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Comparative CFD Modelling of Weather Station Protective Screens

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Abstract

Meteorological temperature sensors are housed in protective screens that block solar radiation while allowing ventilation. The presence of the screen itself introduces a thermal bias in the measurement, whose magnitude depends on screen geometry, meteorological forcing, and sensor placement. Existing CFD studies address one configuration at a time, limiting the ability to compare designs on a common physical basis.

This work presents a conjugate heat transfer framework implemented in OpenFOAM v12 and applied to three screen configurations: a double-door Stevenson screen (S2), a single-door Stevenson screen (S1), and a compact multi-plate shield (MP). Validated against published numerical data to within 7.4%, the framework is used to assess the sensitivity of the bias to meteorological forcing and geometric parameters, and to compare the three designs under identical conditions. Repositioning the sensor toward the lower quarter of the enclosure and selecting a low-conductivity material such as cedar are identified as the most effective levers for bias reduction. The S2 screen outperforms the other two designs under all conditions investigated, with the MP shield performing worst under high solar irradiance.

Keywords: Meteorology, Air temperature measurement, Conjugate Heat Transfer, CFD, OpenFOAM, meteorological screen, radiation shield, thermal bias, Stevenson screen, RANS

1. Introduction

1.1. Context and motivation

Air temperature is one of the most critical variables in meteorology and climatology, with applications ranging from weather forecasting and aviation safety to climate change monitoring. The World Meteorological Organization classifies it as an essential climate variable and sets a required measurement uncertainty of 0.1 K under standard conditions [11]. Achieving this in field conditions is, however, far from straightforward.

Any physical sensor placed in the atmosphere is not a passive observer: it exchanges heat with its environment through convection, conduction, and radiation, and the temperature it records may differ substantially from the true air temperature. This difference has multiple origins — sensor calibration and self-heating, the presence of trees,

roads or buildings in the nearby environment, the type of ground cover in the siting area — but the dominant sources of error under typical conditions are radiative heating of the sensor by direct solar irradiance and the wet-bulb effect induced by precipitation on the sensor surface.

To mitigate these perturbations, protective screens have been in operational use since 1864. These enclosures block direct solar irradiance and precipitation while allowing passive ventilation, and are today the standard for housing meteorological temperature sensors. However, the introduction of a screen into the measurement system introduces its own measurement artefact: it reduces airflow over the sensor and creates an internal microclimate that departs from ambient atmospheric conditions, introducing what is commonly referred to as the *screen-induced thermal bias*.

Today, a wide variety of screen designs are simultaneously in operational use across the world’s meteorological networks, with no universally accepted reference standard [11, 10]. This diversity has direct consequences for the comparability of climate records: Lin et al. [13] documented that the replacement of one screen type by another in the US National Weather Service network introduced a mean offset of approximately 0.4°C in daily maximum temperatures. Such discontinuities, if undetected, can corrupt climate signals — a critical concern given that most climate studies aim to detect temperature trends on the order of tenths of a degree per decade.

1.2. Literature review and research gap

Three main approaches have been developed to study and quantify the screen-induced bias: field intercomparison campaigns, empirical correction models, and numerical modelling [2].

1.2.1. Field intercomparison campaigns

Field campaigns consistently identify solar irradiance and wind speed as the two dominant governing factors — the bias increases sharply with irradiance and decreases with wind speed, a finding that is robust across all climatic settings studied to date [13, 7, 15, 9]. Lin et al. [13] compared four shield types in Nebraska and reported daytime biases up to 2.3°C at low wind speeds, and showed that the airflow velocity inside a shield is two to four times weaker than the ambient wind. Ereil et al. [7] evaluated sixteen designs in Israel, observed a maximum bias of 2.5°C , and showed that the ranking of screen performance is not universal — the relative advantage of one design over another depends on the prevailing climate conditions, a conclusion that complicates the transfer of results from one geographic context to another [7, 12]. The COAT campaign conducted a 14-month intercomparison of ten shield models in the Arctic, recording a maximum divergence of 1.29°C between shields at wind speeds below 2 m s^{-1} , and quantified for the first time the full uncertainty budget associated with such field comparisons [9].

1.2.2. Empirical correction models

Various authors have proposed transfer functions relating the screen-induced bias to measurable meteorological variables such as wind speed, solar irradiance, and solar

elevation angle [15, 5]. These approaches can effectively reduce the bias for a known screen in a given environment, but they are inherently site- and shield-specific: coefficients calibrated on one dataset cannot be reliably transferred to a different screen design or climate, and crucially, they provide no spatial information on the internal flow and temperature fields that drive the bias.

1.2.3. Numerical modelling

CFD-based approaches address these limitations by resolving the three-dimensional physics inside the screen. Several studies have demonstrated their viability for quantifying and correcting radiation errors, using coupled fluid–solid simulations to assess the sensitivity of the thermal bias to wind speed, solar angle, and surface properties [17, 14]. Applied to the RMI double-door Stevenson screen, a conjugate heat transfer simulation using a k – ε turbulence model computed a bias of 1.13°C under 800 W m^{-2} solar irradiance and 1 m s^{-1} wind — consistent with field observations — and revealed internal flow structures that are inaccessible to experimental measurement, such as the low-velocity zone created by the door frame around the sensor [6].

Despite these contributions, an important gap remains. Most existing CFD studies address one screen configuration at a time, with identical physical assumptions and boundary conditions never applied simultaneously across multiple configurations. Without such a common framework, it is impossible to isolate the influence of specific geometric parameters—louvre pitch, ventilation area, door configuration—on the thermal bias, compare designs from different families in a physically consistent manner, or identify which geometric modifications would most effectively reduce measurement error. Furthermore, all detailed CHT simulations of meteorological screens to date have relied on commercial software, limiting reproducibility and accessibility.

1.3. Objectives

This work applies a consistent CHT framework, implemented in the open-source solver OpenFOAM v12, to multiple screen configurations under identical physical assumptions and boundary conditions. The specific objectives are to validate the framework against published numerical data, to use it to predict the thermal bias ΔT over a range of meteorological variables. The sensitivity of the bias to geometric parameters — louvre inclination, inter-louvre spacing, screen size, and material — is systematically examined to identify the design factors that field intercomparisons alone cannot isolate, and to derive practical recommendations for reducing measurement error. The three screen configurations are finally compared under identical conditions, providing a physically consistent basis for assessing the relative performance of each design.

2. Screens under Investigation

Three screen designs are investigated. Their key parameters are summarized in Table 1.

Double-door Stevenson screen (S2). The reference design of the Royal Meteorological Institute (RMI) of Belgium [6]. A naturally ventilated wooden enclosure of

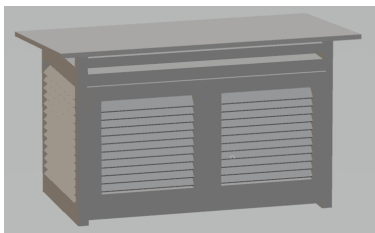
approximately $1060 \text{ mm} \times 580 \text{ mm} \times 740 \text{ mm}$ with two symmetrical louvred doors, a double roof, and a slatted floor.

Single-door Stevenson screen (S1). Similar to S2 except that it only has one door, and an internal volume approximately divided by two. This configuration, besides being commonly used and therefore interesting to model, isolates the effect of dimensional changes.

Compact multi-plate shield (MP). A naturally ventilated stacked-plate design representative of the Gill-type shields widely deployed in automatic weather station networks. They are typically much smaller.

Table 1: Key parameters of the three screens under investigation.

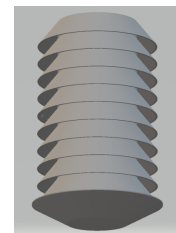
Parameter	S2	S1	MP
Dimensions (mm)	$1060 \times 580 \times 740$	$510 \times 560 \times 740$	$\varnothing 120 \times 200$
Material	Oak wood	Oak wood	Plastic
Ventilation	Louvres (N)	Louvres (N)	Plates (N)



(a) S2



(b) S1



(c) MP

Figure 1: CAD models of the three screen configurations.

3. Methodology

3.1. Overview

The simulation framework is based on steady-state, three-dimensional CHT (conjugate heat transfer), solving the turbulent momentum and energy equations in the fluid region and the heat conduction equation in each solid region simultaneously across one fluid region (air) and several solid regions (screen body, roof, legs, and sensor) using OpenFOAM's `foamMultiRun` framework. Thermal radiation is handled outside the CFD solver through an analytical model that prescribes net radiative heat fluxes as surface boundary conditions, as described in Section 3.5. OpenFOAM v12 is selected for its native multi-region CHT support and its open-source nature.

3.2. Governing equations

The airflow is governed by the incompressible steady-state RANS equations:

$$\nabla \cdot \mathbf{U} = 0 \quad (1)$$

$$\rho (\mathbf{U} \cdot \nabla \mathbf{U}) = -\nabla p + \nabla \cdot [(\mu + \mu_t) (\nabla \mathbf{U} + (\nabla \mathbf{U})^\top)] + \mathbf{f}_b \quad (2)$$

$$\rho c_p (\mathbf{U} \cdot \nabla T) = \nabla \cdot \left[\left(\lambda + \frac{\mu_t c_p}{Pr_t} \right) \nabla T \right] \quad (3)$$

where $\mathbf{U}$ is the mean velocity, p the pressure, μ the dynamic viscosity, μ_t the turbulent viscosity, T the temperature, λ the thermal conductivity, c_p the specific heat capacity, and Pr_t the turbulent Prandtl number. All fluid properties are evaluated at the reference state and summarised in Table 2.

Table 2: Thermophysical properties of air at the reference state.

Property	Symbol	Value	Unit
Reference temperature	T_0	293.15	K
Reference pressure	p_0	101 325	Pa
Density	ρ_0	1.225	kg m^{-3}
Dynamic viscosity	μ	1.81×10^{-5}	Pa s
Thermal conductivity	λ	2.57×10^{-2}	$\text{W m}^{-1} \text{K}^{-1}$
Specific heat	c_p	1005	$\text{J kg}^{-1} \text{K}^{-1}$
Prandtl number	Pr	0.713	—
Turbulent Prandtl number	Pr_t	0.85	—

Each solid region satisfies the steady-state heat conduction equation:

$$\nabla \cdot (\kappa_s \nabla T) + \dot{q} = 0 \quad (4)$$

where κ_s is the material thermal conductivity and $\dot{q}$ the externally prescribed radiative heat source. The thermophysical properties of each solid region are given in Table 3.

Table 3: Thermophysical properties of the solid regions.

Region	Material	ρ (kg m^{-3})	κ_s ($\text{W m}^{-1} \text{K}^{-1}$)	c_p ($\text{J kg}^{-1} \text{K}^{-1}$)
Screen body	Oak wood	750	0.19	2390
Roof	Oak wood	750	0.19	2390
Legs	Steel	7800	50	500
Sensor	Stainless steel	7900	16	500

At every fluid–solid interface, temperature and heat flux continuity are enforced:

$$T_f = T_s, \quad \kappa_f \frac{\partial T}{\partial n} \Big|_f = \kappa_s \frac{\partial T}{\partial n} \Big|_s \quad (5)$$

implemented in OpenFOAM via the `coupledTemperature` boundary condition.

3.3. Buoyancy model

The buoyancy body force $\mathbf{f}_b$ appearing in Eq. (2) is modelled using the Boussinesq approximation, in which density variations are accounted for only in the buoyancy term and treated as linearly proportional to temperature deviations from the reference state:

$$\mathbf{f}_b = -\rho_0 \beta (T - T_0) \mathbf{g}, \quad \beta = \frac{1}{T_0} \quad (6)$$

where $\mathbf{g}$ is the gravitational acceleration vector. The approximation is valid when $\beta |\Delta T| \ll 1$, a condition comfortably satisfied here: with a maximum temperature deviation of the order of 7 K, one obtains $\beta \Delta T_{\max} \approx 0.024 \ll 1$.

3.4. Turbulence model

The flow around and through a meteorological screen involves a complex mix of phenomena: separated shear layers at the louvre gaps, recirculation zones inside the enclosure, and impingement on solid surfaces. A RANS approach with the standard k - ε model is adopted to close the momentum equations. This model has been successfully applied to similar screen geometries in the literature [6, 17], and offers a practical balance between physical fidelity and computational cost for a parametric study involving multiple configurations and operating conditions.

The k - ε model solves two additional transport equations for the turbulent kinetic energy k and its dissipation rate ε :

$$\nabla \cdot (\rho \mathbf{U} k) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \nabla k \right] + P_k - \rho \varepsilon \quad (7)$$

$$\nabla \cdot (\rho \mathbf{U} \varepsilon) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \nabla \varepsilon \right] + C_{\varepsilon 1} \frac{\varepsilon}{k} P_k - C_{\varepsilon 2} \rho \frac{\varepsilon^2}{k} \quad (8)$$

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}, \quad P_k = \mu_t \left(\nabla \mathbf{U} + (\nabla \mathbf{U})^\top \right) : \nabla \mathbf{U} \quad (9)$$

with standard constants $C_\mu = 0.09$, $C_{\varepsilon 1} = 1.44$, $C_{\varepsilon 2} = 1.92$, $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$.

3.5. Radiative heat flux model

OpenFOAM's native radiation models are designed for participating media and are not appropriate for surface-to-surface exchange in transparent air. Net radiative fluxes are therefore computed analytically outside the solver and prescribed as Neumann boundary conditions on the relevant solid faces. The total net flux on surface i is:

$$\dot{q}_i = \dot{q}_{i,\text{SW}} + \dot{q}_{i,\text{LW}} \quad (10)$$

where SW and LW denote the shortwave (solar) and longwave (thermal infrared) contributions respectively.

3.5.1. Shortwave (solar) flux

Three contributions are accounted for: direct beam, diffuse sky, and ground-reflected radiation:

$$\dot{q}_{i,\text{SW}} = \alpha_s [I_{\text{dir}} \max(\mathbf{n}_i \cdot \hat{\mathbf{s}}, 0) + I_{\text{diff}} F_{i,\text{sky}} + \rho_g G_{\text{GHI}} F_{i,\text{ground}}] \quad (11)$$

where I_{dir} is the Direct Normal Irradiance (DNI), $\hat{\mathbf{s}}$ the unit sun vector, $\mathbf{n}_i$ the outward unit normal of surface i , I_{diff} the diffuse horizontal irradiance, $G_{\text{GHI}} = I_{\text{dir}} \cos \theta_z + I_{\text{diff}}$ the Global Horizontal Irradiance (GHI) with θ_z the solar zenith angle, and $F_{i,\text{sky}}$, $F_{i,\text{ground}}$ the view factors given in Table 6. Ground-reflected radiation is treated as isotropic, following the Lambertian assumption adopted by Erell et al. [7], whereby the reflected flux depends only on the view factor toward the ground and the total ground-leaving shortwave irradiance $\rho_g G_{\text{GHI}}$. The shortwave parameters used are summarised in Table 4.

The diffuse horizontal irradiance I_{diff} is estimated from G_{GHI} using the Erbs et al. correlation [8], which relates the diffuse fraction to the clearness index $k_t = G_{\text{GHI}}/I_0$, where $I_0 = I_{\text{sc}} \cos \theta_z$ is the extraterrestrial irradiance and $I_{\text{sc}} = 1361 \text{ W m}^{-2}$ the solar constant:

$$\frac{I_{\text{diff}}}{G_{\text{GHI}}} = \begin{cases} 1.0 - 0.09 k_t & k_t \leq 0.22 \\ 0.9511 - 0.1604 k_t + 4.388 k_t^2 - 16.638 k_t^3 + 12.336 k_t^4 & 0.22 < k_t \leq 0.80 \\ 0.165 & k_t > 0.80 \end{cases} \quad (12)$$

Since I_{diff} appears on both sides through G_{GHI} , Eq. (12) is solved iteratively until convergence, which is typically achieved in fewer than five iterations.

Table 4: Shortwave radiation parameters.

Parameter	Symbol	Value	Unit
Screen shortwave absorptivity	α_s	0.20	—
Ground albedo	ρ_g	0.23	—
Solar constant	I_{sc}	1361	W m^{-2}

3.5.2. Longwave (thermal infrared) flux

The net longwave exchange with sky and ground is:

$$\dot{q}_{i,\text{LW}} = \varepsilon_s [F_{i,\text{sky}} L_{\text{atm}}^{\downarrow} + F_{i,\text{ground}} L_{\text{ground}}^{\uparrow}] - \varepsilon_s \sigma T_s^4 \quad (13)$$

where ε_s is the screen longwave emissivity, σ the Stefan–Boltzmann constant, and T_s the screen surface temperature. The longwave parameters are summarised in Table 5.

Downwelling atmospheric irradiance. $L_{\text{atm}}^{\downarrow}$ is estimated using the Brunt correlation [3], which expresses the effective atmospheric emissivity as a function of the near-surface vapour pressure e_a (hPa):

$$L_{\text{atm}}^{\downarrow} = \sigma T_{\text{air}}^4 (a + b\sqrt{e_a}) \quad (14)$$

where $a = 0.52$ and $b = 0.065 \text{ hPa}^{-1/2}$ are empirical coefficients [3]. The vapour pressure e_a is derived from the relative humidity RH and the saturation vapour pressure via the Magnus formula [1]:

$$e_a = RH \cdot e_{\text{sat}}, \quad e_{\text{sat}} = 6.112 \exp\left(\frac{17.67 (T_{\text{air}} - 273.15)}{T_{\text{air}} - 29.65}\right) \quad (\text{hPa}) \quad (15)$$

Upwelling ground irradiance. $L_{\text{ground}}^{\uparrow}$ is obtained from the ground surface temperature T_g , assumed to emit as a grey body:

$$L_{\text{ground}}^{\uparrow} = \varepsilon_g \sigma T_g^4 \quad (16)$$

T_g is not prescribed but computed by solving the ground surface energy balance:

$$\alpha_g G_{\text{GHI}} + L_{\text{atm}}^{\downarrow} - \varepsilon_g \sigma T_g^4 - h_c (T_g - T_{\text{air}}) - G_{\text{soil}} - LE = 0 \quad (17)$$

G_{soil} (the soil heat conduction flux) and LE (the evapotranspiration latent heat flux) are prescribed as constant values typical of a grass-covered surface in clear-sky daytime conditions ($G_{\text{soil}} = 60 \text{ W m}^{-2}$, $LE = 180 \text{ W m}^{-2}$). Equation (17) is solved numerically for T_g .

Table 5: Longwave radiation parameters.

Parameter	Symbol	Value	Unit
Screen longwave emissivity	ε_s	0.96	—
Stefan–Boltzmann constant	σ	5.67×10^{-8}	$\text{W m}^{-2} \text{K}^{-4}$
Brunt coefficient a	a	0.52	—
Brunt coefficient b	b	0.065	$\text{hPa}^{-1/2}$
Ground shortwave absorptivity	α_g	0.77	—
Ground longwave emissivity	ε_g	0.986	—

3.5.3. View factors

The view factors $F_{i,\text{sky}}$ and $F_{i,\text{ground}}$ represent the fraction of the solid angle seen from each surface that is occupied by the sky and the ground respectively, assuming an unobstructed open-field installation — which is typically the case for meteorological stations. The same values are adopted for both shortwave and longwave exchanges, as they depend only on surface orientation and not on the spectral band. They are derived analytically from surface orientation, following the standard geometric relationships adopted by Erell et al. [7]

Table 6: View factors by surface orientation, common to both shortwave and longwave radiation.

Surface	F_{sky}	F_{ground}
Roof	1.0	0.0
Walls	0.5	0.5
Floor	0.0	1.0

3.6. Model assumptions

Some assumptions govern the physical formulation of the problem. First, steady-state conditions are assumed: since the characteristic thermal response time of a screen is of the order of tens of minutes, and the meteorological forcing varies on timescales of hours under the clear-sky daytime conditions of interest, the screen can be considered to be in quasi-equilibrium with its environment at any given instant, consistent with standard practice in screen performance studies [6, 17]. Second, the Boussinesq approximation is applied for buoyancy, as described in Section 3.3; the small temperature differences involved ensure that this approximation is well satisfied.

A third, structurally important assumption concerns the radiative model. Meteorological screens are specifically designed to prevent direct solar radiation from reaching the sensor: the louvres geometry and enclosure ensure that no optical path exists between the exterior and the sensor. This justifies treating the screen body as an opaque radiation barrier, so that the sensor receives no direct, diffuse, or ground-reflected short-wave flux. As a result, the radiation model reduces to prescribing net radiative fluxes on the screen surfaces only, without requiring complex internal view factors.

A number of additional simplifications are made: air thermophysical properties are evaluated at the reference state and held constant; solid materials are treated as homogeneous and isotropic; the incoming wind is modelled as a uniform horizontal flow; moisture transport and phase change are neglected; conductive heat transfer along sensor wires is not modelled (negligible, [13]), as well as Joule heating (small current pulses, [13]). These approximations are standard in CFD studies of meteorological screens and are not expected to materially affect the predicted thermal bias.

4. Implementation

4.1. Geometry generation

The geometry of the screens are constructed in OpenSCAD, an open-source parametric CAD tool. The parametric approach allows all screen dimensions to be driven by a small set of input variables, facilitating the variation of geometric parameters across configurations. When necessary, the resulting STL surfaces are cleaned and repaired for mesh defects using Blender. The use of exclusively open-source tools throughout the geometry pipeline is a deliberate choice, consistent with the reproducibility requirements of operational meteorological networks.

4.2. Computational domain and mesh

The computational domain extends 8 m in the lateral direction ($x \in [-4, 4]$ m), 25 m in the stream-wise direction ($y \in [-20, 5]$ m), and 8 m in the vertical direction ($z \in [0, 8]$ m), with the screen centered at the origin. The screen bottom is located at $z = 1.5$ m above the ground, placing the sensor at $z \approx 1.85$ m, consistent with the standard meteorological measurement height (typically between 1.25 and 2 m). The inlet is located 5 m upstream of the screen and the outlet 20 m downstream, providing sufficient fetch for the wake to develop without influencing the near-screen flow.

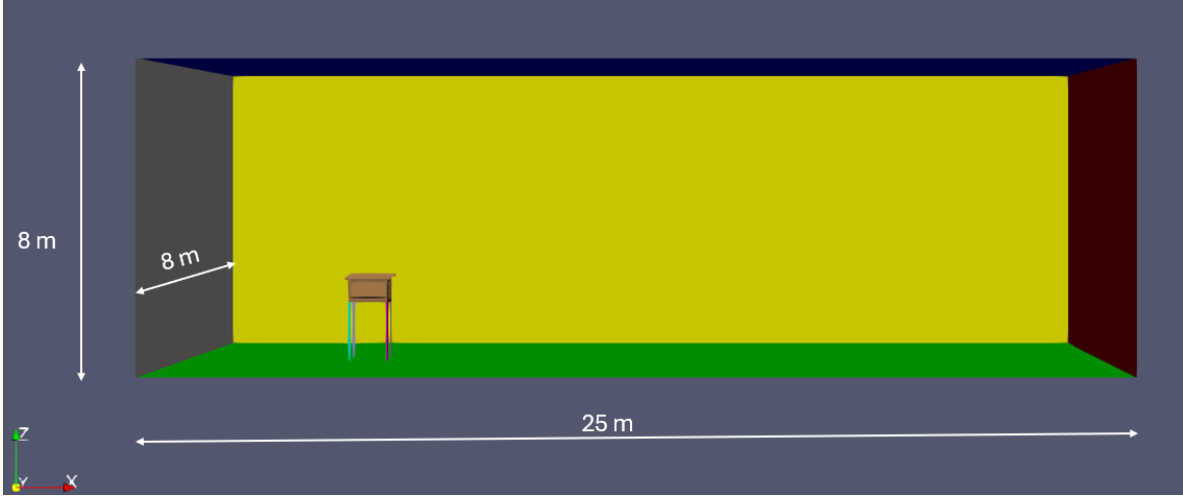


Figure 2: Computational domain and boundary conditions.

The mesh is generated in two stages. A structured background grid is first created with **blockMesh**, using an 18-block layout divided into three stream-wise zones: a front zone ($y \in [1, 5]$ m) refined toward the screen with a grading ratio of 4, a central zone ($y \in [-2, 1]$ m) kept uniform at the base resolution, and a back zone ($y \in [-20, -2]$ m) expanded toward the outlet with a grading ratio of 0.25. These grading ratios allow coarser elements far from the screen, where the resolution can be lowered, saving computational cost.

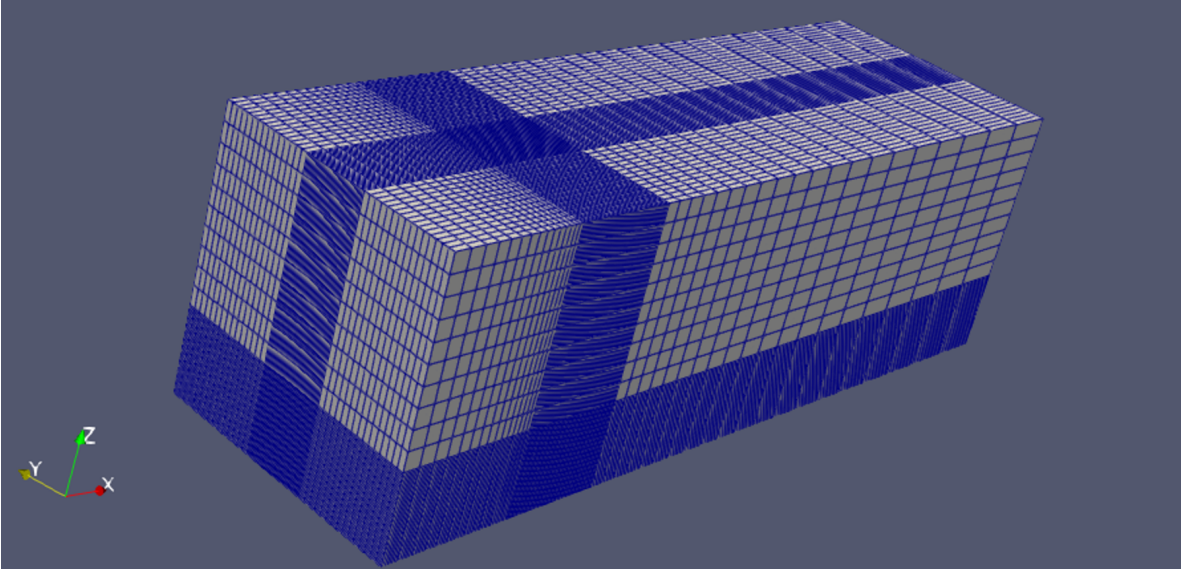


Figure 3: Background grid (`blockMesh`)

The screen geometry, roof, legs, and thermometer are then introduced via `snappyHexMesh`, which performs successive levels of refinement and snaps the mesh to the STL surfaces. Refinement levels of 4–5 are applied to the screen body, legs, and thermometer. Wall treatment is handled by standard wall functions (`kqRWallFunction` and `epsilonWallFunction`). The dimensionless wall distance y^+ was evaluated in post-processing using the `yPlus` function object, and found to lie predominantly in the range 30–100 across the screen surfaces at the reference wind speed of 1 m s^{-1} , consistent with the applicability range of the employed wall functions.

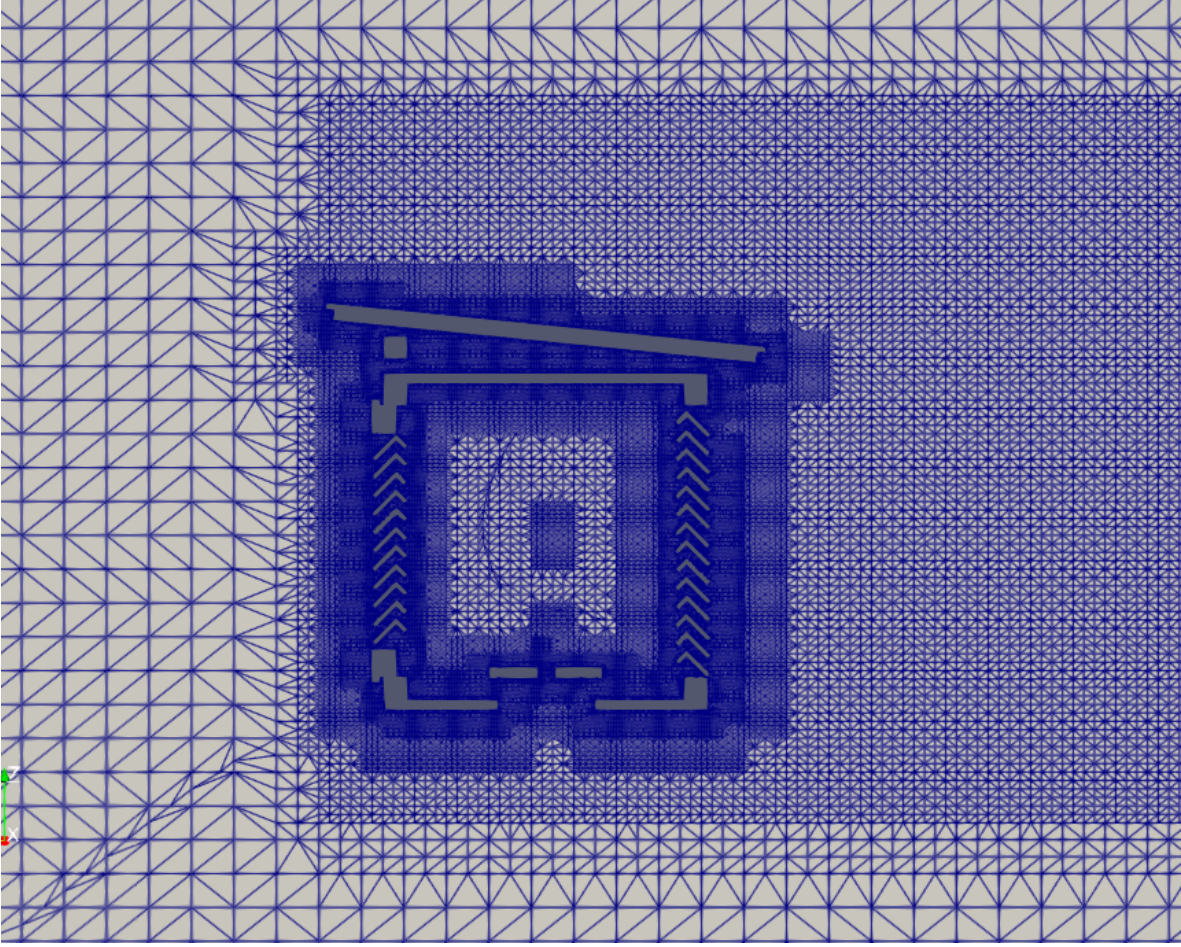


Figure 4: Refined grid (`snappyHexMesh`)

4.3. Radiative patch definition

The net radiative heat fluxes computed by the analytical model of Section 3.5 must be prescribed on specific surface patches of the screen solid region. However, the fluid–solid interface produced by `snappyHexMesh` is a single monolithic patch (`screenToFluid`), with no geometric subdivision. Splitting this interface into named sub-patches — each receiving its own radiative flux — therefore requires a dedicated post-processing step.

This is achieved through a two-stage workflow. First, `topoSetDict` is used to define face sets through a series of geometric constraints and set operations (`boxToFace`, `normalToFace`,...). Second, `createPatchDict` promotes each face set into a named `mappedWall` patch, coupled to its counterpart in the fluid region, on which the boundary condition `qs` is subsequently applied.

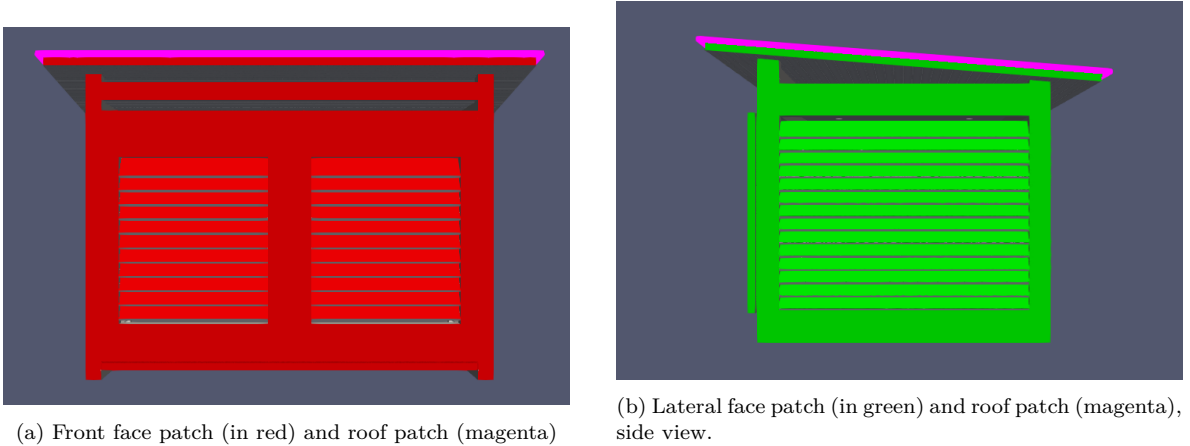


Figure 5: Radiative patches defined on the screen solid region. Each patch receives an independent net radiative heat flux computed from the analytical model.

4.4. Boundary conditions

The boundary conditions are summarized in Table 7. At the inlet ($y = +5$ m), a uniform velocity U_∞ directed in the $-y$ direction is prescribed. The turbulence quantities at the inlet are estimated from Richards and Hoxey [16] atmospheric boundary layer formulas for open terrain ($z_0 = 0.03$ m) at $z_{\text{ref}} = 2$ m:

$$k = \frac{u_*^2}{\sqrt{C_\mu}}, \quad \varepsilon = \frac{u_*^3}{\kappa z_{\text{ref}}}, \quad u_* = \frac{U_\infty \kappa}{\ln(z_{\text{ref}}/z_0)} \quad (18)$$

giving $k = 0.033 \text{ m}^2 \text{ s}^{-2}$ and $\varepsilon = 1.2 \times 10^{-4} \text{ m}^2 \text{ s}^{-3}$ at the reference wind speed $U_\infty = 1 \text{ m s}^{-1}$.

At the outlet ($y = -20$ m), pressure is fixed at the reference value ($p_0 = 101325$ Pa) and an `inletOutlet` condition is applied to all scalar fields to prevent reverse-flow instabilities. The ground is treated as a no-slip isothermal surface (large heat reservoir) at T_g ; the top boundary uses a slip condition for velocity and zero-gradient for temperature; the lateral boundaries are planes of symmetry. All fluid–solid interfaces enforce the `coupledTemperature` condition, ensuring temperature and heat flux continuity as described in Section 3. On external solid faces exposed to radiation, a net radiative heat flux is additionally prescribed as a Neumann condition (`qs`).

Table 7: Summary of boundary conditions.

Boundary	U	p	T	k	ε
Inlet	<code>fixedValue</code>	<code>zeroGradient</code>	<code>fixedValue</code>	<code>fixedValue</code>	<code>fixedValue</code>
Outlet	<code>inletOutlet</code>	<code>fixedValue</code>	<code>inletOutlet</code>	<code>inletOutlet</code>	<code>inletOutlet</code>
Ground	<code>noSlip</code>	<code>zeroGradient</code>	<code>fixedValue</code>	<code>wallFunction</code>	<code>wallFunction</code>
Top	<code>slip</code>	<code>zeroGradient</code>	<code>zeroGradient</code>	<code>slip</code>	<code>zeroGradient</code>
Sides	<code>symmetry</code>	<code>symmetry</code>	<code>symmetry</code>	<code>symmetry</code>	<code>symmetry</code>
Solid interfaces	<code>noSlip</code>	<code>zeroGradient</code>	<code>coupledTemperature</code>	<code>wallFunction</code>	<code>wallFunction</code>

4.5. Numerical schemes and solver settings

Convective terms are discretised with a first-order upwind scheme, which provides robust convergence for the recirculating flows expected inside the screen enclosure. Gradient and Laplacian terms use second-order central schemes (`Gauss linear` and `Gauss linear corrected` respectively), the latter accounting for mesh non-orthogonality. Pressure-velocity coupling is handled by the SIMPLE algorithm with the consistent formulation enabled and one non-orthogonal corrector per iteration. Convergence is declared when the normalised residuals of p , U , h , k , and ε fall below 10^{-4} , 10^{-4} , 10^{-5} , 10^{-4} , and 10^{-4} respectively. Under-relaxation factors of 0.7 are applied to p , U , and h , and 0.5 to k and ε .

4.6. Simulation workflow

For each combination of wind speed and solar irradiance, the simulation proceeds as follows. Radiative heat fluxes are first computed analytically from the model of Section 3.5 and prescribed as `qs` boundary conditions on the relevant solid patches. The multi-region CHT problem is then solved in parallel using `foamMultiRun`, which couples the fluid and solid solvers until convergence. The thermal bias ΔT is finally extracted as the difference between the volume-averaged temperature in the thermometer region and T_∞ .

5. Results and Discussion

5.1. Validation

The simulation framework is validated against the conjugate heat transfer results of Dobre et al. [6], who modelled the same S2 double-door Stevenson screen using the commercial solver COMSOL.

Figure 6 shows a qualitative comparison of the velocity fields obtained by both solvers on a vertical cross-section through the screen, at the same conditions ($U_\infty = 1 \text{ m s}^{-1}$ and $I = 800 \text{ W m}^{-2}$). Figure 7 presents the corresponding temperature fields. In both cases, the overall flow structure and thermal field inside the enclosure are in good agreement.

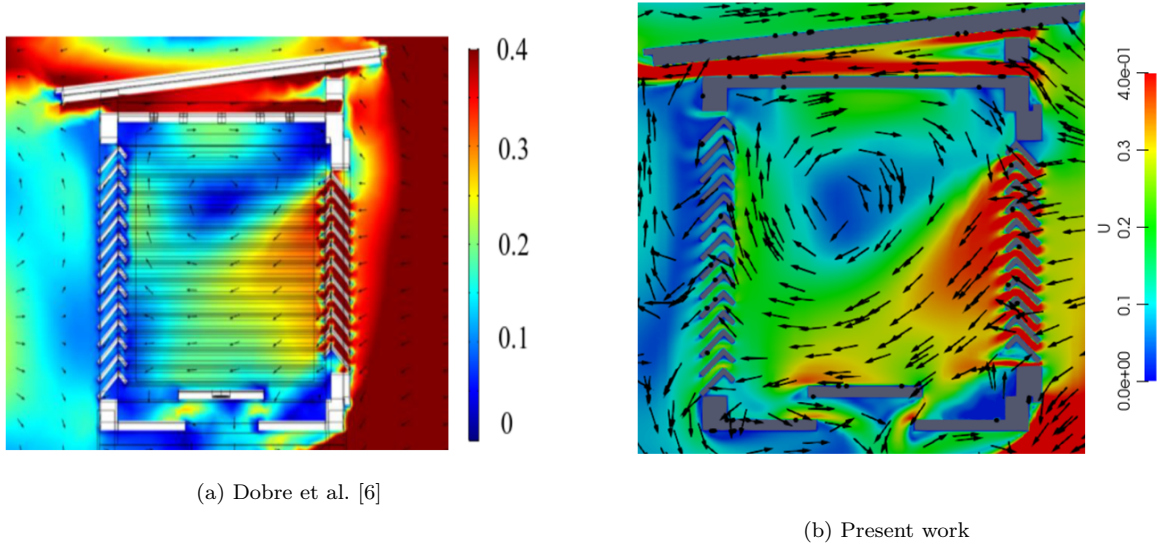


Figure 6: Comparison of the velocity fields on a vertical cross-section through the screen.

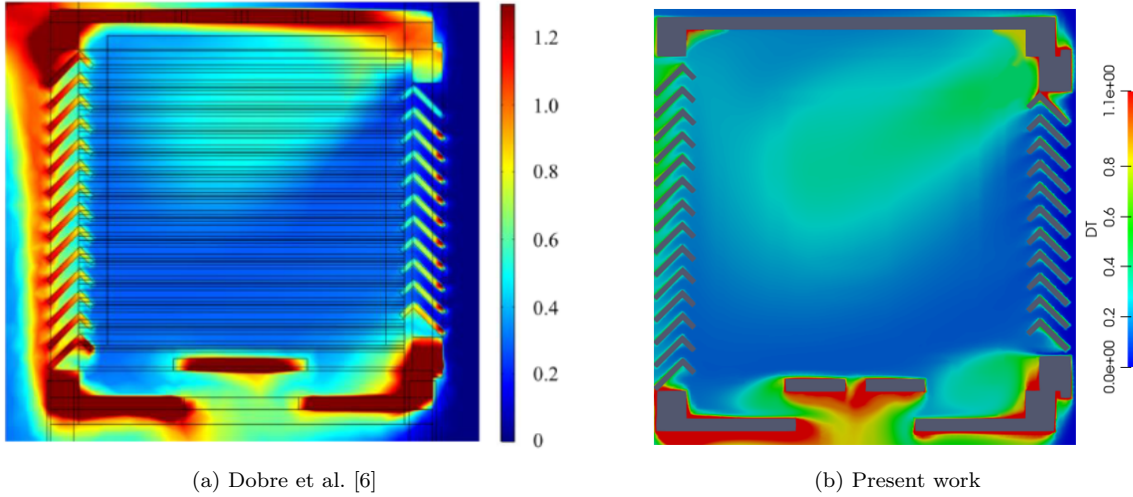
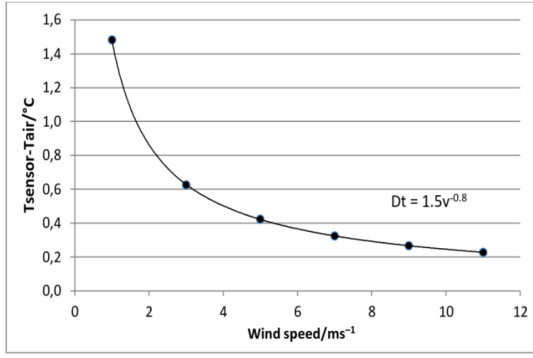
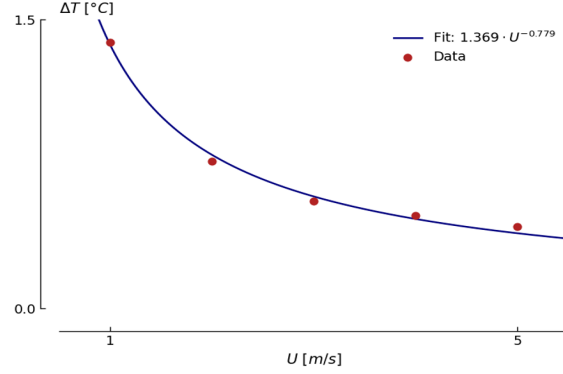


Figure 7: Comparison of temperature fields on a vertical cross-section through the screen.

A quantitative comparison of the thermal bias ΔT as a function of wind speed is shown in Figure 8. The power-law fits to the two datasets, $\Delta T = 1.369 U^{-0.779}$ (present work) and $\Delta T = 1.5 U^{-0.8}$ (Dobre et al. [6]), agree to within a mean relative error of 7.4%, validating the simulation framework for the configurations and conditions considered in this study.



(a) Dobre et al. [6]



(b) Present work

Figure 8: Thermal bias ΔT as a function of wind speed U_∞ at $I = 1000 \text{ W m}^{-2}$: comparison between Dobre et al. [6] and the present simulation.

5.2. Parametric study: influence of meteorological forcing

The validated framework is used to assess the sensitivity of the thermal bias of the S2 screen to several meteorological parameters: wind speed, solar irradiance, solar zenith angle, and wind direction. All simulations in this section use the S2 configuration.

5.2.1. Temperature bias changes with wind speed and irradiance

Figure 9 shows ΔT as a function of U_∞ and I for the present simulation and Yang et al. [17]. Both exhibit the same qualitative behaviour: ΔT scales approximately linearly with I and approximately as U^{-1} with wind speed, consistent with the non-dimensional scaling $X = I/(\rho c_p T U)$ proposed by Nakamura and Mahrt [15] and extended by Cheng et al. [5].

The absolute values differ by approximately one order of magnitude, as the screen of Yang et al. is based on a novel design specifically engineered to minimise thermal bias and indeed outperforms classical screens, as the authors themselves acknowledge [17]; modelling such a design falls outside the scope of the present work. The functional dependence on meteorological forcing is nevertheless in qualitative agreement across both studies.

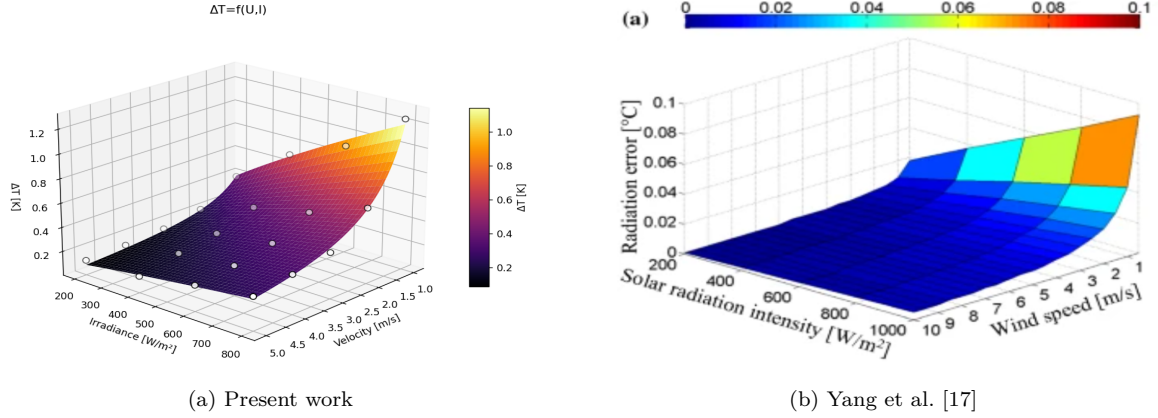


Figure 9: Thermal bias ΔT as a function of wind speed and irradiance for the present simulation and Yang et al. [17].

The worst conditions are, as expected, high solar irradiance combined with low wind speed: the former maximises the radiative heat load on the screen walls, while the latter limits the convective cooling of the screen and the renewal of air inside the enclosure.

5.2.2. Effect of wind direction

Meteorological screens are geometrically asymmetric: the internal depth available for recirculation differs between the two configurations. Two cases are investigated for the S2 screen at reference conditions: wind incident on the door face (reference case) and on the lateral face.

As shown in Figure 10, lateral wind produces a more spatially extended recirculation zone. The heated air is diluted across a larger internal volume, reducing the thermal bias by approximately 7% relative to the front-incidence case. The practical significance of this result is however limited. Standard installation practice orients the door northward to minimise solar exposure during manual readings, irrespective of prevailing wind direction. Furthermore, well-sited meteorological stations are deliberately located in open, unobstructed terrain, where wind direction is inherently variable rather than persistently unidirectional — the latter being characteristic of topographically constrained sites, such as valleys or coastal corridors, typically excluded for a weather station site.

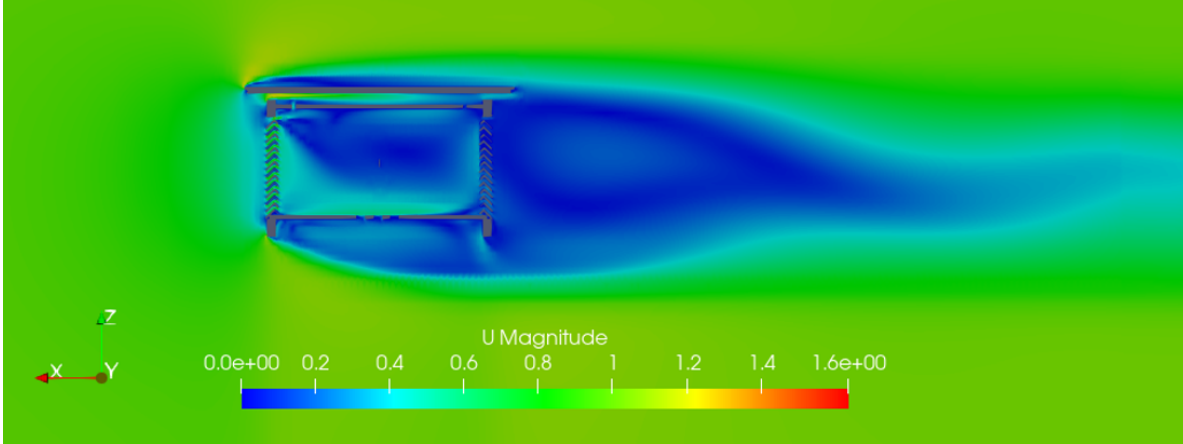


Figure 10: Velocity magnitude field (horizontal cross-section), wind incident on the lateral face at reference conditions ($U_\infty = 1 \text{ m s}^{-1}$, $I = 800 \text{ W m}^{-2}$).

5.2.3. Effect of solar zenith angle

Simulations presented so far prescribe a fixed solar zenith angle $\theta_z = 30^\circ$. To assess the sensitivity of the thermal bias to the sun position, θ_z is varied between 15° and 45° at fixed conditions ($U_\infty = 1 \text{ m s}^{-1}$, $I = 800 \text{ W m}^{-2}$) for the S2 configuration.

Figure 11 shows that ΔT reaches a maximum near $\theta_z \approx 25\text{--}30^\circ$ and decreases on either side. The overall variation across the range studied is modest — approximately 0.15°C — indicating that the screen-induced bias is less sensitive to sun position compared to the irradiance it produces. All subsequent simulations retain $\theta_z = 30^\circ$ as the reference condition.

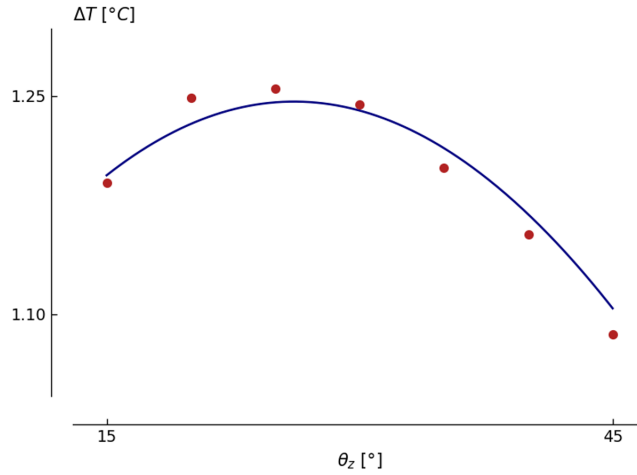


Figure 11: Thermal bias ΔT of the S2 screen as a function of solar zenith angle θ_z , at standard conditions.

5.3. Geometric sensitivity study

The framework is next applied to explore how specific geometric parameters of the S2 screen govern the thermal bias. All simulations in this section use the reference

meteorological conditions ($U_\infty = 1 \text{ m s}^{-1}$, $I = 800 \text{ W m}^{-2}$, $\theta_z = 30^\circ$) unless otherwise stated.

5.3.1. Louvre inclination angle

The louvre inclination angle θ is varied between 30° and 55° relative to the horizontal, with $\theta = 45^\circ$ corresponding to the standard RMI configuration. Figure 12 reveals a clear minimum near $\theta = 45^\circ$, confirming that the standard inclination is close to optimal for the reference conditions. The true optimum occurs at $\theta = 42^\circ$, yielding a reduction in thermal bias of 0.03°C relative to the standard configuration — a marginal gain that suggests the current design is already well-optimised with respect to louvre inclination.

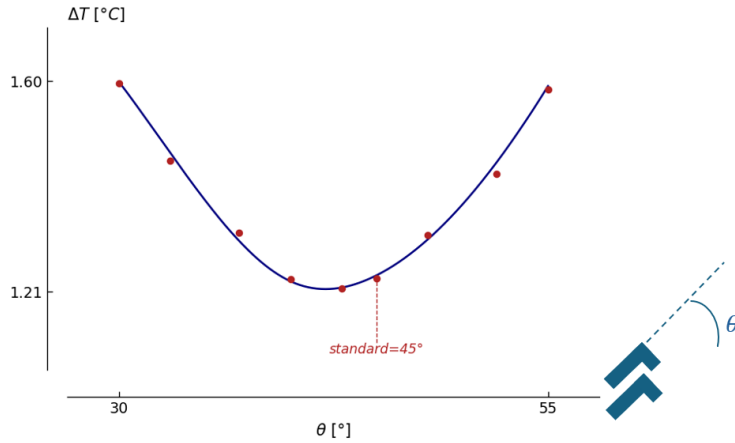


Figure 12: Thermal bias ΔT as a function of louvre inclination angle θ .

5.3.2. Inter-louvre spacing

The spacing d between adjacent louvres is varied from 20 mm to 45 mm, with the reference value $d = 32.5 \text{ mm}$. As shown in Figure 13, ΔT decreases monotonically with increasing d : wider gaps allow greater airflow through the enclosure, enhancing convective cooling of the screen walls and of the air surrounding the sensor.

However, this result must be interpreted with care. In reality, after a certain point, wider inter-louvre gaps will also provide optical paths through which diffuse and ground-reflected shortwave radiation can reach the sensor interior — an effect that is not captured by the present model, which treats the screen as a fully opaque radiation barrier (Section 3.5). The simulated monotonic decrease of ΔT with d is therefore a lower bound; in practice an optimal spacing is expected to exist that balances ventilation gain against increasing direct solar loading on the sensor.

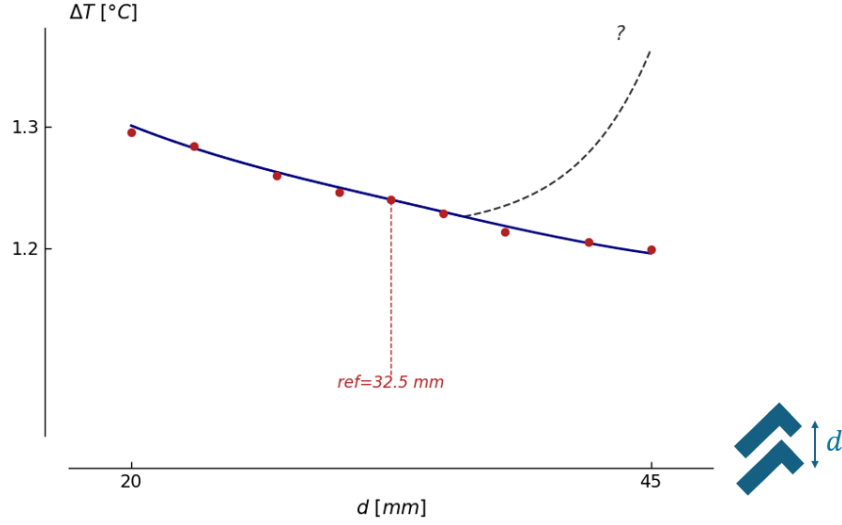


Figure 13: Thermal bias ΔT of the S2 screen as a function of inter-louvre spacing d . The simulated curve (navy) constitutes a lower bound, as the model does not account for shortwave radiation reaching the sensor through wider gaps. The dashed curve illustrates the expected qualitative behaviour in practice.

5.3.3. Overall screen size (scaling factor)

A uniform scaling factor k is applied to all screen dimensions, with $k = 1$ corresponding to the reference S2 geometry. Figure 14 shows that ΔT decreases slightly with increasing k . Larger enclosures present a greater internal volume, which dilutes the warm air inside the screen. The sensitivity is weak — a $\pm 10\%$ change in all dimensions produces a change in ΔT of only $\sim 0.05^\circ\text{C}$.

The trend of decreasing thermal bias with increasing screen size is consistent with the field observations of Buisan et al. [4], who reported lower temperature errors in larger Stevenson screens.

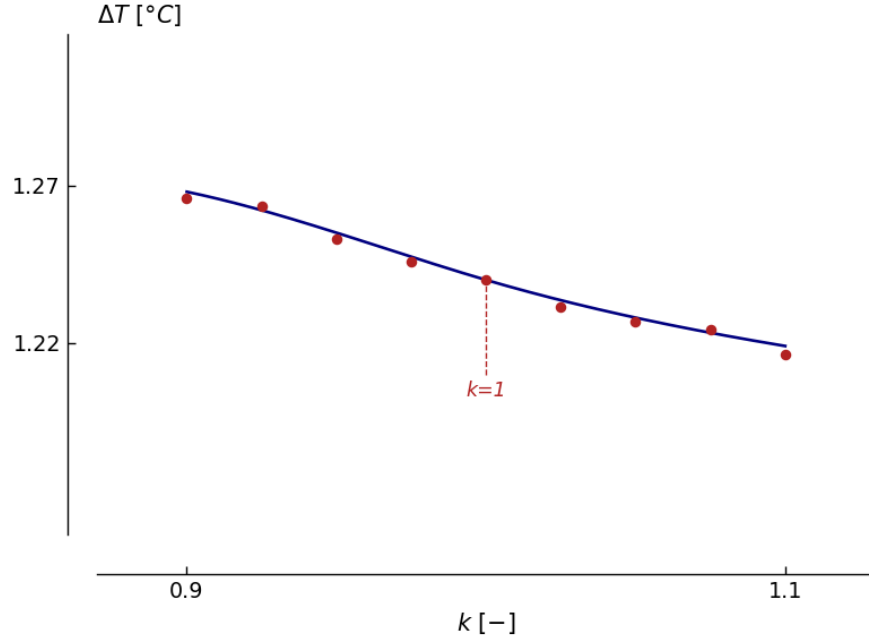


Figure 14: Thermal bias ΔT as a function of the uniform scaling factor k applied to the S2 geometry ($k = 1$: reference dimensions).

5.3.4. Screen material

Four materials are compared at reference conditions, keeping the geometry fixed: cedar, pine, oak, and plastic. Their thermophysical properties are summarized in Table 8.

Table 8: Thermophysical properties of the screen materials compared.

Material	ρ (kg m ⁻³)	κ_s (W m ⁻¹ K ⁻¹)	c_p (J kg ⁻¹ K ⁻¹)
Cedar	380	0.11	1600
Pine	520	0.13	1750
Oak	750	0.19	2390
Plastic	1050	0.20	1400

Figure 15 shows that the spread across all four materials reaches approximately 0.13°C, suggesting that within the family of materials typically used for meteorological screens, the choice of material has a non-negligible influence on the thermal bias.

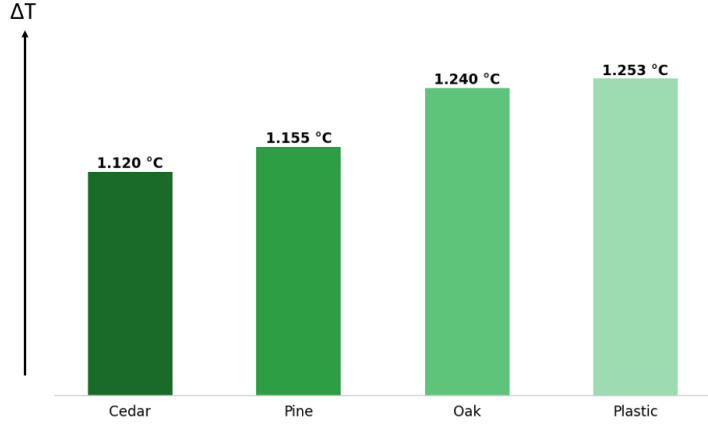


Figure 15: Thermal bias ΔT for four screen materials at reference conditions.

Traditionally, Stevenson screens are constructed from wood. In recent decades, however, plastic has become increasingly prevalent in operational networks, as it eliminates the need for periodic repainting and resists weathering without treatment. From a steady-state thermal perspective, the present results suggest that this operational convenience comes at a cost: plastic, being the most thermally conductive of the four materials tested, produces the largest bias. The spread of approximately 0.13°C across the material family is non-negligible and may contribute to inhomogeneities in long-term climate records when wooden screens are replaced by plastic ones without correction.

5.3.5. Optimal sensor position

Figure 16 shows the air temperature along the vertical centreline of the S2 enclosure, parameterized by the normalised height $\eta \in [0, 1]$ from the screen floor to the screen ceiling. The profile exhibits a pronounced warm layer near the floor ($\eta \approx 0\text{--}0.1$), driven by the heated slatted floor. Above this layer, the temperature drops to a broad minimum around $\eta \approx 0.15\text{--}0.4$, then rises again slightly near the sensor location and the upper louvres.

The sensor, currently positioned at $\eta \approx 0.55$, sits in a region where the local air temperature is higher than the minimum. Repositioning the sensor downward toward $\eta \approx 0.25$ would reduce the near-sensor air temperature by approximately 0.17°C under the reference conditions. However, it should be noted that this result is obtained at a single operating point ($U_\infty = 1\text{ m s}^{-1}$, $I = 800\text{ W m}^{-2}$); the robustness of this recommendation across a wider range of wind speeds and irradiances is verified hereunder.

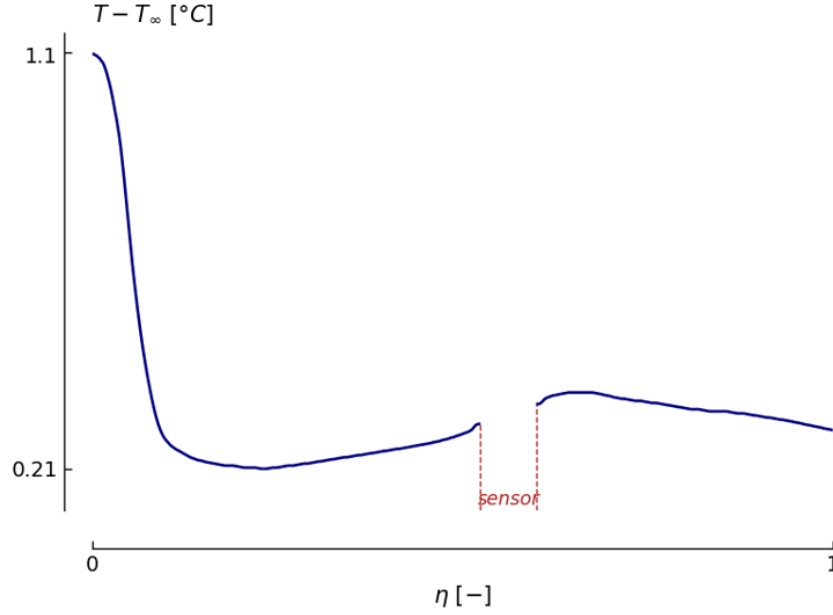


Figure 16: Air temperature along the vertical centreline (normalised height) of the S2 enclosure at reference conditions.

The sensitivity of $\eta_{\min}$ to wind speed and solar irradiance is assessed in Figure 17 to determine whether the recommended downward repositioning of the sensor remains valid across a range of operating conditions. Over the full parametric range investigated ($U_{\infty} \in [1, 5] \text{ m s}^{-1}$, $I \in [200, 1000] \text{ W m}^{-2}$), $\eta_{\min}$ varies only weakly and remains confined within $[0.21, 0.28]$. The position of the temperature minimum along the vertical centreline is therefore largely insensitive to meteorological forcing, and the recommendation to reposition the sensor toward $\eta \approx 0.25$ can be considered robust. This conclusion holds strictly within the steady-state framework adopted in the present work, and transient effects may alter the vertical temperature distribution.

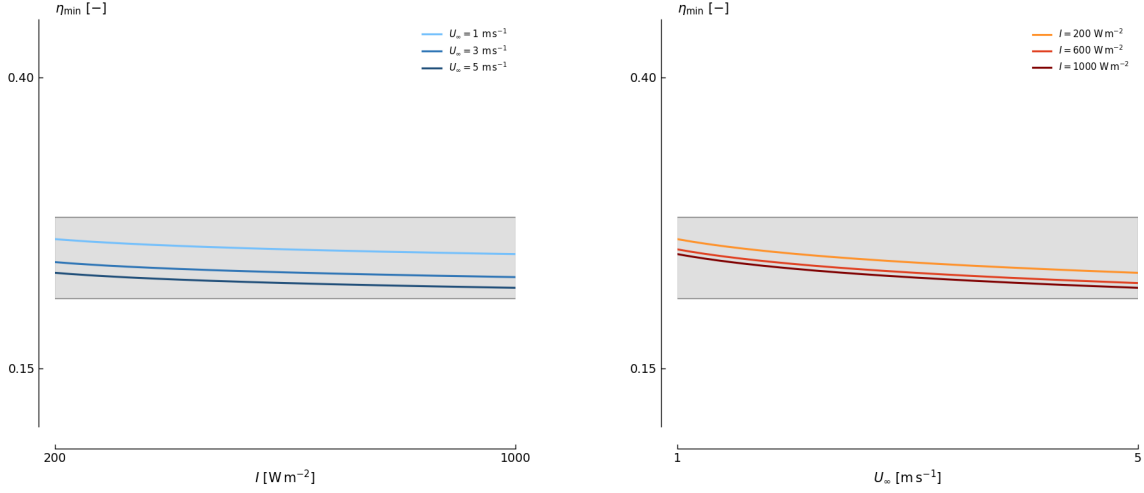


Figure 17: Sensitivity of $\eta_{\min}$ to solar irradiance I (left) and wind speed U_{∞} (right). The shaded band indicates the range within which $\eta_{\min}$ remains across all conditions investigated.

5.4. Comparative analysis

The three screen designs are now compared under identical conditions to assess how their geometry and ventilation principles translate into differences in thermal bias.

5.4.1. S1 and S2 shields

The two common Stevenson screen configurations, S1 and S2, are compared in Figure 18, which shows $\Delta T_{S1} - \Delta T_{S2}$ as a function of wind speed for both incident wind directions.

In the main direction, the two screens perform almost identically: the difference remains below 0.03°C at $U_{\infty} = 1 \text{ m s}^{-1}$ and decreases with higher wind speeds. This suggests that the lateral space available to the incoming airflow has little influence on the thermal bias, despite S2 offering nearly twice that of S1 in this direction.

For the lateral wind, the difference is substantially more pronounced: S1 exhibits a bias 0.14°C higher than S2 at $U_{\infty} = 1 \text{ m s}^{-1}$, decreasing progressively with wind speed. Overall, S2 outperforms S1 across all conditions investigated, though the difference is only operationally significant under lateral wind incidence at low wind speeds.

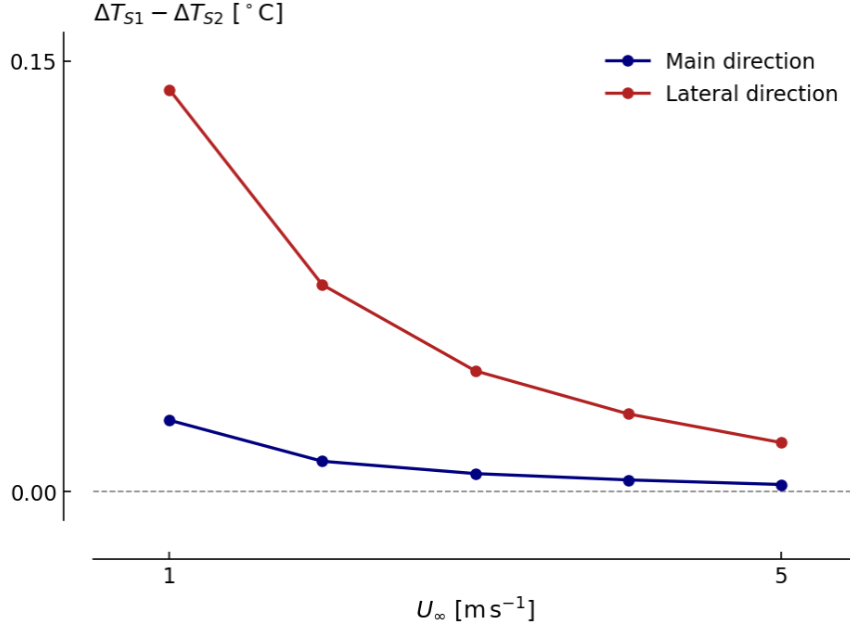


Figure 18: Difference in thermal bias $\Delta T_{S1} - \Delta T_{S2}$ as a function of wind speed U_∞ at $I = 800 \text{ W m}^{-2}$, for wind incident on the main (door) face and the lateral face.

5.4.2. Multi-plate shield

Field intercomparison campaigns have found that multi-plate screens perform worse than Stevenson screens under high solar irradiance [7], yet the present simulation predicts the opposite. This apparent contradiction stems directly from the opaque radiation barrier assumption of Section 3.

For Stevenson-type screens, this assumption is justified: the double-louvre geometry blocks all optical paths to the sensor, and the enclosed volume promotes the formation of a warm recirculation zone around the sensor — the dominant error mechanism, well captured by the CFD. For the MP screen, the situation is fundamentally different. As illustrated in Figures 19 and 20, air flows freely through the inter-plate gaps with no recirculation developing around the sensor, and the interior remains close to ambient temperature. This convective advantage is accurately represented in the simulation. What the model cannot capture, however, is the partial transmittance of shortwave radiation through those same gaps: by design, the stacked-plate geometry cannot simultaneously provide both unrestricted ventilation and a fully opaque radiation barrier. Lin et al. [13] quantified the solar radiation loading ratio — defined as the fraction of incoming global solar radiation reaching the sensor inside the shield — for a Gill-type multi-plate screen, and found values ranging from 2 to 12% depending on solar angle, the highest among all shields tested in their study. At the reference solar angle $\theta_z = 30^\circ$, this ratio is approximately 6%, confirming that direct shortwave loading on the sensor cannot be neglected.

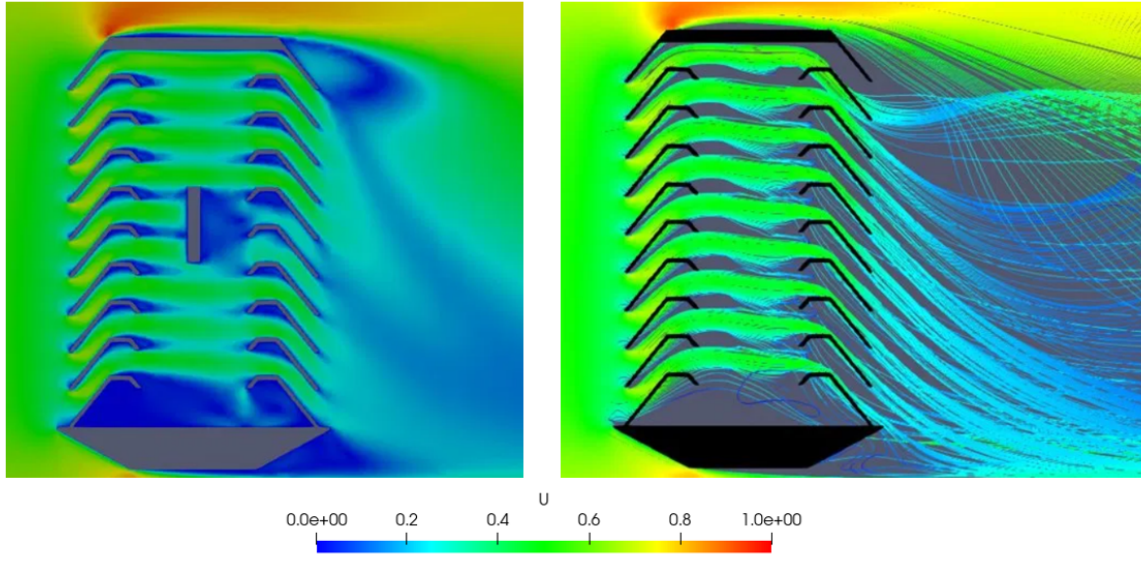


Figure 19: Velocity magnitude and streamlines for the MP screen at reference conditions (vertical cross-section).

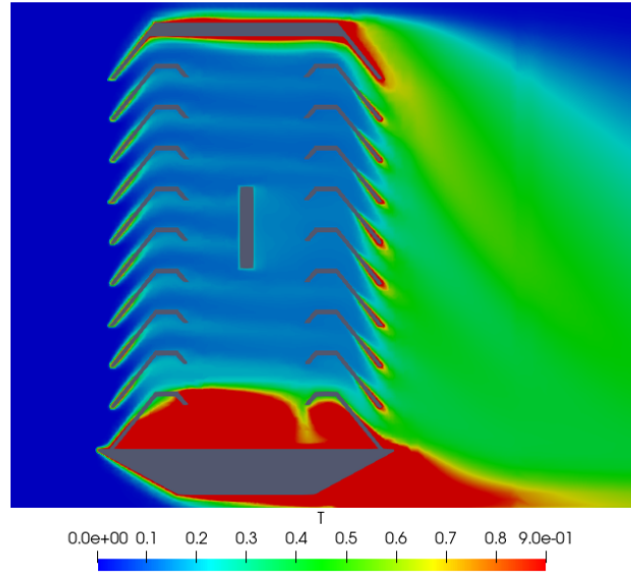


Figure 20: Temperature field for the MP screen at reference conditions.

In its current form, the CFD framework cannot account for this additional radiative loading, as the radiation model treats all screens as fully opaque barriers. To partially remedy this limitation, an additional heat flux is prescribed directly on the sensor surface, estimated as:

$$\dot{q}_{\text{sensor}} = r \cdot \alpha_{\text{sensor}} \cdot \text{GHI} \quad (19)$$

where $r = 6\%$ is the solar loading ratio from Lin et al. [13] at $\theta_z = 30^\circ$, $\alpha_{\text{sensor}} = 0.4$ is the shortwave absorptivity of the stainless steel sensor, and $\text{GHI} = I_{\text{dir}} \cos \theta_z + I_{\text{diff}}$, yielding $\dot{q}_{\text{sensor}} = 20.8 \text{ W m}^{-2}$. It should be noted that the exact geometry of the Gill-type shield tested by Lin et al. is not known, and their loading ratio therefore serves as an approximation for the present MP design. This approach does not replace a full radiative model — which would require computing view factors between the sensor, the plates, and the exterior, and solving the radiative transfer problem inline with the CFD — but provides a physically motivated first-order correction.

The corrected bias difference $\Delta T_{MP} - \Delta T_{S2}$ is shown in Figure 21.

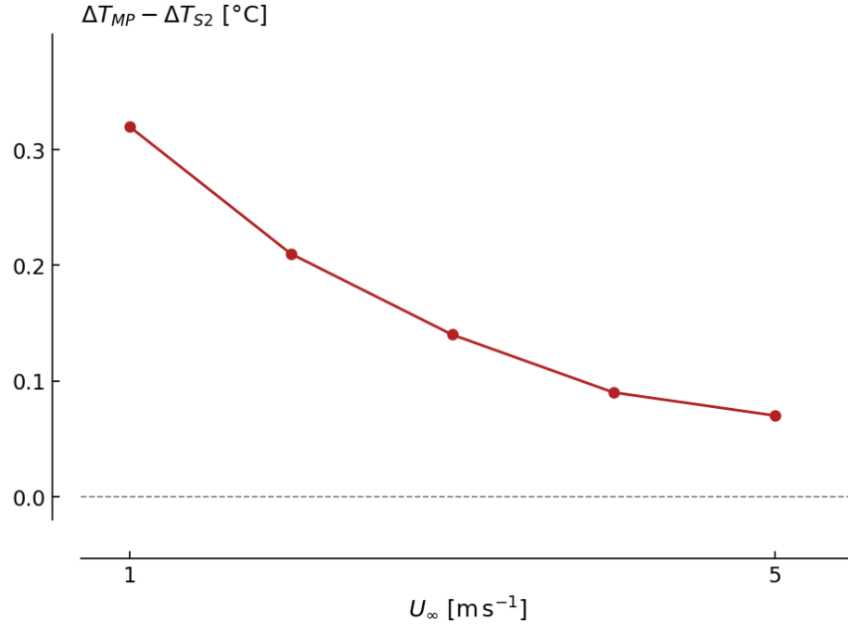


Figure 21: Difference in thermal bias $\Delta T_{MP} - \Delta T_{S2}$ as a function of wind speed U_∞ at $I = 800 \text{ W m}^{-2}$.

The MP shield exhibits a bias higher than the S2 screen, consistent with the qualitative finding of Erell et al. [7] that Gill-type screens perform worse than Stevenson screens under high solar irradiance. A direct quantitative comparison with field data is, however, not straightforward: Erell et al. report instantaneous differences of up to 1.76°C between a Gill-type screen and a Stevenson screen, but these peaks correspond to near-calm wind conditions and are inherently transient in nature. Their mean daily difference of approximately 0.08°C , on the other hand, integrates over night-time hours and variable cloud cover, yielding a much lower effective irradiance than the 800 W m^{-2} prescribed here. The present result therefore lies within a physically plausible range, though confirming it quantitatively would require field measurements under controlled, identical conditions. As wind speed increases, the difference narrows progressively from 0.32°C to 0.07°C , reflecting the diminishing influence of the fixed radiative load relative to the increasing convective cooling at the sensor surface.

6. Conclusion

This work presented a conjugate heat transfer framework implemented in the open-source solver OpenFOAM v12 for the simulation of meteorological screen thermal bias. Applied to three screen configurations under identical physical assumptions and boundary conditions, the framework offers a reproducible and computationally accessible alternative to commercial CFD tools for screen performance assessment.

The framework was validated against the numerical results of Dobre et al., reproducing the power-law dependence of the thermal bias on wind speed to within a mean relative error of 7.4%. It was then used to predict the bias over a parametric range of meteorological conditions, identify the geometric parameters that most strongly govern it, and compare the three screen designs under identical conditions.

On the meteorological side, solar irradiance and wind speed are confirmed as the dominant factors. Wind direction and solar zenith angle have a more moderate influence. On the geometric side, the most actionable levers for bias reduction are the choice of material and the vertical position of the sensor within the enclosure. By placing the thermometer lower in the screen, the bias was found to decrease consistently across all meteorological conditions investigated. The louvre inclination angle, by contrast, is already close to optimal in the RMI reference design.

In the comparative analysis, the S2 screen outperforms S1 across all conditions, though the difference is operationally significant only under lateral wind incidence at low wind speeds. The MP shield, after applying a first-order correction for direct solar loading on the sensor, exhibits a higher bias than S2 at low wind speeds — consistent with field observations — with the difference narrowing progressively at higher wind speeds. The overall ranking under the reference conditions is therefore $S2 > S1 > MP$ in terms of measurement accuracy.

The present framework has, however, important limitations. The radiation model treats all screens as fully opaque barriers, which is physically justified for Stevenson-type screens but structurally inadequate for the MP shield, for which the solar loading correction applied here remains a first-order approximation. The steady-state assumption also means that transient phenomena — such as the thermal lag of the screen structure under rapidly changing meteorological conditions — fall outside the scope of this work and represent a distinct source of bias not addressed here.

Future work should incorporate an internal radiative transfer model capable of handling partial shortwave transmittance through louvre and plate gaps, which would remove the opaque barrier assumption and enable a more physically complete treatment of certain shields. Extending the framework to transient simulations would further broaden its applicability, in particular for the study of screen response times and lag-induced errors.

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Declaration on Generative AI

During the preparation of this work, the author used Claude (Anthropic) and ChatGPT (OpenAI) to assist with language, structure, and L^AT_EX formatting, as well as to support debugging, including the interpretation of source code and documentation. These tools were not used to generate scientific content, analyse data, or produce conclusions. After using these tools, the author reviewed and edited all content as needed and takes full responsibility for the content of this paper.

References

- [1] Alduchov, O.A. and Eskridge, R.E. Improved Magnus form approximation of saturation vapour pressure. *Journal of Applied Meteorology*, 35(4):601–609, 1996.
- [2] Bernard, J., K  ravec, P., Morille, B., Bocher, E., Musy, M., and Calmet, I. Outdoor air temperature measurement: a semi-empirical model to characterize shelter performance. *Climate*, 7(2):26, 2019.
- [3] Brunt, D. Notes on radiation in the atmosphere. *Quarterly Journal of the Royal Meteorological Society*, 58(247):389–420, 1932.
- [4] Buis  n, S.T., Azor  n-Molina, C., and Jim  nez, Y. Impact of two different sized Stevenson screens on air temperature measurements. *International Journal of Climatology*, 35(14):4408–4416, 2015.
- [5] Cheng, X., Su, D., Li, D., Chen, L., Xu, W., Yang, M., Li, Y., Yue, Z., and Wang, Z. An improved method for correction of air temperature measured using different radiation shields. *Advances in Atmospheric Sciences*, 31(6):1460–1468, 2014.
- [6] Dobre, M., Sestan, D., and Merlone, A. Air temperature measurement uncertainty associated to a mounting configuration temperature sensor-radiation shield. In *WMO Technical Conference on Meteorological and Environmental Instruments and Methods of Observation (TECO)*, Amsterdam, October 2018.
- [7] Erell, E., Leal, V., and Maldonado, E. Measurement of air temperature in the presence of a large radiant flux: an assessment of passively ventilated thermometer screens. *Boundary-Layer Meteorology*, 114(1):205–231, 2005.

- [8] Erbs, D.G., Klein, S.A., and Duffie, J.A. Estimation of the diffuse radiation fraction for hourly, daily and monthly-average global radiation. *Solar Energy*, 28(4):293–302, 1982.
- [9] Garcia Izquierdo, C., Hernandez, S., Parrondo, M., Casas, A., Viola, A., Mazzola, M., Merlone, A., and Roulet, Y.-A. COAT Project: Intercomparison of thermometer radiation shields in the Arctic. *Atmosphere*, 15(7):841, 2024.
- [10] ISO. *Meteorology—Air Temperature Measurements—Test Methods for Comparing the Performance of Thermometer Shields/Screens and Defining Important Characteristics*. International Standard 17714:2007, ISO, Geneva, Switzerland, 2007.
- [11] Jarraud, M. *Guide to Meteorological Instruments and Methods of Observation (WMO-No. 8)*. World Meteorological Organization, Geneva, Switzerland, 2008.
- [12] Lacombe, M., Bousri, D., Leroy, M., and Mezred, M. *WMO Field Intercomparison of Thermometer Screens/Shields and Humidity Measuring Instruments, Ghardaïa, Algeria, November 2008–October 2009*. WMO/TD-No. 1579, World Meteorological Organization, Geneva, Switzerland, 2011.
- [13] Lin, X., Hubbard, K.G., Walter-Shea, E.A., Brandle, J.R., and Meyer, G.E. Some perspectives on recent in situ air temperature observations: modeling the microclimate inside the radiation shields. *Journal of Atmospheric and Oceanic Technology*, 18:1470–1484, 2001.
- [14] Liu, Q., et al. Computational fluid dynamic design and experimental study of a temperature sensor array used in climate reference station. *Journal of Atmospheric and Oceanic Technology*, 36(9):1627–1638, 2019.
- [15] Nakamura, R. and Mahrt, L. Air temperature measurement errors in naturally ventilated radiation shields. *Journal of Atmospheric and Oceanic Technology*, 22(7):1046–1058, 2005.
- [16] Richards, P.J. and Hoxey, R.P. Appropriate boundary conditions for computational wind engineering models using the $k-\epsilon$ turbulence model. *Journal of Wind Engineering and Industrial Aerodynamics*, 46–47:145–153, 1993.
- [17] Yang, J., Ge, X., Liu, Q., and Sun, Z. Design and experimental research of a temperature sensor applied to surface air temperature monitoring. *Measurement*, 182:109719, 2021.

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