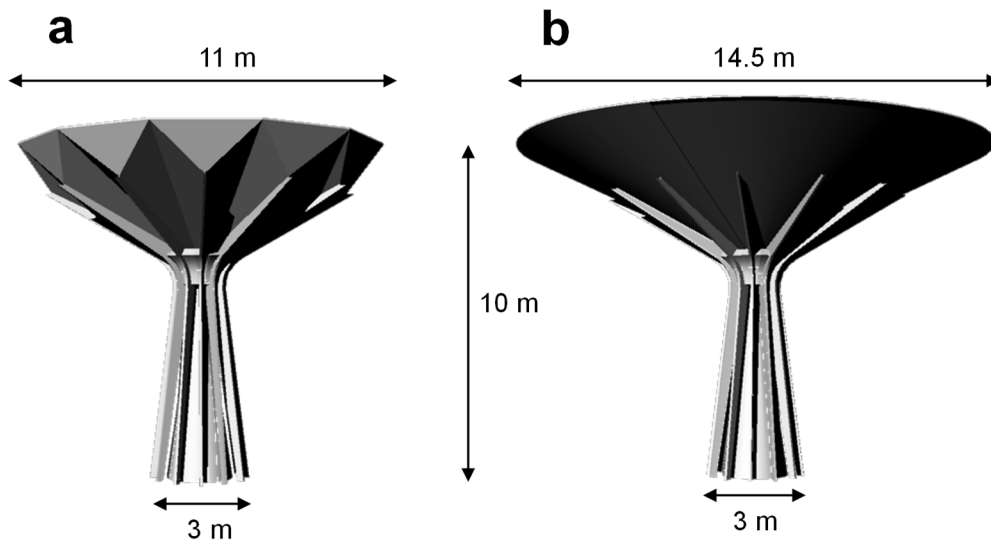


Fig. 2. Geometry of the W-cone (a) and S-cone (b) in the CFX simulation.

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128

129

130 2.2. Ansys CFX simulation

131

132 As already noted, the Ansys CFX code version 18 is utilised to analyse the proposed design of the
 133 cones and test the aerodynamic effects on the airflow velocity over the inner surfaces of the condenser.
 134 $k - \varepsilon$ turbulence model is used to simulate mean flow characteristics for turbulent flow conditions. In
 135 order to capture the aerodynamic effect of the installation, the geometry includes the lower stand
 136 (trunk of the tree) and all the details of the supporting structure that could affect the airflow in the
 137 main areas of interest.

138 The computational size of the domain was developed in line with the recommendations for the
 139 assessment of wind conditions around buildings (Mochida et al, 2006). The inlet, the sides and the
 140 upper boundary were set at 5 times the cone height H whilst the outflow boundary was set at $15H$
 141 behind the cone ensuring an appropriate development of the wakes produced by the interaction of the
 142 flow with the geometry itself. Unobstructed upstream conditions were also considered as the most
 143 likely scenario to take place on site where the condensing trees will be installed.

144 As summarised by Blocken et al. (2006), the following requirements were implemented:

145 (i) Sufficient height mesh resolution in the vertical direction in order to allow the height of the first
 146 cell to be smaller than 1m

- 147 (ii) Horizontal homogeneous atmospheric boundary layer flow in upstream and downstream regions
 148 (iii) Ensuring that the distance between the centre of the first mesh cell and the ground, z_p , is smaller
 149 than the height of the terrain physical roughness z_c .

150 Due to the symmetrical design of the condensing trees, a rectangular computational domain was
 151 created for this analysis.

152 The specification of the atmospheric boundary layer profile is crucial for a correct simulation of the
 153 airflow environment. The boundary layer is described as the lower portion of the atmosphere where
 154 the wind is largely affected by friction with the surface of the earth. This friction generates turbulence
 155 and depends on the type of terrain and geometrical obstructions on the surface. The CFD model used a
 156 semi-empirical logarithmic function (see e.g. Pal Arya, 1988) to describe the wind profile $V(z)$:

157

$$158 \quad V(z) = V_{10} \frac{\ln\left(\frac{z}{z_c}\right)}{\ln\left(\frac{10}{z_c}\right)} \quad (1)$$

159

160 where z_c is the roughness length (taken to be $z_c = 0.1$ m) and V_{10} is wind speed at 10 m off the ground.

161 The logarithmic profile is normally used as inlet condition. The region near to the ground of the
 162 domain required special attention as this area is where the interaction between flow and walls happens.

163 If this area of contact is not appropriately defined imprecisions may arise in the CFD assessment. To
 164 eliminate this risk a fine mesh was applied to the ground with an inflation layer of 3 steps, whilst the
 165 “wall function” was automatically assigned when the $k - \varepsilon$ model was selected.

166 Side walls and upper domain were modelled using symmetry boundary conditions, whilst a zero-static
 167 pressure boundary condition was applied to the outlet. The three-dimensional, steady Reynolds-
 168 averaged Navier-Stokes (RANS) equations was solved to simulate the wind flows within the domain.

169 Turbulence was modelled with the $k - \varepsilon$ model as mentioned above.

170 The computational domain was discretised with a tetrahedral mesh of 6 million cells. An inflation
 171 layer of 3 steps, where the first layer's height measured at 0.25m, ensured the adequate mesh
 172 resolution at ground level. Several mesh sensitivity tests were carried out. No relevant differences
 173 have been identified in the velocity fields of the upstream and downstream flows.

174 Because of the shape of the faceted crown, the W-cone is expected to produce local acceleration of the
175 flow and accentuated deflection of the flow streamlines. The S-cone instead has a smoother external
176 surface, therefore this shape is expected to reduce flow disruption and facilitate flow acceleration away
177 from the crown.

178

179 *2.3. COMSOL Multiphysics simulation*

180

181 The commercial software, COMSOL Multiphysics, was used to analyze the heat transfer and fluid
182 flow taking place around the cones, also using the $k - \varepsilon$ turbulence model to simulate the mean flow
183 characteristics for turbulent flow conditions. The goal of this modeling effort is to predict the
184 temperature of the condenser surfaces for different values of air velocity. In the fluid domains, the
185 continuity equations as well the Navier-Stokes and the energy conservation equations have to be
186 solved simultaneously.

187 The cones structure is made of Styrofoam to ensure thermal isolation. The simulation of the dew
188 condenser heat exchange takes in account the following processes:

189 (i) Thermal behavior of the radiative material and the insulation material, including its thermal
190 conductivity, thermal expansion, density, heat capacity and emissivity in the atmospheric window
191 whenever known (see Table 1). For simplicity, the radiative and insulation materials have been taken
192 the same (Styrofoam).

193 (ii) Radiative cooling power, which depends on the condenser geometry and also on atmospheric
194 conditions (condenser and sky emissivity, air temperature T_a , cloud cover N).

195 (iii) Incoming diffusive and convective (free or forced) heat exchange with air, which depends on
196 wind speed V , wind speed direction and condenser geometry. The condensation process is not
197 accounted for in the model ("dry air" approximation). The equations of momentum, continuity and
198 temperature are solved by using the numerical code.

199 In the simulation only typical night conditions are considered, e.g. clear sky $N=0$, $T_a = 288.15$ K
200 (15°C), relative humidity $RH = 80\%$. These conditions correspond to a dew point temperature $T_d =$

201 11.8°C. Standard numerical values are used for the air properties (density, thermal conductivity,
 202 specific heat, etc., see Table 1). Emissivities are taken in the atmospheric window. The most
 203 interesting result of the simulation is the local temperature of the condenser, from which the dew yield
 204 can be deduced (see Appendix 1).

205

	$k (Wm^{-1}K^{-1})$	$\nu (m^2.s^{-1})$	$C_p (J kg^{-1}K^{-1})$	$\rho (kg m^{-3})$	$\alpha (10^{-3} K^{-1})$	ε
Air (15°C)	0.026	$2.2 \cdot 10^{-6}$	1006	1.17	3.4	0.80 ^a
Styrofoam	0.035	$8.9 \cdot 10^{-7}$	1300	30	0.20	0.60 ^b

206

207

208 Table 1. Some thermophysical properties of air and condenser. k : Thermal conductivity. ν : kinematic
 209 viscosity. C_p : Specific heat; ρ : density; α : volumic thermal expansion coefficient; ε : emissivity,
 210 hemisphere integrated, wavelength window. (a): Average for atmosphere, see Beysens (2018). (b) 5 μ m
 211 wavelength.

212

213 Boundary conditions for air velocity are the same as with CFX, namely air velocity follows the
 214 classical logarithmic variation with respect to elevation z according to Eq. (1), with $z_c = 0.1$ m:

215 (ii) “No-slip” conditions on condenser surfaces and ground.

216 (iii) “Open frontier” on ceiling and the two vertical sides of the fluid domain.

217 (iv) “Convective flow” at the exit.

218 Heat transfer from the air to the surface of the condenser occurs by forced convection. The complete
 219 energy equation has thus to be solved in the fluid domain using the velocities found from the solutions
 220 of the continuity equation and conservation of momentum equation (Navier-Stokes). The energy
 221 equation describing this heat transfer process is given by:

222

$$\rho C_p \frac{DT}{Dt} = k \nabla^2 T \quad (2)$$

223

224

225 where ρ is the density of air, C_p is the specific heat of air, k is the thermal conductivity of air, and T the
 226 air temperature. The effect of convection on the heat transfer process is taken into account in the
 227 derivative term DT/Dt of Eq. 2:

228

$$229 \quad \frac{DT}{Dt} = \frac{\partial T}{\partial t} + V \cdot \text{grad}(T) \quad (3)$$

230

231 where V is the velocity field in the air. The steady state case was solved, such as the time derivative
 232 term $\partial T/\partial t$ in Eq. 3 is set to zero. A condition of continuity of heat flow has been taken for the upper
 233 edges and the external face of the cones.

234 The temperature of the condenser will be lower than that of the ambient air due to the radiative
 235 cooling. Natural convection will thus appear, enhancing heat transfer between air and condenser. This
 236 effect can be taken into account by a temperature dependence of the fluid density (buoyancy effect) in
 237 the conservation of momentum equation. However, the coupling forced-natural convection is difficult
 238 to account (it needs a long iterative process) and the effect of convection, which matters only for weak
 239 air flow velocity is neglected in this study.

240 In the solid domain, conduction is modeled using a simple heat equation. No-slip boundary condition
 241 is for all condenser surfaces. The Reynolds number, which compares the inertial forces with the
 242 viscous forces and determines whether the flow is turbulent or laminar, reads as

243

$$244 \quad \text{Re} = \frac{VL}{\nu} \quad (4)$$

245

246 Here $L \approx 10$ m is the condenser typical lengthscale; ν is air kinematic viscosity and V is wind speed,
 247 giving $\text{Re} \approx 7 \times 10^5 V$. Laminar flow is expected for a Reynolds number lower than a critical value,
 248 which depends on the geometry. It is on order of 2000 for cylinders but can be reduced depending on
 249 the geometry. As a matter of fact, turbulence was observed even for wind speed as low as 10^{-4} m.s⁻¹.

250 The Navier Stokes equations are solved with the $k - \varepsilon$ turbulence module because it performs quite
251 well for external flow problems around complex geometries.

252 After some trials, it was considered for the simulation a box of 75 m long, 20 m wide and 8 m high
253 whose top is at 8m above the lower part of the cones, in accord with the specifications noted in Section
254 2.2. The large value of the box length is necessary because the effects of pressure matter and were
255 seen on smaller boxes.

256 All of the meshes created in COMSOL Multiphysics were physics controlled and automatically
257 generated. The meshes varied in size ranging from a coarser mesh all the way up to a normal mesh.
258 Several mesh sensitivity tests were carried out to demonstrate the dependency of the results on various
259 types of meshes. No relevant differences have been identified in the velocity field of the upstream and
260 downstream flow and in the temperature field.

261 The final aim of the CFD numerical is to estimate the dew water output with respect to the
262 meteorological parameters. This is a very general and difficult task and up to now several
263 simplifications have been made to retain only the most important parameters. As the actual yield per
264 surface area, h , depends on the supersaturation reached by the condensing substrate, the determination
265 of the temperature drop ($T_a - T_c$) (T_c is the condenser surface temperature) with respect to air
266 temperature can represent the dew yield as shown in Appendix 1. Total yield is calculated from h and
267 the condensing surface area, S_c ($=hS_c$).

268

269 **3. Results and discussion**

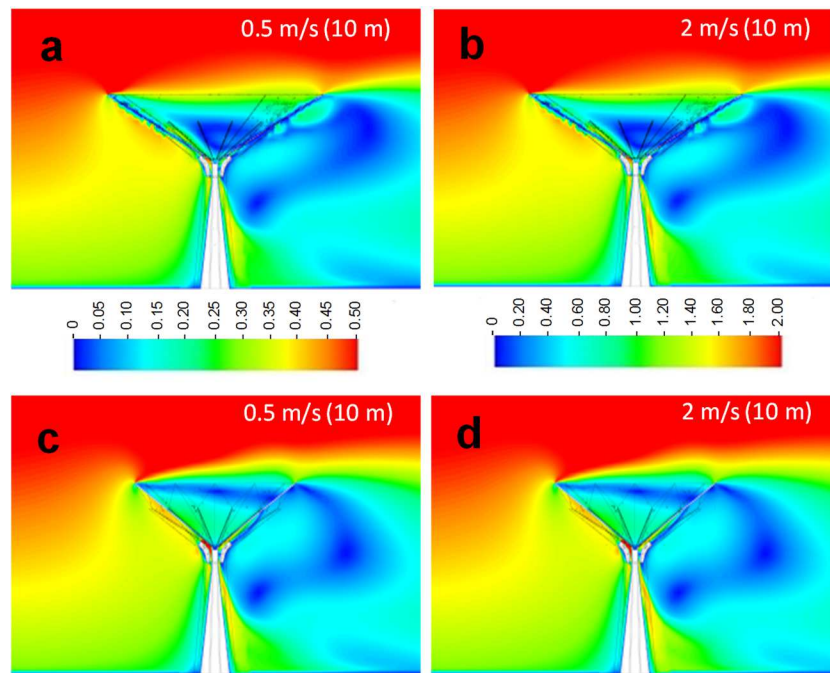
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271 *3.1. Air flow*

272 The two S- and W-shapes have been tested with the Ansys CFX simulation with incoming air velocity
273 0.5, 1, 2, 3, 4, 5 m.s⁻¹ at 10 m elevation. The main result is that, while the inside air flows are different
274 for S-and W-shapes, the variation of wind speed does not appreciably affect the aerodynamic
275 performances of the two cones. One thus only report data in Figs. 3 and 4 corresponding to the wind
276 speed values 0.5 and 2 m.s⁻¹.

277 Figures 4abcd compares the aerodynamic effects on a symmetry plane along the wind direction for S-
 278 and W- cones for wind speed values 0.5 and 2 m.s⁻¹. Increasing the wind speed only reinforces the fact
 279 that lower wind speed is seen in the S-cone central air volume than in the W-cone. In addition, above
 280 the cones, higher streamline development above the W-cone signs a more turbulent airflow than above
 281 the S-cone.

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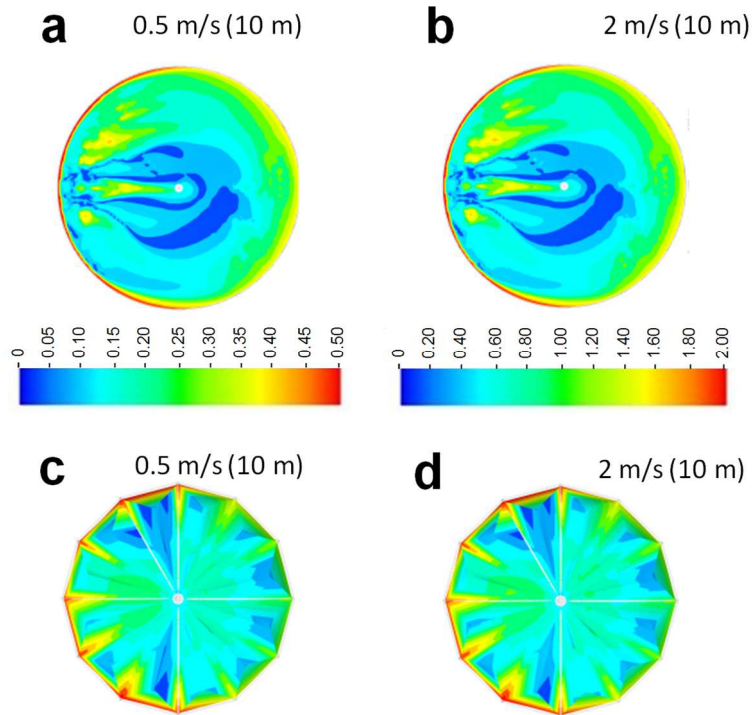


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284 Fig. 3. Air flow (from left to right) at two different incoming velocities on vertical sections at the cone axes, on the symmetry
 285 plane along the wind direction. (a,b): S-cone. (c,d): W-cone. Air flow is from left to right.

286 Figures 4abcd visualize the wind velocity on the internal surface of the condenser, which is of prime
 287 importance for convective heat transfer. Increasing the incoming wind velocity changes has very little
 288 effect. In contrast, wind on S-cone internal surface is seen to be lower than on the W-cone surface.

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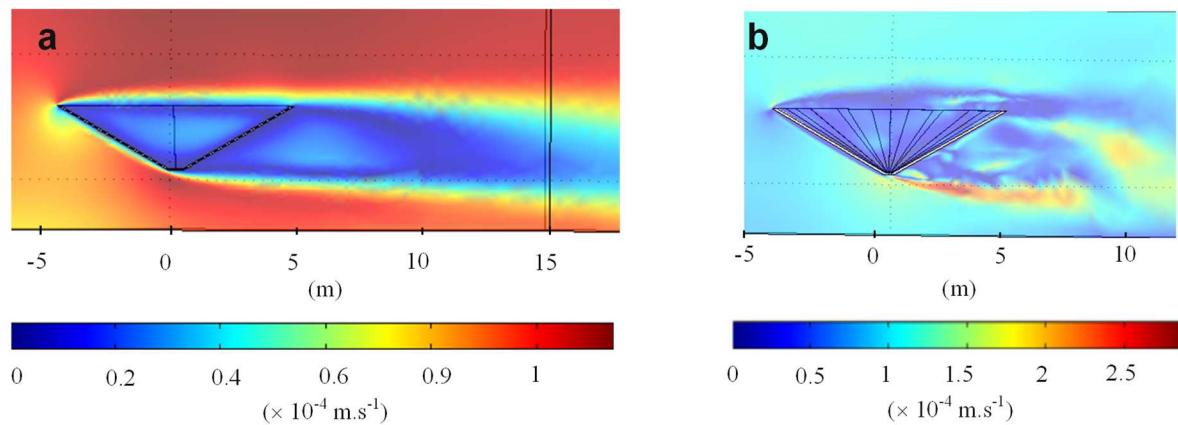
292 Fig. 4. Air flow (from left to right) at two different incoming velocities on vertical sections at the cone axes, on the symmetry
 293 plane along the wind direction. (a-b): S-cone. (c-d): W-cone. Air flow is from left to right.

294

295 One thus sees that condenser' surface velocity is largely affected by external and internal geometry.
 296 Smoother geometries help to maintain low wind speed at condenser' surface. Velocity at condenser'
 297 surface is higher in the W-cone than in the S-cone as caused by the triangulated external surfaces and
 298 exposed highest edges of the condenser. One thus expects enhanced convective heat transfer with the
 299 W-cone.

300 Some studies have been carried out with COMSOL at two typical wind speeds at 10 m elevation, a
 301 very low speed of 10^{-4} m.s^{-1} where the Reynolds number $Re \approx 70$ and a speed of 2 m.s^{-1} ($Re \approx$
 302 1.4×10^6). The latter value is a mean value for dew formation (which in average occurs between 0 and
 303 4.4 m.s^{-1} , see Beysens, 2018) and corresponds to the mean wind speed during night at the site where
 304 such condensers should be implemented. The low speed value (10^{-4} m.s^{-1}) was chosen because the
 305 simulation with CFX, which was looking for variations in aerodynamic patterns, did not find
 306 significant differences between high and low wind speed. The aerodynamics of the design performs

307 similarly at all wind conditions tested. Then the decision of modeling the wind at 10^{-4} m.s^{-1} has been
 308 taken as this represents a “little to no-wind” scenario that is likely to be experienced on site. In
 309 addition, this low wind condition helps to reach convergent solutions in a reasonably short time span
 310 by reducing turbulent effects. Although turbulence is seen even at low wind speed (Fig. 5), stagnation
 311 of the flow is also observed, which limits the convective heat exchanges and facilitates dew formation.
 312



313
 314 Fig. 5. Turbulence observed at low wind speed 10^{-4} m.s^{-1} for (a) S-cone and (b) W-cone. Air flow is from left to right.

315

316 3.2. Surface temperature

317

318 Temperature of the inner surface of the cones are reported in Figs. 6abcd for both W- and S-cones at
 319 the two wind speeds of 10^{-4} m.s^{-1} and 2 m.s^{-1} and summarized in Table 2. For 10^{-4} m.s^{-1} windspeed, the
 320 mean cooling with respect to air temperature is found to be 5.85 K (S-cone) and 6.55 K (W-cone), and
 321 for 2 m/s windspeed, 5.49 K (S-cone) and 5.25 K (W-cone). Corrugation increases on one hand
 322 radiative cooling but on the other hand reduces cooling by enhanced convective heat exchange. At low
 323 speed, convective heat exchange is similar for both smooth and corrugated surfaces, and corrugation
 324 increases cooling for the W-cone. At higher air flow velocities, convective heat exchange is larger for
 325 the W-cone and cooling is eventually found smaller than for the smooth cone. However, the difference
 326 (0.24 K) remains small. Details of air temperature are also shown in Fig. 6ef. Air is seen to cool down
 327 after having passed near the cone edge.

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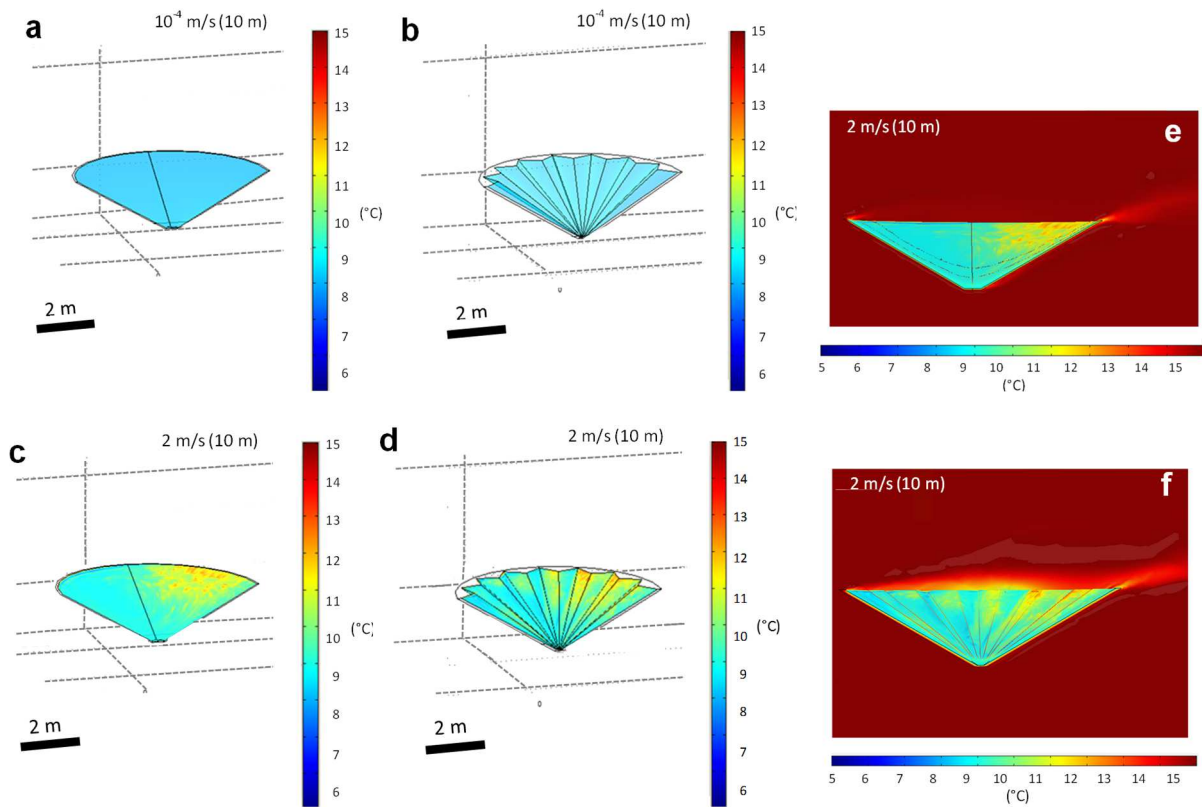


Fig. 6. (a-d): Surface temperatures of S- cone and W-cones. Air flow is from left to right. (a,b): Windspeed 10^{-4} m.s^{-1} . (c,d): Windspeed 2 m.s^{-1} . Air temperature is shown in (e,f). Significant cooling with respect to the initial temperature of 15°C is observed.

335

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338

339

340 4. Conclusion

341

342

343

From a detailed analysis of air flows around corrugated W-cone and smooth S-cone, it appears clearly that, although the inside air flows are different for the two shapes, the variation of wind speed does not

	$V(\text{m.s}^{-1})$	$\langle T_c \rangle$ ($^\circ\text{C}$)	$\langle T_c \rangle - T_a$ ($^\circ\text{C}$)
S-cone	10^{-4}	9.15	5.85
	2	9.51	5.49
W-cone	10^{-4}	8.45	6.55
	2	9.75	5.25

Table 2. Results of the simulation at two different wind speed V showing the mean surface temperature $\langle T_c \rangle$ and mean cooling temperature $\langle T_c \rangle - T_a$ with respect to air temperature $T_a = 15^\circ\text{C}$.

344 appreciably affect the aerodynamic performances of the two cones. The scooping effect however
345 performs better in the S-shape than in the W-shape, with wind speed on the internal surface of the S-
346 cone lower compared to the W-shape. The shape of the crown in the S-shape cone indeed amplifies the
347 upward trajectory of the wind that is redirected away from the inner area of the cone as it hits the edge
348 of attack; this geometrical feature helps significantly in reducing the wind speed on the internal surface
349 of the S-cone compared to the W-shape.

350 From these air flow characteristics, it naturally follows that the W-cone, which radiates more, also
351 cools more at low windspeed than the S-cone where the convective heat transfer is low. In contrast, at
352 large windspeed, the W-cone cools less as an effect of enhanced convective heat transfer in the
353 corrugated structure. For typical air flow velocity of 2 m.s^{-1} , the difference in mean cooling between
354 both shapes remains, however, small (0.24 K). Since cooling temperatures are nearly proportional to
355 dew yield for the weak cooling temperatures as encountered in dew condensation, it means that the
356 corrugated W-cone, which exhibit better dew drop collecting properties, will give a final better dew
357 collection yield per surface area than the corresponding smooth S-cone. Total yield will obviously has
358 to account for the different W and S cone actual surface areas. In the example reported here with 24
359 faces for the W-cone, the S-cone exhibits a surface larger than the W-cone.

360

361

362 **References**

363

364 Agam, N., Berliner, P.R., 2006. Dew formation and water vapor adsorption in semi-arid
365 environments—a review. *J. Arid. Environ.* 65, 572–590.

366 Beysens, D., 2018. *Dew water* (Rivers Publisher, Gistrup).

367 Beysens, D., Brogini, F., Milimouk-Melnytchouk, I., Ouazzani, J., Tixier, N., 2013. New
368 architectural forms to enhance dew collection. *Chemical Engineering Transactions* 34, 79-84.

- 369 Bliss R.A., 1961. Atmospheric radiation near the surface of the ground. *Solar Energy* 5, 103-20.
- 370 Blocken, B., Stathopoulos, T., Carmeliet, J., 2006. CFD simulation of the atmospheric boundary layer:
371 wall function problems. *Atmospheric Environment* 41, 238-252.
- 372 Clus, O., Ouazzani, J., Muselli, M., Nikolayev, V. S., Sharan, G. Beysens, D., 2009. Comparison of
373 various radiation-cooled dew condensers using computational fluid dynamics. *Desalination* 249,
374 707-712.
- 375 Kaseke, K., Wang, L., Seely, M., 2017. Non-rainfall water origins and formation mechanisms. *Science*
376 *Advances* 3, e1603131.
- 377 Kaseke, K., Wang, L., 2018. Fog and dew as potable water resources - maximizing harvesting
378 potential and water quality concerns. *GeoHealth* 2, 327-332.
- 379 Liu, M., Cen, Y., Wang, C., Gu, X., Bowler, P., Wu, D., Zhang, L., Jiang, G., Beysens, D., 2020.
380 Foliar uptake of dew in the sandy ecosystem of the Mongolia Plateau: A life-sustaining and carbon
381 accumulation strategy shared differently by C3 and C4 grasses. *Agricultural and Forest*
382 *Meteorology* 287, 107941.
- 383 Medici, M.-G., Mongruel, A., Royon, L., Beysens, D., 2014. Edge effects on water droplet
384 condensation, *Phys. Rev. E* 90, 062403.
- 385 Mochida, A., Tominaga, Y., Yoshie, R., 2006. AIJ guideline for practical applications of CFD to wind
386 environment around buildings. The 4th International Symposium on Computational Wind
387 Engineering (CWE2006, Yokohama) 533-536.
- 388 Pal Arya, S., 1988. *Introduction to Micrometeorology* (Academic Press, Inc., San Diego).
- 389 Tomaszewicz, M., Abou Najm, M., Beysens, D., Alameddine, I., El-Fadel, M., 2015. Dew as a
390 sustainable non-conventional water resource: a critical review. *Environmental Reviews*, 23, 425-
391 442.
- 392
- 393 **Appendix 1**
- 394

395 This calculation reproduces the main features of the model developed in Beysens (2018). When
 396 considering an energy balance equation it is possible to elaborate a relation between surface
 397 temperature without condensation, under dry air, and dew yield obtained with humid air at same
 398 temperature.

399 Let us consider a condenser surface thermally isolated from below of surface area S_c . For condenser
 400 mass dM around a point of coordinate (x, y, z) with condensing surface area dS_c , the energy balance
 401 reads as:

$$402 \frac{dT_c}{dt} (dM C_c + dm C_w) = R_i dS_c + R_{he} + R_{cond} \quad (A1)$$

403
 404 Here T_c is the surface temperature of the condenser, dm is the mass of water condensate, C_c and C_w are
 405 the specific heats of the condenser materials and water, respectively. Without condensation $R_{cond} = 0$
 406 and at equilibrium $dT/dt = 0$, the condenser surface temperature reaches T_{c0} under radiative cooling
 407 $R_i(x,y,z)$ balanced by convective heat losses $R_{he} = a(x,y,z) (T_a - T_{c0})$. The latter us Newton law with a the
 408 convective heat transfer coefficient. With T_a the air temperature, Eq. A1 becomes:

$$409 0 = R_i(x, y, z) - a [T_a - T_{c0}(x, y, z)] \quad (A2)$$

410
 411 It follows the determination of the local convective heat transfer coefficient $a(x, y, z)$:

$$412 a(x, y, z) = \frac{R_i(x, y, z)}{[T_a - T_{c0}(x, y, z)]} \quad (A3)$$

413
 414 Let us now consider dew condensation. The condensation rate is $R_{cond} = L_c \dot{m}$ where $\dot{m} = dm/dt$. At
 415 thermal equilibrium $dT/dt = 0$, it follows from Eq. A1 for the dew yield per unit surface:

$$416 \frac{d\dot{m}}{dS_c} = \frac{R_i(x, y, z)}{L_c} \left[1 - \frac{T_a - T_c}{T_a - T_{c0}(x, y, z)} \right] \quad (A4)$$

421

422 Making the simplification $T_c \approx T_d$, with ρ_w the liquid water density, the dew yield per unit surface area
 423 can be written as

424

$$425 \quad \dot{h} = \frac{1}{\rho_w} \frac{dm}{dS_c} \approx \frac{R_i(x,y,z)}{\rho_w L_c} \left[1 - \frac{T_a - T_d}{T_a - T_{c0}(x,y,z)} \right] \quad (\text{A5})$$

426

427 Defining the mean value of variable $Q(x, y, z)$ on surface S_c by

428

$$429 \quad \langle Q \rangle = \frac{1}{S_c} \int_{S_c} dS_c, \quad (\text{A6})$$

430

431 equation A5 can be rewritten as

432

$$433 \quad \langle \dot{h} \rangle = \frac{1}{\rho_w L_c} \left[\langle R_i(x, y, z) \rangle - (T_a - T_d) \left\langle \frac{R_i(x,y,z)}{T_a - T_{c0}(x,y,z)} \right\rangle \right] \quad (\text{A7})$$

434

435 Depending on the shape of the condenser, approximations can be made to relate $\langle \dot{h} \rangle$ to $\langle R_i \rangle$ and
 436 $\langle T_a - T_{c0} \rangle$. Either R_i is a weak function of (x, y, z) ,

437

$$438 \quad \langle \dot{h} \rangle = \frac{R_i}{\rho_w L_c} \left[1 - \frac{(T_a - T_d)}{T_a - \langle T_{c0}(x,y,z) \rangle} \right], \quad (\text{A8})$$

439

440 or T_{c0} is a weak function of (x, y, z) :

441

$$442 \quad \langle \dot{h} \rangle = \frac{\langle R_i(x,y,z) \rangle}{\rho_w L_c} \left[1 - \frac{(T_a - T_d)}{T_a - T_{c0}} \right] \quad (\text{A9})$$

443

444 Both Eqs. A8 and A9 can be rewritten as

445

446
$$\langle \dot{h} \rangle = \frac{\langle R_i \rangle}{\rho_w L_c} F_{a,d}(T_{c0}) \quad (\text{A10})$$

447

448 where the function $F_{a,d}(T_{c0})$ is

449

450
$$F_{a,d}(T_{c0}) = 1 - \frac{T_a - T_d}{T_a - \langle T_{c0} \rangle} \quad (\text{A11})$$

451

452 Figure A1 represents the variations of $F_{a,d} = 1 - 3.4/(T_a - \langle T_{c0} \rangle)$ as a function of $T_a - \langle T_{c0} \rangle$

453 corresponding to the current nightly conditions $T_a = 288.15$ K (15 °C), RH = 80%, with dew point

454 temperature $T_d = 11.6^\circ\text{C}$ and $T_a - T_d = 3.4^\circ\text{C}$. Data have been drawn only for positive \dot{h} values, that is for

455 $T_a - \langle T_{c0} \rangle > 3.4^\circ\text{C}$, and a temperature cooling range that does not exceeds the maximum value 10°C .

456 In this range [3.4°C - 10°C], the variation of F can be approximated by the linear relationship

457

458
$$F_{a,d} \approx A(T_a - \langle T_{c0} \rangle) \quad (\text{A12})$$

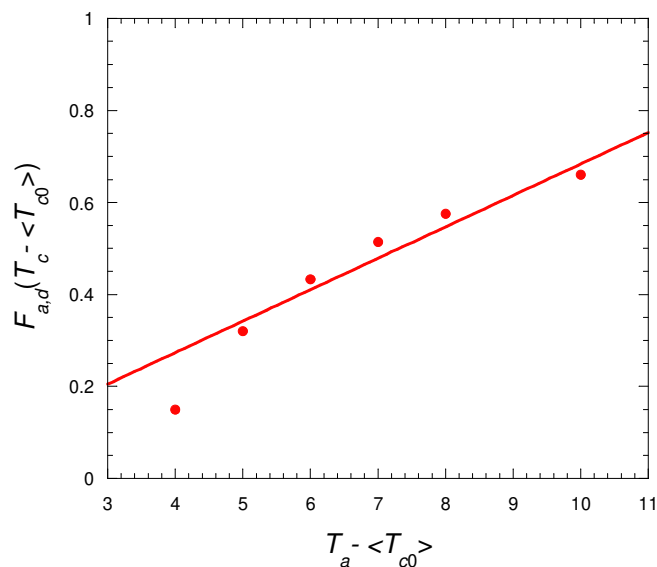
459

460 This linear relationship thus means that the dew yield \dot{h} is approximately proportional to the

461 temperature cooling ($T_a - \langle T_{c0} \rangle$). Such a proportionality has indeed been observed by Clus et al.

462 (2009) when comparing measured dew yields and temperature drop determined by CFD.

463



464

465 Fig. A1. Variations of $F_{a,d}$ as a function of $T_a - \langle T_{c0} \rangle$ according to Eq. A11 for typical night conditions $T_a = 288.15$ K
466 (15°C), $\text{RH} = 80\%$, corresponding to $T_a - T_d = 3.4^\circ\text{C}$, in the range $[3.4^\circ\text{C} - 10^\circ\text{C}]$ (see text). The line is a fit to Eq. A12 with $A =$
467 0.068 ± 0.003 and a correlation coefficient $R = 0.95$.