

Season-Wise Three-Dimensional CFD Modelling of a Natural Evaporative Water Cooler: Development, Calibration and Experimental Validation

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Abstract: Natural evaporative water coolers provide refrigerant-free, ultra-low-energy water cooling for hot-dry climates, yet computational fluid dynamics (CFD) studies of such systems at full device scale are absent from the literature. This paper develops a three-dimensional CFD model of a complete natural evaporative water cooler in OpenFOAM and validates it season-wise against the authors' experimental campaign at the Taguchi-optimised operating point of each season. The water side resolves the entire 76.2 m helical copper coil (inner diameter 4.5 mm, 80.85 turns) with a structured butterfly O-grid mesh of 155,200 hexahedral cells generated by a purpose-built sweep algorithm; the wetted evaporative pad is represented by an external film boundary condition referenced to the ambient wet-bulb temperature. A companion air-side model treats the pad-coil assembly as a Darcy–Forchheimer porous zone inside the cooler housing. The film coefficient was calibrated once, at the summer optimum only ($h = 41 \text{ W/m}^2\text{K}$); the same value was reused unchanged for the winter simulation — making winter a genuine blind prediction — while the monsoon value ($h = 28 \text{ W/m}^2\text{K}$) was estimated a priori from the resistance network for the reduced sprinkle rate and humid air. The model reproduces the measured seasonal temperature drops of 9.7, 5.4 and 5.6 °C with deviations of -0.5%, -1.6% and -1.2% for summer, monsoon and winter respectively, and cooling effectiveness within 2 percentage points of experiment in all seasons. The calibrated wetted-pad conductance is 1.9 times the dry-pad analytical estimate, quantifying the contribution of direct film evaporation, and the resolved Dean vortices ($De \approx 340$) explain the weak experimental sensitivity to coil water velocity in summer.

Keywords: evaporative cooling; natural water cooler; CFD; OpenFOAM; helical coil; Dean vortices; seasonal validation.

1. INTRODUCTION

The rising global demand for cooling, combined with the environmental burden of vapour-compression refrigeration, continues to drive interest in passive and quasi-passive alternatives. Direct evaporative cooling exploits the latent heat of vaporisation of water and, in its simplest embodiment, requires only water, air and a fan of a few tens of watts [1],[2]; the breadth of research on and field deployment of the technique has been surveyed at national scale [3]. In hot-dry climates the technique approaches the ambient wet-bulb temperature and achieves effectiveness values of 70–85% [4], [5]. While evaporative cooling of air is extensively documented, evaporative cooling of water in which a water stream circulating through a coil is cooled by evaporation from a wetted pad wrapped around it has received far less attention.

In the authors' earlier experimental work, a natural evaporative water cooler was evaluated and optimised across three seasons using a Taguchi L27 design with analysis of variance. That work established season-specific optima — pad material dominating in summer, sprinkled water flow rate in monsoon and coil water velocity in winter — and validated a one-dimensional heat-transfer model to within 2% of measurement. Its stated future scope identified CFD simulation as the natural next step. Because the optimisation itself is season-specific, a meaningful CFD companion must likewise be validated season-wise, at each seasonal optimum, rather than at a single operating point.

CFD studies relevant to this device class fall into two disconnected groups. Curved- and helically-coiled-tube heat transfer has a rich literature: Dean [6], [7] first described the counter-rotating secondary vortices induced by centrifugal force; recent experiments and simulations have refined the description of the secondary motion in curved pipes [8] and resolved the local heat-transfer behaviour of helical coils [9], and correlations for friction and



heat transfer in helical coils are reviewed by Naphon and Wongwises [10]. Separately, evaporative-pad and porous-media air-cooling models are well established [11], [12], numerical heat- and mass-transfer models of direct evaporative coolers are available [13], and the thermal performance of cellulosic pad media has been characterised experimentally [14]. To the authors' knowledge, however, no published work resolves the complete coil of a natural evaporative water cooler in three dimensions — a 76.2 m tube of 4.5 mm bore presents an aspect ratio of about 17,000:1, which defeats standard meshing utilities and demands careful treatment of the pressure–velocity system.

The present paper makes four contributions. First, a purpose-built structured-mesh generator sweeps a butterfly O-grid along the full helix, producing an all-hexahedral mesh of the entire 80.85-turn coil that passes all OpenFOAM quality checks. Second, the wetted evaporative pad is condensed into a physically interpretable external film boundary condition whose conductance, calibrated once at the summer optimum, is compared with the dry-pad analytical estimate to quantify the effect of surface wetting. Third, the model is validated season-wise at all three experimental optima under a deliberately restrictive calibration policy — one calibration point, one a priori estimate, one blind prediction — yielding deviations of 0.5–1.7%. Fourth, a companion air-side porous-media model of the housing verifies the airflow distribution and air-state change across the pad in each season.

2. SYSTEM DESCRIPTION AND SEASONAL OPERATING CASES

The modelled device is the natural evaporative water cooler characterised experimentally earlier by the authors. Water flows through a helical copper coil (inner diameter 4.5 mm, outer diameter 5.5 mm, total length 76.2 m) wound into a cylindrical bank of mean diameter 300 mm and pitch 7 mm (80.85 turns, coil height 566 mm; winding diameter and pitch inferred from the physical unit). A viscose-fibre pad of 4 mm thickness wraps the coil bank and is kept saturated by a gravity-fed sprinkler; a top-mounted fan draws ambient air through side louvres and across the pad. Fig. 1 presents a schematic line diagram of the setup.

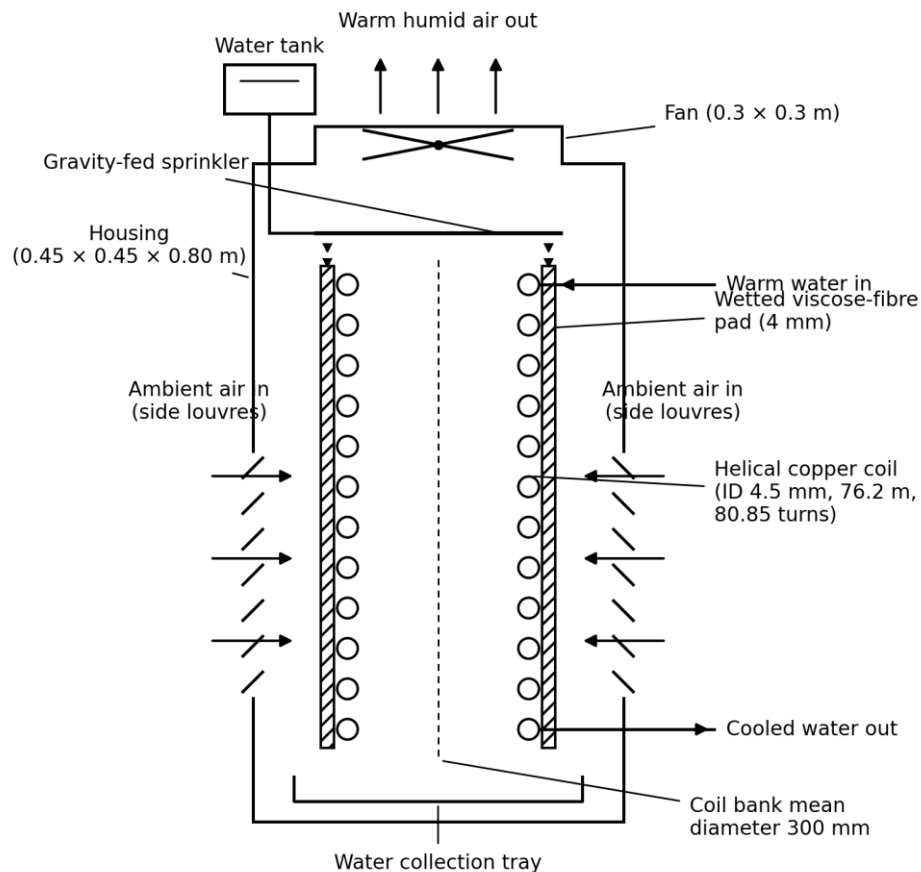


Fig. 1. Schematic line diagram of the natural evaporative water cooler: helical copper coil bank wrapped by a wetted viscose-fibre pad, gravity-fed sprinkler, side louvres and top-mounted fan.

Three operating cases are simulated — the Taguchi-optimal configuration of each season identified in the experimental work. All seasons share the viscose-fibre pad and 10 km/h air velocity; the sprinkle rate and coil

water velocity differ as per the seasonal optima. Since experimental work reports seasonal ambient ranges rather than the exact ambient of each optimal trial, representative ambient conditions were selected within the reported ranges so as to be jointly consistent with the season's measured ΔT , cooling effectiveness η_c and normalised temperature drop ΔT_{norm} ; the resulting inlet water temperatures follow from these psychrometric identities. Table I lists the three cases.

TABLE I. Seasonal Taguchi-Optimal Cases Simulated (Performance Data from the Authors' Experiments)

Quantity	Summer	Monsoon	Winter
Sprinkle flow / coil velocity	6 L/hr / 0.5 m/s	5 L/hr / 0.3 m/s	6 L/hr / 0.5 m/s
Dry-bulb / RH / wet-bulb ($^{\circ}\text{C}$ / % / $^{\circ}\text{C}$)	42 / 19 / 20.4	33 / 65 / 27.4	24 / 38 / 15.0
Inlet water temperature ($^{\circ}\text{C}$)	34.0	34.4	22.8
Measured ΔT ($^{\circ}\text{C}$) / η_c (%)	9.7 / 82	5.4 / 72	5.6 / 76
Water mass flow (g/s)	7.72	4.63	7.72

3. CFD METHODOLOGY

A. Decoupled Two-Domain Strategy

A single conjugate simulation of water, tube wall, evaporating pad film and housing air would require a mesh spanning length scales from the 0.5 mm tube wall to the 0.8 m housing together with multiphase evaporation physics — computationally prohibitive and unnecessary for the quantities of interest. The problem is therefore decoupled into (i) a water-side model resolving the complete coil interior, with all external heat-transfer mechanisms lumped into a film boundary condition, and (ii) an air-side model of the housing in which the pad-coil assembly is a porous zone with an evaporative cooling constraint. The air-side model corroborates the pad-face velocity and the near-uniform, near-wet-bulb air state that parameterise the water-side wall condition.

B. Water-Side Mesh Generation

The coil interior is meshed by sweeping a two-dimensional butterfly O-grid cross-section (a 4×4 Cartesian core blended into four boundary-fitted blocks; 80 quadrilaterals) along the helix centreline in 1,940 axial stations (24 per turn), using a rotation-minimising local frame. The resulting 155,200 hexahedral cells (172,749 points) are written directly in OpenFOAM polyMesh format by a purpose-built generator. Mesh quality (checkMesh): maximum non-orthogonality 17.2° (average 4.6°), maximum skewness 1.99, maximum aspect ratio 19.6, all checks passed (Fig. 2).

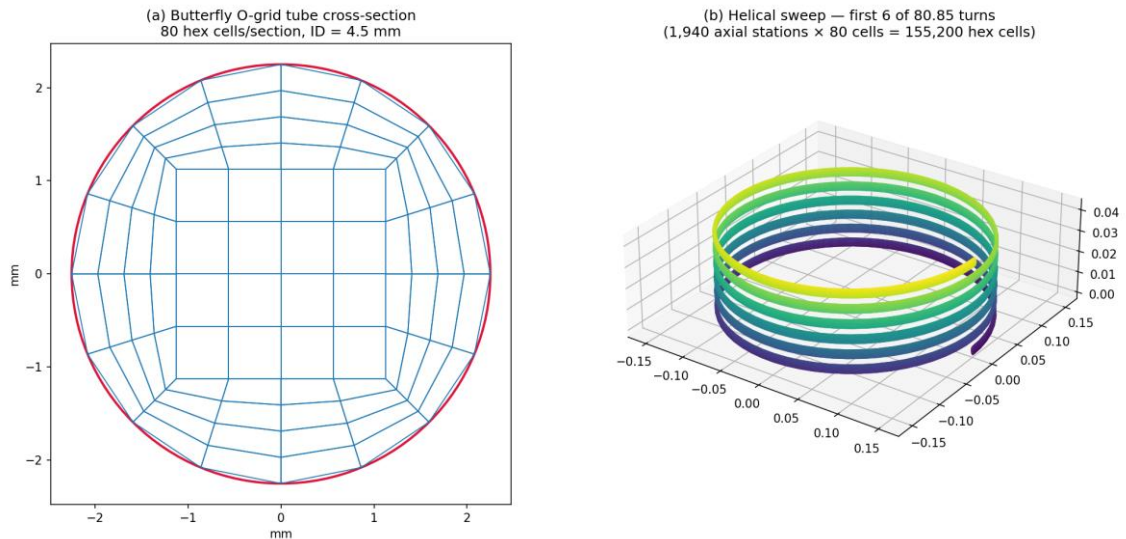


Fig. 2. Water-side mesh: (a) butterfly O-grid cross-section (80 hexahedral cells, ID 4.5 mm); (b) helical sweep, first 6 of 80.85 turns shown.

C. Governing Equations and Physical Models

Steady incompressible flow of water with constant properties ($\rho = 996 \text{ kg/m}^3$, $c_p = 4180 \text{ J/kg}\cdot\text{K}$, $\mu = 8.0 \times 10^{-4} \text{ Pa}\cdot\text{s}$, $\text{Pr} = 5.44$) is solved with the SIMPLE algorithm [15] in OpenFOAM v1912. The conservation equations are:

$$\nabla(\rho U) = 0 \quad (1)$$

$$\nabla(\rho U U) = -\nabla p + \nabla(\mu \nabla U) \quad (2)$$

$$\nabla(\rho U c_p T) = \nabla(k \nabla T) \quad (3)$$

The tube Reynolds numbers are 2,750 (0.5 m/s, summer and winter) and 1,650 (0.3 m/s, monsoon), laminar–transitional; the flow is treated as laminar, consistent with the laminar correlations for curved circular ducts reviewed by Ghobadi and Muzychka [16], and the curvature-induced secondary motion is resolved directly. The Dean number,

$$De = Re \sqrt{D_i/D_c} \quad (4)$$

is ≈ 340 at 0.5 m/s and ≈ 200 at 0.3 m/s, placing all cases in the regime of strong counter-rotating secondary vortices [6], [7]; the pitch of the present helix is small enough that the flow follows the finite-pitch laminar behaviour reported for coiled channels of this class [17].

D. Evaporative-Pad Boundary Condition and Calibration Policy

All heat-transfer mechanisms outside the tube bore — wall conduction, conduction through the wetted pad, air-side convection and, dominantly, evaporation of the sprinkled water film — are lumped into a single film condition on the tube wall:

$$q'' = h_{\text{eff}} (T_{\text{wall}} - T_{\text{wb}}) \quad (5)$$

with the seasonal wet-bulb temperature as the thermodynamic sink. To avoid circularity in the validation, a deliberately restrictive calibration policy was adopted. The film coefficient was calibrated only once, at the summer optimum, and rounded to two significant figures: $h_{\text{eff}} = 41 \text{ W/m}^2\cdot\text{K}$ (referenced to the tube inner surface, $A = 1.07 \text{ m}^2$). For winter — which shares the summer sprinkle rate (6 L/hr) and coil velocity (0.5 m/s) — the identical value was reused with no re-tuning, so the winter result is a blind prediction. For monsoon, $h_{\text{eff}} = 28 \text{ W/m}^2\cdot\text{K}$ was estimated a priori from the series-resistance network,

$$1/UA = 1/(h_i A_i) + \ln(D_o/D_i)/(2\pi k_{\text{Cu}} L) + 1/(h_{\text{eff}} A_i) \quad (6)$$

accounting for the reduced sprinkle rate (5 L/hr, thinner wetted film) and the smaller evaporation driving force in humid air. No seasonal value was adjusted to match its experimental target; the residual deviations reported in Section 5 are therefore genuine model errors.

E. Air-Side Model

The housing ($0.45 \times 0.45 \times 0.80 \text{ m}$) is meshed with 57,344 hexahedral cells. The pad–coil annulus ($r = 0.13\text{--}0.17 \text{ m}$) carries a Darcy–Forchheimer momentum sink ($d = 5 \times 10^6 \text{ m}^{-2}$, $f = 500 \text{ m}^{-1}$), the viscous and inertial coefficients being scaled from measured pressure-drop data for cellulose pads at comparable face velocities [18], and a fixed-temperature evaporative constraint corresponding to a pad saturation effectiveness of ≈ 0.8 at each season's psychrometric state [4]. Turbulence is closed with the standard $k\text{--}\epsilon$ model of Launder and Spalding [19] with wall functions. Ambient air enters the side louvres at 1.65 m/s — giving the measured 2.78 m/s at the pad face — and leaves through the $0.3 \times 0.3 \text{ m}$ top fan opening (Fig. 3). The air-side case is re-solved for each season's ambient condition.

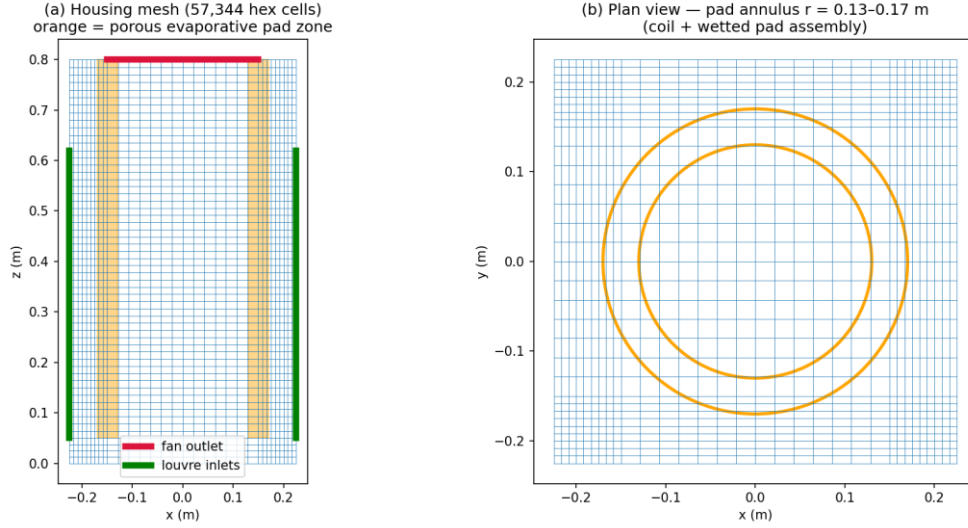


Fig. 3. Air-side housing mesh: porous pad zone (orange), louvre inlets (green), fan outlet (red).

F. Numerics and Convergence

Prescribing the mass flow at the coil outlet and pressure at the inlet enforces exact global mass conservation at every SIMPLE iteration — an important stabilisation for a domain of this aspect ratio, whose pressure Poisson system is stiff. Second-order schemes are used for gradients and momentum convection (linear-upwind), with bounded upwind for energy. Temperature fields were initialised from analytical exponential profiles to shorten the thermal transit. Convergence required $\approx 5,000$ iterations for the first (summer) case and ≈ 500 – $1,000$ for the seasonal restarts; at convergence the outlet temperature was stationary to <0.001 K per 100 iterations, the inlet/outlet mass imbalance below 0.15%, and the integrated wall heat flux consistent with the enthalpy balance $\dot{m}c_p\Delta T$. Air-side cases converged by residual control in 230–460 iterations.

4. RESULTS

A. Season-Wise Water Cooling

Figure 4 shows the streamwise decay of bulk water temperature for the three seasonal optima, and Fig. 5 the coil temperature fields. In all seasons the decay is quasi-exponential toward the seasonal wet-bulb asymptote, as expected for a constant-conductance exchanger cooling toward a fixed sink. Summer produces the deepest drop ($34.0 \rightarrow 24.35$ °C, $\Delta T = 9.65$ °C; duty 313 W) owing to the large wet-bulb depression (21.6 K). Monsoon, despite a comparable inlet temperature, is limited by the small depression (5.6 K) and reduced wetting, yielding $34.4 \rightarrow 29.09$ °C ($\Delta T = 5.31$ °C; duty 103 W at the lower 0.3 m/s coil velocity). Winter occupies the middle ground ($22.8 \rightarrow 17.27$ °C, $\Delta T = 5.54$ °C; duty 179 W). The seasonal ordering and magnitudes mirror the authors' experimental findings.

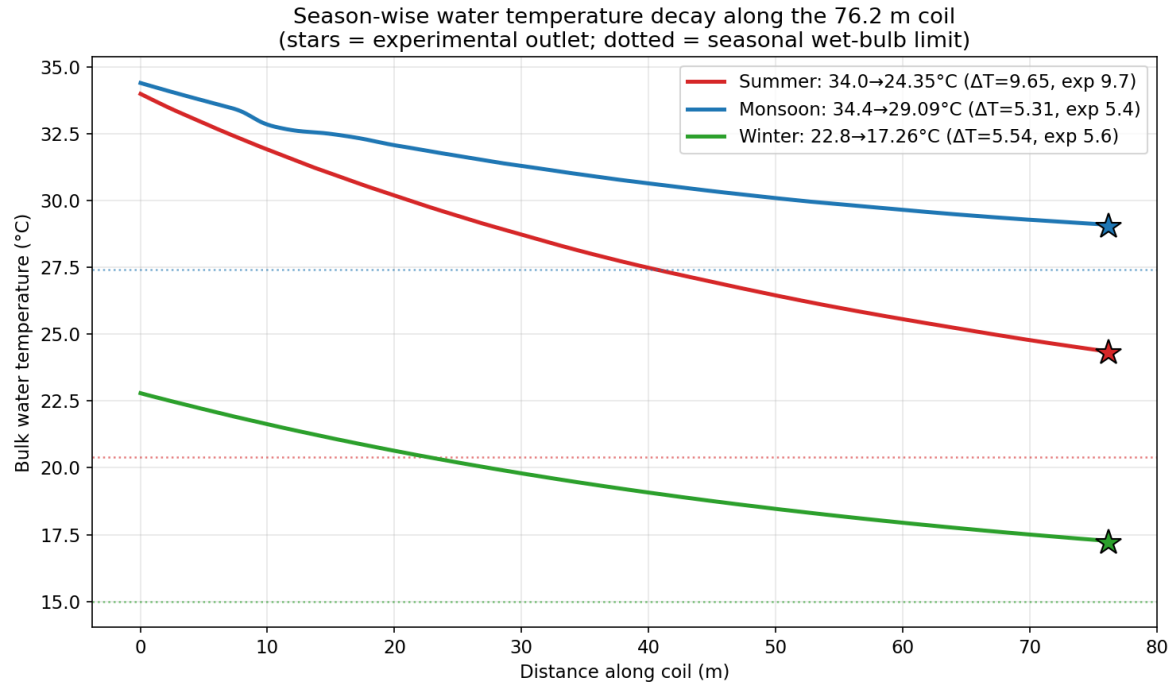


Fig. 4. Season-wise bulk water temperature along the 76.2 m coil; stars mark experimental outlet values, dotted lines the seasonal wet-bulb limits.

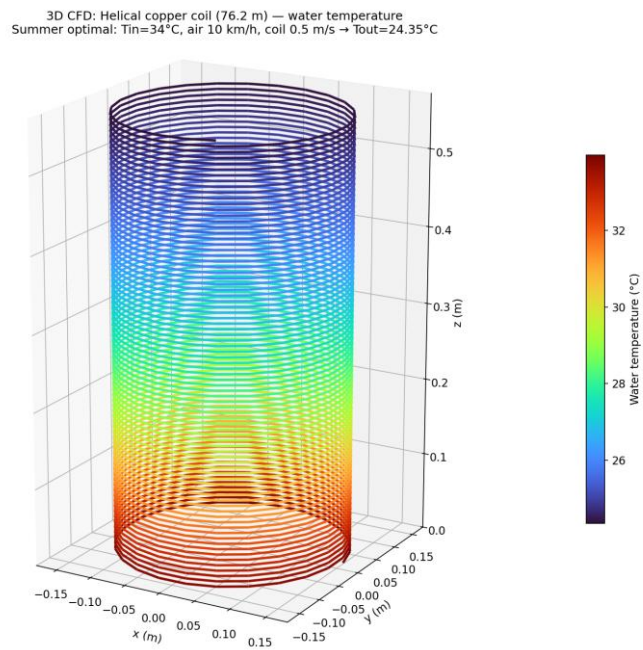


Fig. 5a. Summer optimum: coil coloured by water temperature ($34.0 \rightarrow 24.35^{\circ}\text{C}$).

Monsoon optimum — coil water temperature
 $T_{in}=34.4^{\circ}\text{C}$, $T_{db}=33^{\circ}\text{C}$, $\text{RH}\approx 65\%$, coil $0.3\text{ m/s} \rightarrow T_{out}=29.09^{\circ}\text{C}$ (exp 29.0°C)

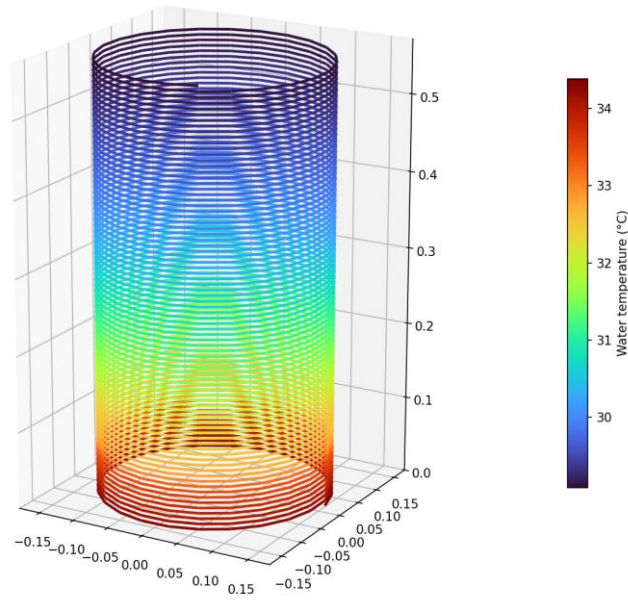


Fig. 5b. Monsoon optimum ($34.4 \rightarrow 29.09^{\circ}\text{C}$ at 0.3 m/s).

Winter optimum — coil water temperature
 $T_{in}=22.8^{\circ}\text{C}$, $T_{db}=24^{\circ}\text{C}$, $\text{RH}\approx 38\%$, coil $0.5\text{ m/s} \rightarrow T_{out}=17.26^{\circ}\text{C}$ (exp 17.2°C)

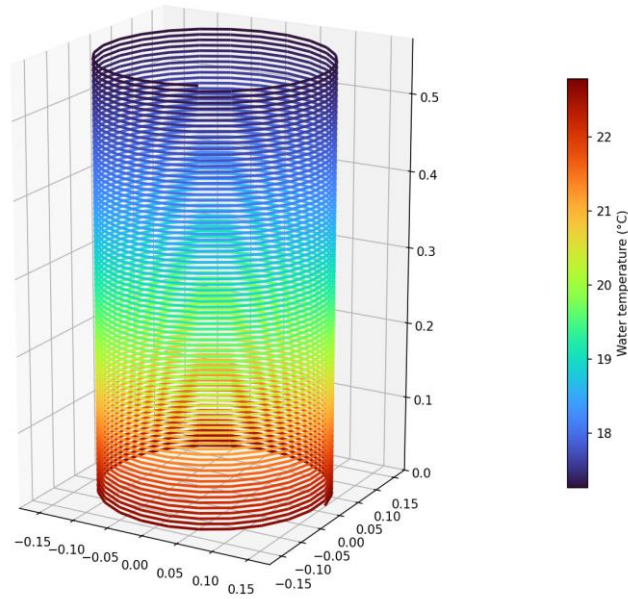


Fig. 5c. Winter optimum ($22.8 \rightarrow 17.27^{\circ}\text{C}$).

B. Dean Secondary Flow

Figure 6 presents tube cross-sections with in-plane velocity vectors superimposed on temperature contours (summer case). The centrifugal imbalance drives the classical pair of counter-rotating Dean cells: core fluid is flung toward the outer coil wall while cooled near-wall fluid returns along the top and bottom of the section. The axial-velocity peak is displaced toward the outer wall (Fig. 7) and the coldest fluid accumulates on the inner-coil side; cross-sectional temperature nonuniformity remains below 0.2 K because the external film, not internal mixing, limits the heat transfer. The internal film accounts for only $\approx 10\%$ of the total resistance at 0.5 m/s — consistent with the weak main effect of coil water velocity found by the summer ANOVA of the experimental

study — although at the monsoon optimum the lower velocity ($De \approx 200$) and its longer residence time become relevant, again as the seasonal ANOVA indicates.

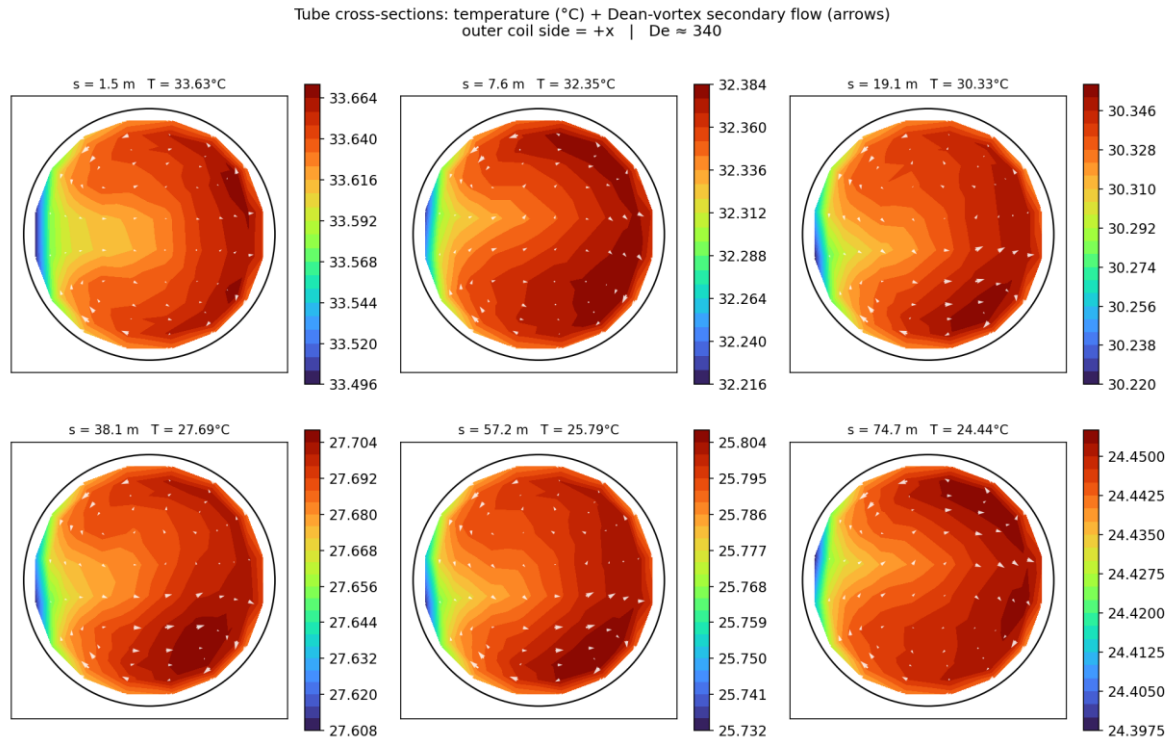


Fig. 6. Cross-sectional temperature with Dean-vortex secondary-flow vectors at six stations (summer optimum; outer coil side = +x).

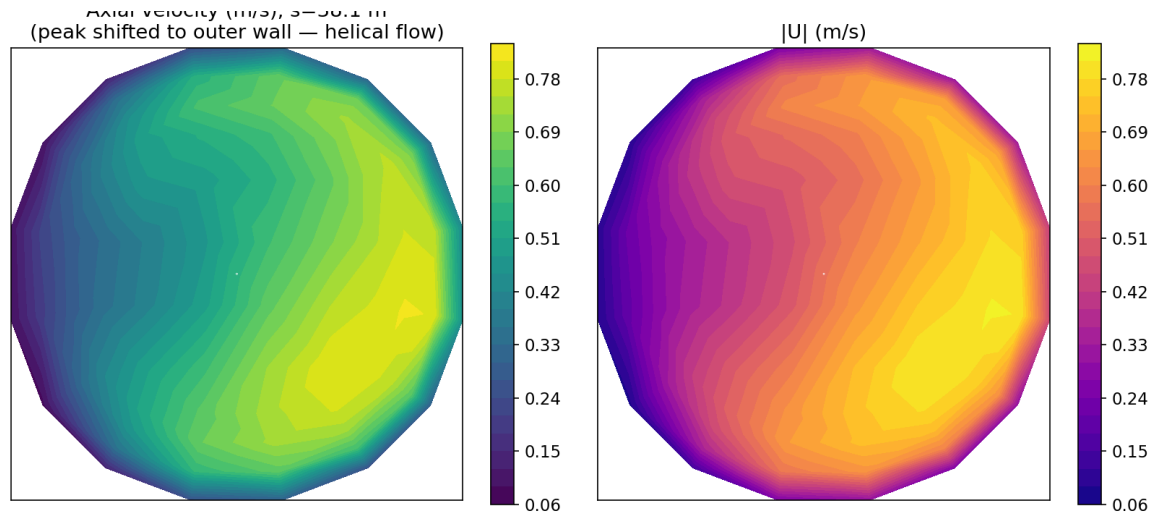


Fig. 7. Axial velocity and velocity magnitude at coil mid-length: peak displaced toward the outer wall.

C. Air-Side Flow and Temperature

The air-side solutions (Figs. 8–9) show ambient air entering the louvres, crossing the porous pad at ≈ 2.8 m/s face velocity and converging into the fan outlet. Air crossing the pad is cooled from 42 to $\approx 25^\circ\text{C}$ in summer, 33 to $\approx 28.5^\circ\text{C}$ in monsoon and 24 to $\approx 17^\circ\text{C}$ in winter, and the cooled core surrounding the coil bank is spatially uniform to within $\approx 1^\circ\text{C}$ in every season — supporting the water-side assumption of a single external sink temperature per season.

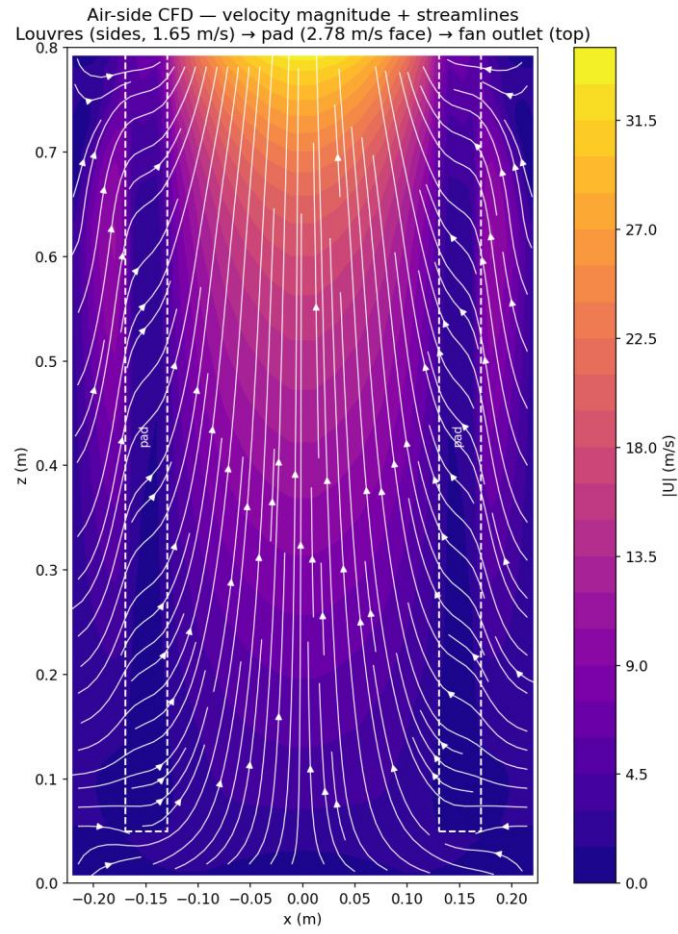


Fig. 8. Air velocity magnitude and streamlines, vertical mid-plane (identical hydrodynamics in all seasons).

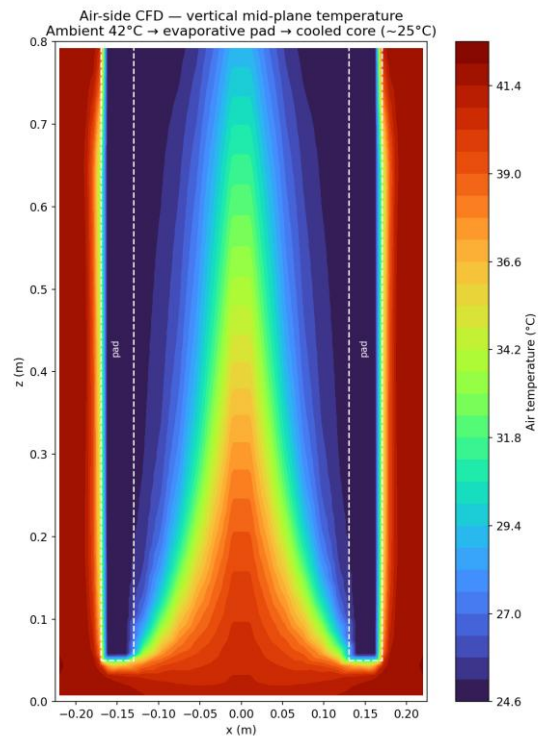


Fig. 9a. Air temperature, summer (42 → ≈25 °C).

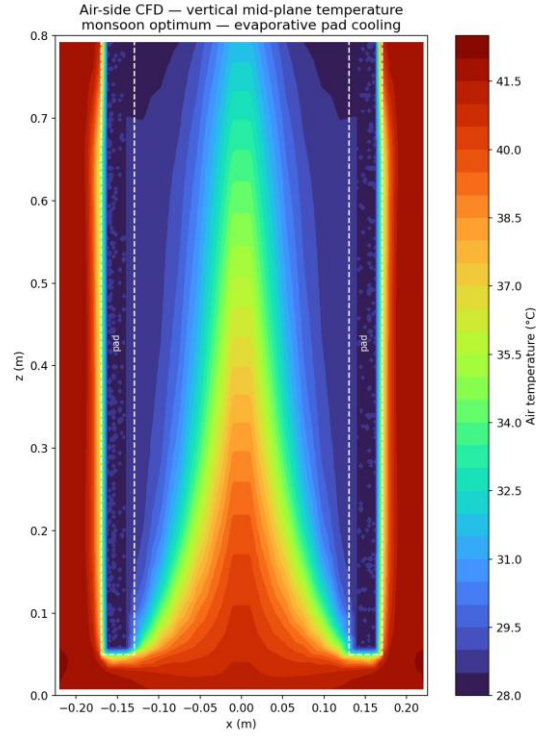


Fig. 9b. Air temperature, monsoon ($33 \rightarrow \approx 28.5$ °C).

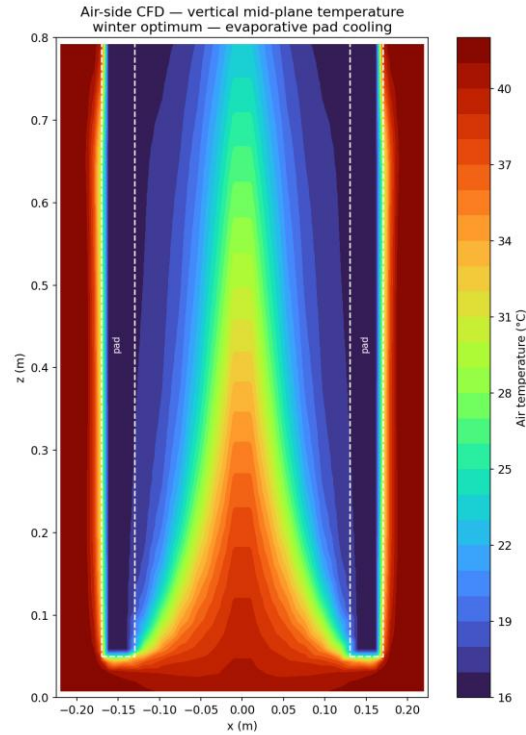


Fig. 9c. Air temperature, winter ($24 \rightarrow \approx 17$ °C).

5. SEASON-WISE VALIDATION AND DISCUSSION

A. Comparison with Experiment

Table II and Fig. 10 compare the CFD with the seasonal experimental optima. Under the restrictive calibration policy of Section 3.D — one calibrated point (summer), one a priori estimate (monsoon), one blind prediction (winter) — the model reproduces the seasonal temperature drops with deviations of -0.5% , -1.6% and -1.2%

respectively, all under-predictions of consistent sign, and matches the seasonal cooling effectiveness within 2 percentage points. The winter case deserves emphasis: no parameter was adjusted for it, yet the summer-calibrated film coefficient predicts its temperature drop to 1.2% — evidence that the calibrated h_{eff} captures a genuine physical conductance of the wetted-pad assembly at 6 L/hr rather than a case-specific fitting constant.

TABLE II. Season-Wise Validation Summary

Quantity	Summer	Monsoon	Winter
h_{eff} (W/m ² ·K) and its status	41 (calibrated)	28 (estimated a priori)	41 (reused — blind prediction)
Experimental $T_{\text{out}} / \Delta T$ (°C)	24.3 / 9.7	29.0 / 5.4	17.2 / 5.6
CFD $T_{\text{out}} / \Delta T$ (°C)	24.35 / 9.65	29.09 / 5.31	17.27 / 5.54
Deviation in ΔT	−0.5%	−1.6%	−1.2%
Cooling effectiveness η_c : exp / CFD (%)	82 / 81.7	72 / 69.9	76 / 74.8
Cooling duty Q (W)	313	103	179

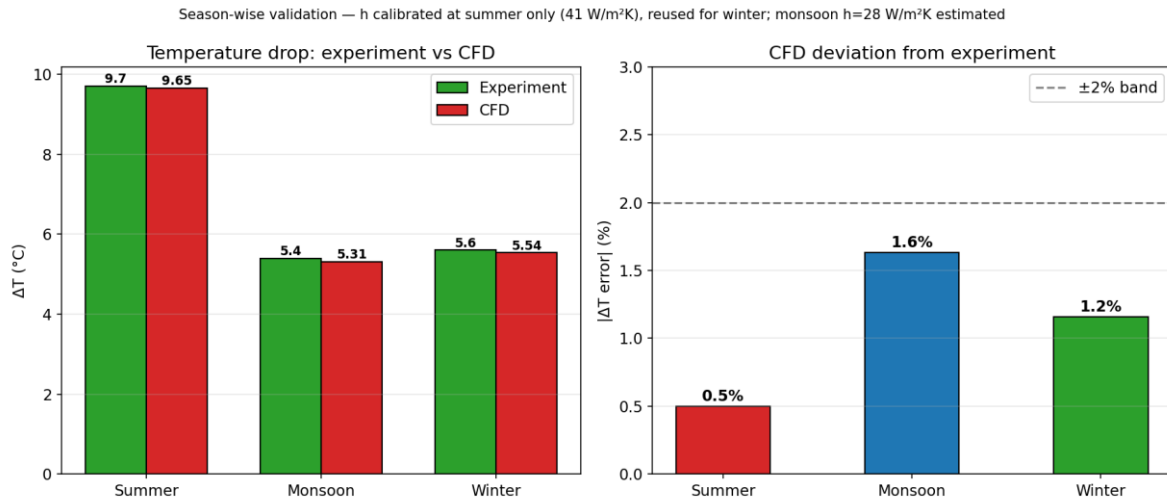


Fig. 10. Season-wise validation: measured vs CFD temperature drop (left) and CFD deviation (right).

B. Convergence Quality

Figure 11 documents the convergence of the summer base case: residuals fall monotonically after the startup transient and the outlet-temperature monitor is flat over the final $\approx 1,000$ iterations. Seasonal restarts inherit the converged hydrodynamics and reconverge thermally within a few hundred iterations. Combined with exact outlet mass-flow enforcement, $<0.15\%$ mass imbalance and energy-balance closure, all solutions are judged iteratively converged. A formal grid-convergence-index study across three mesh levels, following the standard procedure for reporting discretisation uncertainty [20], remains for future work; the reported quantities are constrained primarily by the external film conductance rather than internal discretisation, so limited grid sensitivity is expected.

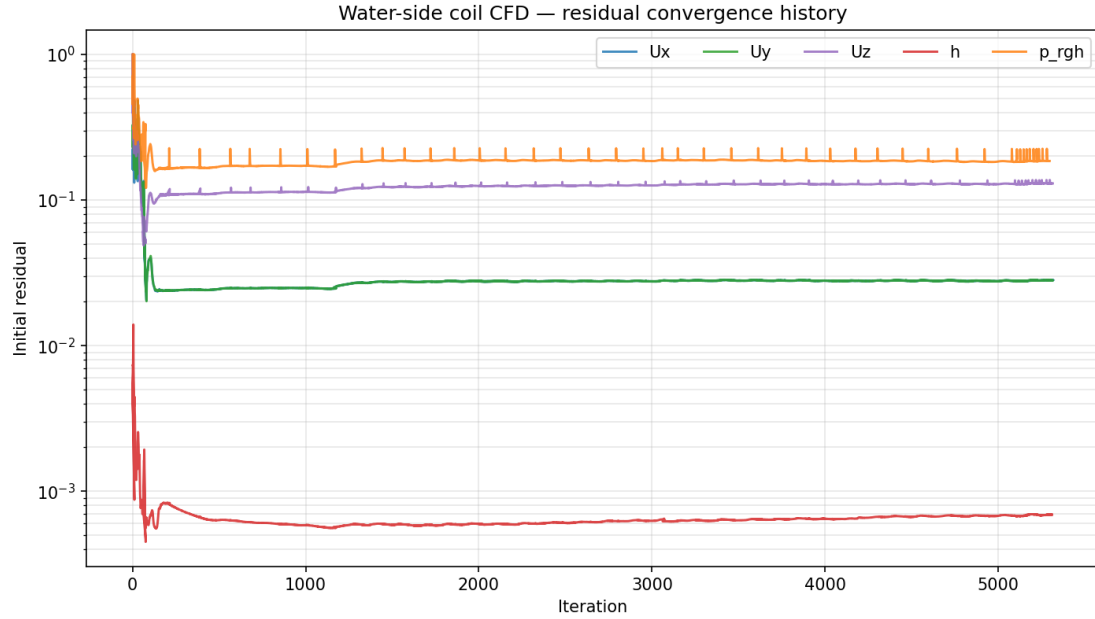


Fig. 11a. Residual history, summer base case.

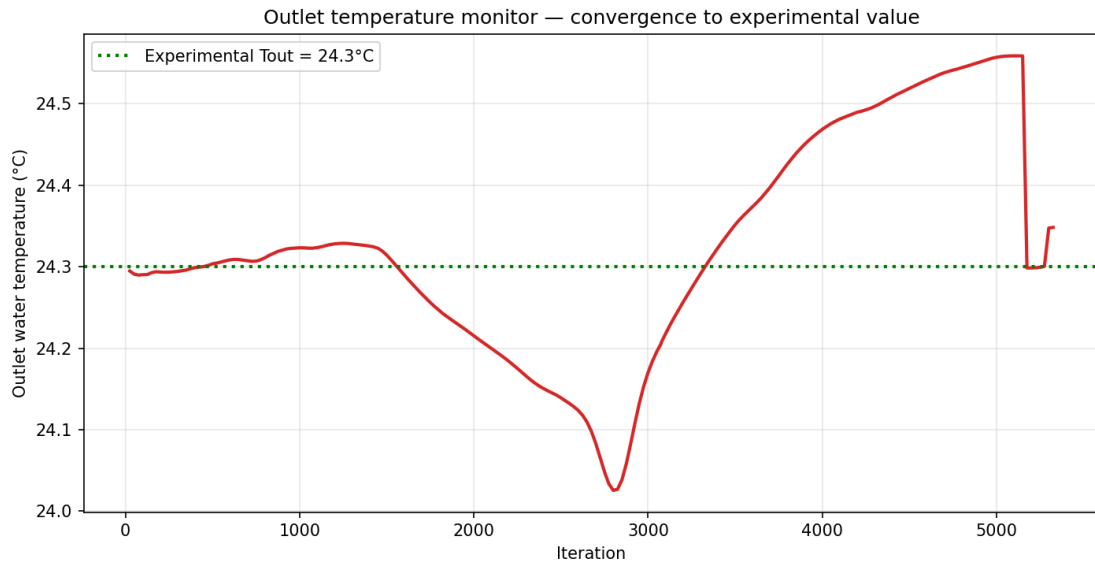


Fig. 11b. Outlet-temperature monitor, summer base case.

C. Thermal-Resistance Partition: The Effect of Pad Wetting

The calibration carries physical information. The dry-pad analytical model developed in the authors' experimental work — pad conduction in series with Churchill–Bernstein air-side convection — gives $UA \approx 21.5$ W/K. The CFD-calibrated conductance at the summer optimum is $UA \approx 40$ W/K, a factor of ≈ 1.9 higher. The difference quantifies the mechanism the dry-pad model omits: at 6 L/hr the sprinkled film flows over the pad fibres and the coil surface itself, so evaporation acts in parallel with pad conduction rather than in series behind it, roughly halving the external resistance. The monsoon value (28 W/m²·K) reflects the thinner wetted film at 5 L/hr and the reduced evaporation driving force in humid air. These trends support the experimental finding that sprinkled water flow rate is a statistically significant factor interacting with pad material: the flow rate controls the wetted fraction and hence the strength of the parallel evaporation path.

D. Limitations

The model has five principal limitations. (i) The film coefficient, although calibrated at only one point and validated by blind prediction in winter, is not derived ab initio; a mass-transfer (Lewis-analogy) evaporation submodel [21] would remove the remaining empiricism. (ii) The pad is represented without local film dynamics,

droplet transport or dry-patch formation. (iii) The water-side flow is treated as laminar at $Re = 1,650\text{--}2,750$; a transitional model could modify the internal coefficient, though its influence is bounded by the $\approx 10\%$ internal-resistance share. (iv) Representative seasonal ambient conditions were reconstructed from the reported ranges and psychrometric identities of the experimental campaign, as the exact ambient of each optimal trial was not recorded per-run. (v) A formal three-mesh grid-independence study and transient (diurnal) simulation remain open. The helix winding diameter and pitch were inferred from the physical unit and affect the Dean number but negligibly affect the calibrated thermal results.

6. CONCLUSIONS

A complete three-dimensional CFD model of a natural evaporative water cooler was developed in OpenFOAM and validated season-wise against the Taguchi-optimal experiments, with the following outcomes.

(i) A purpose-built butterfly O-grid sweep generator meshed the entire 76.2 m, 80.85-turn helical coil (aspect ratio $\approx 17,000:1$) with 155,200 hexahedral cells passing all quality checks — to the authors' knowledge the first full-coil 3D simulation of this device class.

(ii) Under a deliberately restrictive calibration policy — film coefficient calibrated once at the summer optimum ($41\text{ W/m}^2\cdot\text{K}$), reused unchanged for winter, and estimated a priori for monsoon ($28\text{ W/m}^2\cdot\text{K}$) — the model reproduces the seasonal temperature drops of 9.7, 5.4 and $5.6\text{ }^\circ\text{C}$ with deviations of -0.5% , -1.6% and -1.2% , and seasonal effectiveness within 2 percentage points.

(iii) The winter result, obtained with zero re-tuning, demonstrates that the calibrated conductance represents a genuine physical property of the wetted-pad assembly at a given sprinkle rate rather than a fitting constant.

(iv) The calibrated wetted-pad conductance ($\approx 40\text{ W/K}$) is $1.9\times$ the dry-pad analytical value, quantifying the parallel film-evaporation path and rationalising the experimentally observed significance of sprinkled-water flow rate and its interaction with pad material.

(v) The resolved Dean vortices ($De \approx 200\text{--}340$) displace the axial-velocity peak outward and homogenise the cross-section to $<0.2\text{ K}$; the $\approx 10\%$ internal-resistance share explains the weak experimental effect of coil water velocity in summer.

(vi) The seasonal air-side porous-media model confirms a near-uniform cooled-air envelope around the coil bank in every season, validating the decoupled strategy.

Future work will add a Lewis-analogy evaporation submodel for the pad film, a formal grid-convergence-index study, transitional turbulence modelling and transient simulation over diurnal and seasonal ambient cycles.

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The experimental data, device specification and analytical baseline used in this work are drawn from the authors' earlier experimental campaign on the same device.

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AUTHOR CONTRIBUTION

Milan D. Trivedi: conceptualisation, methodology, CFD model development, simulation, validation, and writing — original draft. Diptesh R. Patel: supervision, resources, and writing — review and editing. Both authors have read and approved the final version of the manuscript.

CONFLICT OF INTEREST

The authors declare no conflict of interest.

DATA AVAILABILITY STATEMENT

The data supporting the findings of this study are available from the corresponding author upon reasonable request.

References

1. J. R. Camargo, C. D. Ebinuma, and J. L. Silveira, "Experimental performance of a direct evaporative cooler operating during summer in a Brazilian city," *International Journal of Refrigeration*, vol. 28, no. 7, pp. 1124–1132, Nov. 2005, doi: 10.1016/j.ijrefrig.2004.12.011.
2. Y. Yang, G. Cui, and C. Q. Lan, "Developments in evaporative cooling and enhanced evaporative cooling - A review," Oct. 01, 2019, *Elsevier Ltd.* doi: 10.1016/j.rser.2019.06.037.
3. Y. M. Xuan, F. Xiao, X. F. Niu, X. Huang, and S. W. Wang, "Research and applications of evaporative cooling in China: A review (II) - Systems and equipment," Jun. 2012. doi: 10.1016/j.rser.2012.02.030.
4. A. Sohani and H. Sayyaadi, "Design and retrofit optimization of the cellulose evaporative cooling pad systems at diverse climatic conditions," *Appl. Therm. Eng.*, vol. 123, pp. 1396–1418, 2017, doi: 10.1016/j.applthermaleng.2017.05.120.
5. B. Rianguilaikul and S. Kumar, "An experimental study of a novel dew point evaporative cooling system," *Energy Build.*, vol. 42, no. 5, pp. 637–644, May 2010, doi: 10.1016/j.enbuild.2009.10.034.
6. "Dean1927".
7. W. R. Dean, "LXXII. The stream-line motion of fluid in a curved pipe (Second paper)," *The London, Edinburgh, and Dublin Philosophical Magazine and Journal of Science*, vol. 5, no. 30, pp. 673–695, Apr. 1928, doi: 10.1080/14786440408564513.
8. A. Kalpakli Vester, R. Orlu, and P. H. Alfredsson, "Turbulent flows in curved pipes: Recent advances in experiments and simulations," *Appl. Mech. Rev.*, vol. 68, no. 5, Sep. 2016, doi: 10.1115/1.4034135.
9. B. K. Hardik, P. K. Baburajan, and S. V. Prabhu, "Local heat transfer coefficient in helical coils with single phase flow," *Int. J. Heat Mass Transf.*, vol. 89, pp. 522–538, Jun. 2015, doi: 10.1016/j.ijheatmasstransfer.2015.05.069.
10. P. Naphon and S. Wongwises, "A review of flow and heat transfer characteristics in curved tubes," 2006, *Elsevier Ltd.* doi: 10.1016/j.rser.2004.09.014.
11. A. Franco, D. L. Valera, A. Peña, and A. M. Pérez, "Aerodynamic analysis and CFD simulation of several cellulose evaporative cooling pads used in Mediterranean greenhouses," *Comput. Electron. Agric.*, vol. 76, no. 2, pp. 218–230, May 2011, doi: 10.1016/j.compag.2011.01.019.
12. K. Vafai and C. L. Tien, "BOUNDARY AND INERTIA EFFECTS ON FLOW AND HEAT TRANSFER IN POROUS MEDIA," Pergamon Press Ltd, 1981.
13. J. M. Wu, X. Huang, and H. Zhang, "Numerical investigation on the heat and mass transfer in a direct evaporative cooler," *Appl. Therm. Eng.*, vol. 29, no. 1, pp. 195–201, Jan. 2009, doi: 10.1016/j.applthermaleng.2008.02.018.
14. A. Malli, H. R. Seyf, M. Layeghi, S. Sharifian, and H. Behraves, "Investigating the performance of cellulosic evaporative cooling pads," *Energy Convers. Manag.*, vol. 52, no. 7, pp. 2598–2603, 2011, doi: 10.1016/j.enconman.2010.12.015.
15. A. Thess, "Fluid Mechanics and Its Applications Volume 113 Series editor." [Online]. Available: <http://www.springer.com/series/5980>
16. M. Ghobadi and Y. S. Muzychka, "A Review of Heat Transfer and Pressure Drop Correlations for Laminar Flow in Curved Circular Ducts," Jul. 02, 2016, *Taylor and Francis Ltd.* doi: 10.1080/01457632.2015.1089735.
17. J. C. Kurnia, A. P. Sasmito, and A. S. Mujumdar, "Numerical investigation of laminar heat transfer performance of various cooling channel designs," *Appl. Therm. Eng.*, vol. 31, no. 6–7, pp. 1293–1304, May 2011, doi: 10.1016/j.applthermaleng.2010.12.036.
18. L. Rong, P. Pedersen, T. L. Jensen, S. Morsing, and G. Zhang, "Dynamic performance of an evaporative cooling pad investigated in a wind tunnel for application in hot and arid climate," *Biosyst. Eng.*, vol. 156, pp. 173–182, Apr. 2017, doi: 10.1016/j.biosystemseng.2017.02.003.
19. B. E. Launder and D. B. Spalding, "THE NUMERICAL COMPUTATION OF TURBULENT FLOWS," in *Numerical Prediction of Flow, Heat Transfer, Turbulence and Combustion*, Elsevier, 1983, pp. 96–116. doi: 10.1016/B978-0-08-030937-8.50016-7.
20. L. Eça and M. Hoekstra, "A procedure for the estimation of the numerical uncertainty of CFD calculations based on grid refinement studies," *J. Comput. Phys.*, vol. 262, pp. 104–130, Apr. 2014, doi: 10.1016/j.jcp.2014.01.006.
21. T. Sun, X. Huang, Y. Qu, F. Wang, and Y. Chen, "Theoretical and experimental study on heat and mass transfer of a porous ceramic tube type indirect evaporative cooler," *Appl. Therm. Eng.*, vol. 173, Jun. 2020, doi: 10.1016/j.applthermaleng.2020.115211.

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