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Energy performance and indoor airflow analysis of a healthcare ward designed with resource conservation objectives

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- Thermal comfort
- Healthcare ward
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- Heat gain

ABSTRACT

With the outbreak of COVID-19, the urgency of wide-scale healthcare infrastructure development has been felt globally for human survival. To accommodate a large infected population, copious wards are to be built within the prevalent constraints of land, power and material availability. This study designs a two-bed modular healthcare ward which is shrunk in size to minimize the requirement of space and other construction commodities such as materials, labour and power. Additionally, HVAC energy usage is accounted for conservation. The health safety and thermal comfort of occupants are regulated by monitoring indoor environment attributes while pushing towards a resource-efficient structure. Two popular envelope thermal retrofits viz. phase change materials and thermal insulation are tested to conceive gains in terms of improved energy performance of the ward. Various ward designs contest with their energy performance and occupant’s health safety and comfort characteristics in a multicriteria decision making process for delivering the most favourable solution. Subsequently, the most suitable solution is offered by a design involving thermal insulation retrofit with 8 ACH fresh air supply rate and 26°C inlet air temperature. The proposed design can support developing nations to contrive quick response to pandemic outbreaks with reduced construction (cost, time) and energy loads.

1. Introduction

Healthcare facility/ward built to accommodate multiple patients requires an indoor environment where airflow and temperature distribution are to be meticulously controlled. Airflow pattern across and within such enclosure plays a pivotal role in reducing (or amplifying) the risk of airborne transmission of infection from one occupant to another [1]. In parallel, the indoor temperature distribution has to be carefully maintained within thermal comfort range considering the fragile health conditions of occupants. Consistent control of both airflow and thermal attributes round the clock could be energy-intensive, particularly for facilities situated in regions where prevailing climate acts against achieving indoor thermal comfort (eg. significantly hot or cold climates).

As the entire world has been affected by the novel coronavirus (COVID-19/SARS-CoV-2) which is suspected to have airborne transmission potential [2,3], copious new wards are to be constructed to treat patients and quarantine affected people. The virus carries the potential of mutating into different strains [4], of which, little is known with high certainty about the transmission, infectious particle size and lifespan characteristics. While housing such patients, it is to be kept in mind that they may carry any form of airborne infection. Such pathogens vary in their sizes and therefore traverse different distances while entrained in the airflow. Accordingly, several numerical and experimental studies have been conducted to analyze the transmission of pathogens within control volumes [5–7]. In this context, there is a general consensus that the base airflow within an enclosure is a key determinant of pathogen distribution [8,9]. Also, preventing airflow from possible zones of contamination to cleaner zones minimizes disease spread [10]. We should therefore emphasize on ward designs which involve safe airflow characteristics and reduced indoor air retention to rapidly and comprehensively deal with diverse unforeseen pathogens, which has been a critical issue lately.

The pervasive COVID-19 crisis has also created an awareness that a large population can get infected at the same time with diseases having airborne transmission potential. The healthcare facilities in developing

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countries require a substantial expansion to tackle widespread health crises such as COVID-19. However, the ubiquitous resource constraints (such as land, material and labour availabilities, energy requirements etc) have to be dealt in a judicious manner during such wide-scale infrastructure upgrade, given the already faltered economy because of the pandemic. Small size modular wards could effectively minimize resource requirements for construction as well as maintenance.

In most of the studies on healthcare wards that focus on occupants’ safety and comfort indoors [11,12], the critical aspect of limited space and material availability for construction is not considered to determine the ward morphology. This is perhaps because the possibility of a contagious disease outbreak at a global scale (and thereby the requirement of vast avenues for patient isolation and treatment) was less anticipated until COVID-19. Where lean measures for dwelling construction were emphasized [13,14], although indoor thermal comfort was investigated on multiple occasions, other vital characteristics of indoor air (such as...
The two-bed ward to be designed in this study is targeted to have minimal resource requirements while ensuring the health safety of occupants. Adhering to this view, we set the ceiling height to be 2.7 m which is the minimum recommended value in Ref. [26]. The minimum separation between two occupants should be 1 m as per COVID-19 guidelines [27]. To account for other possible airborne contaminants, the width of the ward is set to 3.66 m where a minimum distance of 2 m separates between two occupants should be 1 m as per COVID-19 guidelines [27]. To account for other possible airborne contaminants, the width of the ward is set to 3.66 m where a minimum distance of 2 m.
can be maintained between the two occupants. The length of the room is set to 3.08 m where a clearance of 1 m is given for the front door opening and a minimum gap of 0.1 m is given between the patient bed and the nearest wall. A floor mounted air diffuser having an inlet area of 0.224 m² and a minimum gap of 0.1 m is given between the patient bed and the floor. A 15 W LED ceiling lamp (that generates 4.5 W heat) [30] is placed symmetrically at the centre of the two-bed room to supply fresh air. A 15 W LED ceiling lamp (that generates 4.5 W heat [29]) is used for indoor illumination purpose. The two occupants are considered in sleeping position in their respective beds. The dimensions of the manikin are taken from the model proposed by Miyanaga et al. [28].

The structural anatomy of the ward envelope is adopted from the real-time architectural survey conducted by Ahsan [31] and presented in Fig. 1(a). For retrofitting purpose, 1 cm thick retrofit layer (PCM/thermals) is planned to be deployed at the junction between brick layer and outer gypsum plaster. Table 1 lists the material properties of the envelope’s base components along with the thermal retrofits to be explored for energy performance enhancements.

Analysis of microflows occurring within the thin PCM layer is beyond the scope of the present study and therefore PCM layer is modelled as a solid where the change of phase is taken into account considering variable specific heat (Eqs. (1)–(3)) as done previously in similar studies [20,21].

\[
c_p = c_p \text{ when } T < T_S
\]  

(1)

c_p = c_p \text{ when } T > T_S

(3)

The governing equations for fluid flow and heat transfer (Eq. (6)-(10)) are solved using the finite volume method based commercial CFD code Ansys FLUENT 19.2 [37].

Continuity equation:  
\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{v}) = 0
\]  

(6)

Momentum equation:  
\[
\frac{\partial}{\partial t} (\rho \mathbf{v}) + \nabla \cdot (\rho \mathbf{v} \mathbf{v}) = - \nabla p + \nabla \cdot (\tau) + \rho \mathbf{g}
\]  

(7)

Boussinesq approximation (Eq. (8)) is used to model air density as a function of temperature for the buoyancy term in the momentum

\[
c_p = \frac{LH}{(T_L - T_S)} + \frac{(c_{pl} + c_{ps})}{2} \text{ when } T_S < T < T_L
\]  

(2)

Owing to the modular nature of the ward, it is considered that similar units are replicated in the ward’s lateral directions, thus leading to symmetry boundary conditions on the side walls. The front wall with a wooden door is considered to open to a shaded non-air-conditioned corridor. The rear wall (facing north) with the outlet grille and the roof of the ward are exposed to sunlight and outdoor ambient temperature. The retrofitting layers are planned to be installed in the roof and rear wall where high temperature swings are likely to occur because of sun exposure. The boundary conditions are summarized in Table 2. The thermo-fluidic behaviour of the ward is investigated under the transient diurnal variation of key weather parameters (presented in Fig. 2) for the hot summer month of May in New Delhi, India [32,34]. The outdoor convective heat transfer coefficient is taken as 22.7 W/m²K for New Delhi [34].

The effect of solar radiation incident on a surface is accounted by defining sol-air temperature in Eqs. (4) and (5) [32].

\[
T_{sol} = T_s + \frac{abs \times I}{h_v} - \frac{ems \times \Delta R}{h_v} \text{ (for roof)}
\]  

(4)

\[
T_{sol} = T_s + \frac{abs \times I}{h_v} \text{ (for wall)}
\]  

(5)

The values of ‘abs’ and ‘ems’ for gypsum layer are taken as 0.25 and 0.88 respectively [36].

Fig. 2. Monthly average air temperature and incident solar radiation in New Delhi for the month of May.
ρequation.

Energy equation for fluid domain:

\[
\frac{\partial}{\partial t}(\rho E) + \nabla \cdot \left( \rho \mathbf{u} E \right) = \nabla \cdot \left( \rho \mathbf{k} \nabla T \right) - H + \left( \rho \mathbf{u} \cdot \nabla \right) E + S
\]

where \( E = H - \frac{\rho}{\rho_o} + \frac{\mathbf{v}^2}{2} \) is the total energy of the system.

The standard k-ε model (Eq. (11)-(12)) is used for turbulence modelling after testing the congruence of numerical results with previous experimental data (presented in Section 2.2).

\[
\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho \mathbf{u} k) = \frac{\partial}{\partial x_j}\left( \frac{\mu_k}{\sigma_k} \frac{\partial k}{\partial x_j} \right) + C_1 \frac{\varepsilon}{k} (C_k G_k + C_\omega G_\omega) - C_2 \rho \varepsilon^2
\]

Energy equation for solid domain:

\[
\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_j}(\rho \mathbf{u} \varepsilon) = \frac{\partial}{\partial x_j}\left( \frac{\mu_\varepsilon}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_j} \right) + \frac{\varepsilon}{k} \left( \rho \mathbf{u} \cdot \nabla \right) k - \frac{\varepsilon}{k} S
\]

where \( \Gamma \) denotes the diffusion coefficient of \( \varepsilon \) and is defined with Eq. (15) [38]. \( S_\varepsilon \) is the source term for \( \varepsilon \) in Eq (14) which depends on the density of air.

\[
\Gamma = 2.89 \rho_i \times 10^{-4} + \frac{\rho_{\text{air}}}{0.7}
\]

The mean age of air (MAA) is defined as the average time required for the supply air to reach a particular point within the indoor domain through various possible flow paths [38]. MAA therefore indicates the freshness of indoor air. Calculation of MAA involves the solution of an additional scalar transport equation as mentioned below [38].

\[
\frac{\partial}{\partial x_j}\left( \rho_{\text{MAA}} \mathbf{u} \right) - \frac{\partial}{\partial x_j}\left( \rho_{\text{MAA}} \mathbf{u} \right) = S_{\text{MAA}}
\]

where the source term for MAA is defined by the adjoint of the boundary condition and provides a measure of the freshness of indoor air.

The standard k-ε model (Eq. (11)-(12)) is used for turbulence modelling after testing the congruence of numerical results with previous experimental data (presented in Section 2.2).

Execution of the CFD model formulated through the above-mentioned equations and boundary conditions will furnish vital design statistics pertaining to the ward such as indoor air temperature (and thermal comfort conditions thereof), indoor heat gain characteristics, airflow pattern and mean age of air inside the ward. To select an adequate HVAC setting with effective envelope retrofit strategy that would optimally meet all these design criteria, later in the study, a popular multicriteria decision making tool called Technique for Order Preference by Similarity to an Ideal Solution (TOPSIS) is employed. TOPSIS is based on the data normalization, weight assignments and Euclidian distance calculation between ideal/anti-ideal targets and different designs. A detailed stepwise explanation of the methodology is provided by Zhang [41]. We mention below the equations (in the sequence of computation) used to implement TOPSIS on the data matrix of the present study (presented in Section 3.5) featuring HVAC performance attributes for different design scenarios arising from the parametric analysis.

Standardized matrix: \( r_j = \frac{x_{ij}}{\sum_{i=1}^{m} x_{ij}} \) for \( i = 1, 2, \ldots, m; j = 1, 2, \ldots, n \)

Proportion calculation: \( p_j = \frac{x_{i1}}{\sum_{i=1}^{m} x_{ij}} \) for \( i = 1, 2, \ldots, m; j = 1, 2, \ldots, n \)

where \( r_j \) denotes standardized scores.

Entropy calculation for \( j \)th feature: \( \varepsilon_j = -\frac{1}{\ln m} \sum_{i=1}^{m} p_{ij} \ln p_{ij} \)

Entropy weight calculation for \( j \)th feature: \( w_j = \frac{1 - \varepsilon_j}{\sum_{j=1}^{n} (1 - \varepsilon_j)} \)

where \( w_j \) represents the weight for the feature/criterion \( j \) calculated by entropy method.

Weighted matrix calculation: \( v_{ij} = w_j r_{ij} \) for \( i = 1, 2, \ldots, m; j = 1, 2, \ldots, n \)

where \( v_{ij} \) denotes weighted score.

Ideal soln: \( A^+ = \sum_{i=1,2,\ldots,m} \left( \min_{j=1,2,\ldots,n} v_{ij} \right) \) and \( A^- = \sum_{i=1,2,\ldots,m} \left( \max_{j=1,2,\ldots,n} v_{ij} \right) \)

Anti-ideal soln: \( A^- = \sum_{i=1,2,\ldots,m} \left( \min_{j=1,2,\ldots,n} v_{ij} \right) \) and \( A^+ = \sum_{i=1,2,\ldots,m} \left( \max_{j=1,2,\ldots,n} v_{ij} \right) \)

where \( A^+ \) is the anti-ideal solution which possesses the worst values for each feature/criterion \( v_{ij} \) and \( v_{ij} \) is the opposite of the ideal solution.
Table 3

| Room component | Inlet 1 (Area 0.0684 m²) | Inlet 2 (Area 0.0684 m²) | Outlet (Area 0.5187 m²) | Radiant wall | Floor, Ceiling, other walls |
|----------------|--------------------------|--------------------------|--------------------------|--------------|----------------------------|
| Boundary condition | Velocity inlet (1.2 m/s), inlet air temperature 25.4 °C | Velocity inlet (1.38 m/s), inlet air temperature 25.4 °C | Pressure outlet (at atmospheric pressure) | Temperature (19 °C) | Temperature (19 °C) |

Fig. 3. (a) Geometry of reference test room (b) Meshed geometry.

Fig. 4. Velocity profiles on mid-plane A (present model versus reference literature).
\( J_j \) denotes the most optimal value of \( j^{th} \) feature when it is profitable in nature and \( J_j \) denotes the most optimal value of \( j^{th} \) feature when it is unprofitable in nature.

Euclidian distance from ideal soln \((S^+)\): 
\[
S^+ = \sqrt{\sum_{j=1}^{n} (v_{ij} - v_{ij})^2} 
\]  
(25)

Euclidian distance from anti-ideal soln \((S^-)\): 
\[
S^- = \sqrt{\sum_{j=1}^{n} (v_{ij} - v_{ij})^2} 
\]  
(26)

Closeness degree calculation \((C_i^+)\): 
\[
C_i^+ = \frac{s_j^i}{s_j^i + s_i^j}, \quad i = 1, 2, \ldots, m 
\]  
(27)

Higher value of \( C_i^+ \) represents greater optimality of a particular design option (and vice versa). Therefore, the best design is selected as the one which yields the highest value of \( C_i^+ \).

In order to check the reliability of the numerical modelling and solution procedure adopted in the present work, detailed validation is initially carried with the experimental works of Chafi & Halle [42] as explained in the next section.

2.2. Validation with experimental data

To validate the present model, the experimental observations by Chafi & Halle [42] pertaining to the indoor conditions of a test room are studied. The reference test room dimensions are 4.88 m (length) \( \times \) 3.66 m (width) \( \times \) 3 m (height) which bear similarity to the dimensions of the present study's model viz. 3.08 m (length) \( \times \) 3.66 m (width) \( \times \) 2.7 m (height). Fig. 3(a) presents the computational domain of the reference test room along with its salient details. Hot air enters into the room through two grilled diffusers located at the floor and exhaust air passes out through the outlet vent located at the ceiling. The experimentally measured dimensions and boundary conditions are mentioned in Table 3.

The computational domain is meshed with tetrahedral elements and mesh refinements are performed in the vicinities of air inlets and outlet as well as near room corners and surfaces (Fig. 3(b)). Analysis is performed in commercial CFD solver Ansys FLUENT 19.2. The standard \( k-\epsilon \) model is used for turbulence modelling. The experimentally measured boundary and inlet/outlet conditions mentioned in Table 3 are applied to the model. Pressure-Implicit with Splitting of Operators (PISO) algorithm is employed with second-order upwind discretization scheme for all solution variables to obtain the solution. The Boussinesq
approximation is used to model temperature dependence of air density. The convergence criteria are set as $5 \times 10^{-4}$ for all residuals except for energy residual where a limiting value of $10^{-6}$ is used. As the solution progresses and the residuals approach the defined limits of convergence, the temperature and velocity values are observed to change minimally at the design points of comparison with the reference study. Considering this point and also referring to existing literature \cite{43}, these criteria are then adopted for further simulations.

The values of velocity and temperature are extracted at those points (on the mid-plane A in Fig. 3(a) for different floor heights and distances from radiant walls) from the numerically computed solution where the same quantities were experimentally measured by Chafi & Halle. Figs. 4 and 5 compare the results obtained with the present numerical model against the results obtained experimentally as well as numerically by Chafi & Halle \cite{42}. It is observed that the velocity and temperature distributions obtained in the present work are in good agreement with the previous experimental values. Furthermore, the estimates of the present model are on par (and in several cases closer to the experimental values) with the numerical results of Chafi & Halle \cite{42}. This concordance between the present model’s results and the existing experimental data thus asserts the authenticity of the present model.

Fig. 6. Visualization of line 1, and reference points A, B, C.

Fig. 7. Grid independence check for (a) Temperature (c) Velocity (e) MAA and Timestep size independence check for (b) Temperature (d) Velocity (f) MAA.
The improvements in the results (in terms of proximity to the experimental values) of the present numerical model as compared to the previous models developed in Ref. [42] are mostly because of (i) the finer grid adopted in the present study in contrast to the highly coarse grids used in Ref. [42] where computational time saving was a major investigation objective of the reference study (ii) the use of two-equation $k$-$\varepsilon$ turbulence model in the present study in contrast to the zero equation turbulence model used in Ref. [42] (again with more emphasis on computational time saving).

### 2.3. Grid and timestep size independence study

For the healthcare ward considered in the present work, three meshes with different number of elements (684k elements for grid 1; 712k elements for grid 2; 900k elements for grid 3) are tested for grid size independence study. For each of these grids, values of temperature, velocity and MAA are extracted at 50 points along a line located at 0.8 m height from the floor, spanning from head to foot of patient on a plane longitudinally dividing the patient’s body into symmetric left and right halves. This line (Line 1) along with 3 equidistant reference points (Reference Points A, B, C) located on the line are shown in Fig. 6 for clear visualization. The observed quantities are noted at 2 p.m. of the diurnal cycle where a high gradient of temperature exists between room interior and exterior. Ventilation characteristics of 12 air changes per hour (ACH) and 24°C inlet air temperature are used for all three grids. Fig. 7 (a,c,e) compares temperature, velocity and MAA values for the three grids. The minuscule changes in the observed quantities over different grids confirm the grid independence of the solution. The grid with 684k elements is selected considering computation time and storage space. It may be possible to further coarsen the grid to have even lower computational requirements while maintaining spatial discretization invariability of results.

So far as temporal discretization is concerned, similar existing literature show that large timestep sizes (~30s–60s) could be used [44]. This is justified in a sense that the ambient conditions (eg. temperature, solar radiation) change at a very slow rate over a diurnal cycle. The 684k grid is thus simulated with different timestep sizes and results are noted at 2 p.m. of the diurnal cycle (Fig. 7(b,d,f)). The negligible changes in the observed quantities for different timestep sizes indicate the timestep size independence of the solution and thereby 30 s timestep size was selected for the present study. Timesteps larger than 30 s are not used as they require large number of iterations for convergence in each timestep.

### Table 4

| Design Case no. | ACH | Inlet air temperature | Retrofit       |
|----------------|-----|-----------------------|----------------|
| 1              | 12  | 24 °C                 | None           |
| 2              | 10  |                       |                |
| 3              | 8   |                       |                |
| 4              | 6   |                       |                |
| 5              | 12  | 26 °C                 |                |
| 6              | 10  |                       |                |
| 7              | 8   |                       |                |
| 8              | 6   |                       |                |
| 9              | 12  | 24 °C                 | PCM            |
| 10             | 10  |                       |                |
| 11             | 8   |                       |                |
| 12             | 6   |                       |                |
| 13             | 12  | 26 °C                 |                |
| 14             | 10  |                       |                |
| 15             | 8   |                       |                |
| 16             | 6   |                       |                |
| 17             | 12  | 24 °C                 | Thermal insulation |
| 18             | 10  |                       |                |
| 19             | 8   |                       |                |
| 20             | 6   |                       |                |
| 21             | 12  | 26 °C                 |                |
| 22             | 10  |                       |                |
| 23             | 8   |                       |                |
| 24             | 6   |                       |                |

The test matrix of different design cases investigated.

![Fig. 8. Average (diurnal) deviation from operative temperature (°C).](image-url)
3. Results and discussion

The validated numerical model is then used to evaluate the thermo-fluidic behaviour of the present ward interior with dimensions close to the reference test room considered for validation. A detailed parametric study is carried out considering different ACH values, inlet air temperatures and retrofitting options for the given ward to engender changes in the indoor environment and eventually to determine the most favourable design from a holistic standpoint of occupant’s health safety, comfort and ward’s energy exchange characteristics. Although a high air supply rate of 12 ACH is recommended for new healthcare wards [45], we investigate the effect of lowering the ACH value in small steps to see if adequate indoor conditions could be maintained with lower supply rates. Altogether, 24 different design cases are investigated which are listed in Table 4.

3.1. Thermal comfort of occupants

To quantitatively represent the indoor thermal conditions for all the cases considered, the deviation of diurnal average of volumetric (DAV) mean indoor air temperature from the operative temperature (−25.8°C) is plotted for each case in Fig. 8. Case 21 with the smallest deviation is deemed to be the most suitable design strictly in this context whereas Case 17 turns out to be the least suitable design with maximum...
deviation.

To have a qualitative insight of indoor thermal comfort and to observe the effect of retrofitting, we select the Case 23 (which is deemed as an overall favourable design in a later Section 3.5) and its retrofit counterparts (Cases 7&15) with the same fresh air supply characteristics (ACH value and inlet temperature). The temperature distribution at 12 p.m. and 12 a.m. are depicted for each case in Fig. 9. According to the IMAC model [40], the 90% acceptability limit for indoor thermal comfort is ±1.5°C of operative temperature i.e., 24.3°C–27.3°C for the present study. Fig. 9 shows the accumulation of hot air near the roof which can be attributed to the buoyancy effect of air, heat gain through the roof and heat generation by the ceiling lamp. While the near roof region shifts towards thermally uncomfortable temperature range, the regions near the patients fall within desirable thermal comfort levels. The structural materials of the building envelope (with/without retrofit) manifest as thermal mass which induce lags in heat propagation from outdoor to indoor environment. The input heat during day hours slowly propagates through the envelope (having high thermal mass and thermal resistance) and reaches the indoor environment towards the evening hours, thereby causing temperature rise with a phase lag with outdoor temperature. This effect is more pronounced in case of PCM retrofitted envelope (Fig. 9(d)) owing to the high heat storage capacity added by the retrofit material.

Overall, the fresh air supply characteristics considered in the present study appears to chiefly govern the indoor thermal comfort conditions while shadowing the effect of retrofitting in this aspect. This is also

![Figure 10](image.png)

**Fig. 10.** Diurnal average of pointwise temperature variations in the proximity of patient for different design cases.

![Figure 11](image.png)

**Fig. 11.** Heat gain and cooling energy supplied for different design cases.
evinced in Fig. 8 where a change of 2°C in the inlet air temperature could tip the entire balance of thermal comfort (from best in Case 21 to worst in Case 17) where all other design parameters are identical.

A closer inspection of temperature levels near the patient is performed by extracting temperature values at Reference Points A, B and C designated in Fig. 6. This exercise is carried out to check if any sharp temperature gradient exists in the proximity of the patient from head to foot (which is undesirable). The diurnal average of these point temperature values for each case is plotted in Fig. 10. Most of the cases demonstrate comfortable temperature distribution (between 24.3°C and 27.3°C) in the vicinity of the patient resting in the bed. Only in some cases (viz. Cases 1, 2, 9, 10, 17, 18) the patient is likely to feel a slight cooling sensation near the legs. A somewhat noticeable temperature disparity (~0.5°C) exists between Reference Point B (near abdomen)
and Reference Point C (near legs) for the design cases with air supply characteristics of 10 ACH and 24°C inlet air temperature (viz. Cases 2, 10, 18) irrespective of the retrofitting strategy deployed. Such disparities should be avoided by opting for other available design options.

The diurnal variations of volumetric mean indoor air temperature and reference points temperatures in the proximity of patient for different design cases are provided in the supplementary data.

### 3.2. Cooling energy supplied and indoor heat gain

The healthcare ward requires a consistent supply of cooling energy in the form of fresh cold air introduced into the ward. The diurnal average rate of cooling energy supply can be estimated from the difference in the energy content between air at mean outdoor temperature ($T_{\text{mean outdoor}} = 32.7$°C from Fig. 2) and air at diffuser inlet temperature ($T_{\text{inlet}} = 24$°C/$26$°C) as expressed in Eq. (28).

$$q = \dot{m}_{\text{c}} c_{\text{pa}} (T_{\text{mean outdoor}} - T_{\text{inlet}}) \quad (28)$$

The value of $c_{\text{pa}}$ is taken as 1005 J/kg-K [46]. Total diurnal cooling energy supply is then evaluated by multiplying with the diurnal cycle duration (Eq. (29)).

$$Q_{\text{total}} = q \times \Delta t_{\text{diurnal cycle}} \quad (29)$$

Fig. 11 shows the diurnal cooling energy supplied for different cases. An optimal design scenario would be the case where the patients’ health safety and thermal comfort could be maintained with minimum cooling energy supply.

When fresh air is supplied at human thermal comfort levels (24°C–26°C) and the ward is designed to purge off the supplied air with minimum indoor residence time, it could be anticipated that the blown-out air still contains a cogeneration potential owing to its temperature difference with the hot outside ambient air. This cold exhaust air could be used for secondary cooling purposes, such as, to cool the condenser (typically operating at a temperature of 50°C [47]) of the air conditioning system generating the cold air for the healthcare ward(s). Heat gain through the ward envelope acts to raise the heat content of the indoor air, thereby reducing the cooling capacity of the exhaust air for secondary cogeneration purposes. A suitable design should account for minimum amount of heat gain through the envelope so that the exhaust air retains greater potential for secondary cogeneration purposes.

The envelope heat gain quantities for the different cases studied are presented in Fig. 11. The effect of retrofitting is evident in this aspect, where heat gain is significantly reduced by retrofitting the envelope. Also, it can be observed in Fig. 11 that the thermal insulation inhibits heat ingress to a greater extent than the PCM for similar air supply characteristics.

Another observation from Fig. 11 is that the cases involving high cooling energy supply also lead to higher heat gain (eg. Cases 1, 9, 17). In these cases, the high rate of cooling energy supply creates colder indoor air conditions near the envelope’s inner surfaces (by high ACH with low air supply temperature) and enhances convective heat transfer from the envelope’s inner surface to the indoor air (due to increased indoor air motion at high ACH). This creates a high gradient of temperature from the exterior to the interior and reduces convective resistance of heat transfer from the envelope’s inner surface to indoor air. These effects in turn accelerate the heat propagation to the interior.

### 3.3. Airflow patterns

To have a visual idea of the airflow patterns within the ward and to understand the effect of retrofit on indoor airflow paths, the overall best Case 23 (explained in Section 3.5) is depicted along with its similar air supply (but different retrofit) counterparts i.e. Cases 7 & 15. 3D streamlines at 12 p.m. and 12 a.m. for each of these cases are displayed from top view to show the formation of vortices within the ward (Fig. 12). In each sub-figure in Fig. 12, formation of two major vortices could be observed towards the legs of patients. These vortices originate due to the mixing of inlet airstreams being rebound from the front and rear walls. This phenomenon also creates two minor vortices towards the heads of patients which are however less prominent (shown by streamlines with less coherent spiral shapes and with lower MAA values) due to the presence of exhaust grille on the rear wall. Air trapped in the major vortices is likely to spend longer time within the ward, thereby elevating the MAA values in such regions. As the major vortices form...
towards the legs of patients and not directly over the heads, there will be lower risk of stale/contaminated air moving towards the patients’ breathing zones.

The effect of air curtain created by the centrally placed air diffuser can also be seen in Fig. 12 which prevents the movement of air from one occupant to the other.

3.4. Mean age of air (MAA)

High rate of fresh air supply is recommended for healthcare wards so that contaminated indoor air can be expelled at the earliest. MAA inside a room depends upon the indoor airflow pattern which in turn is affected by the air temperature distribution and thermal boundary conditions imposed in addition to the rate of fresh air supply. Most of these causative parameters can vary with time of the day and also vary for different design cases considered. The values of volumetric mean MAA and volumetric maximum MAA are calculated on hourly basis for each of the design cases. Also, the diurnal average values of these two attributes are noted for each case (Fig. 13). The diurnal average of volumetric (DAV) mean MAA follows a perceptible trend which shows that for each envelope retrofit (no retrofit, PCM, thermal insulation), DAV mean MAA increases as the fresh air supply rate decreases. This parameter is therefore chiefly dependent upon the fresh air supply rate. However, the same statement does not hold true for DAV maximum MAA. Comparison

![Fig. 14. MAA distribution for (a) Case 7 at 12pm (b) Case 7 at 12am (c) Case 15 at 12pm (d) Case 15 at 12am (e) Case 23 at 12pm (f) Case 23 at 12am.](image-url)
of Case 18 and Case 19 (and similarly Cases 6 vs 7; 14 vs 15; 22 vs 23) reveals an important finding i.e. increasing the rate of fresh air supply need not always lead to a reduction in the DAV maximum MAA value. With a lower rate of fresh air supply, Case 19 leads to a lower value DAV maximum MAA. For this parameter, fresh air supply rate alone cannot thus determine the major trend of variation and urge for proper control of other causative factors to avoid stale air accumulation in certain zone(s) of the ward.

In addition to the reverse trends observed for DAV maximum MAA in Cases 18 and 19, the total heat gain, deviation from thermal comfort and distance of maximum MAA point from patient’s head are also more optimal for case 19 than case 18 (tabulated in Section 3.5) i.e. the design targets are better achieved with the case having relatively lower ACH. The intricate inter-dependences of airflow rate, air temperature and boundary conditions lead to different thermo-fluid dynamics of indoor air for each combination of the influencing variables which may not align with our general intuitions as illustrated for these two cases.

For judicious space utilization in the manuscript, the volumetric distribution of MAA is visualized at 12 p.m. and 12 a.m. for the overall best Case 23 (refer to Section 3.5) along with Cases 7 & 15 (which possess the same fresh air supply characteristics) to understand the effect of different retrofits (Fig. 14). The location of volumetric maximum MAA in each case is marked with red point. Fig. 14 shows that for all the cases, stale air tends to accumulate near the central part of the roof. Therefore, placement of the outlet grille near the roof along the roof central line is a good practice to vent out the stale air (shown by shrinking stale air contours towards the outlet). It also supports the scheme of placing the heads of patients towards the rear wall so that less amount of stale air accumulates above the heads (as compared to above the legs). It should be noted that the outlet grille is placed in the central line with the motive that the exhaust air’s path should be at the largest distance symmetrically from both the patients who are placed towards the side walls to maintain a safe distance from each other. Further, referring to existing studies (such as by Rees and Haves [28], Jadidi et al. [48]) involving floor mounted air diffuser within enclosures, the outlet grille is placed near the roof. Finally, the outlet grille is placed on the rear wall and not on the front wall so as to avoid exhausting the stale air towards the entrance of the room where human presence is likely to be more frequent. Besides these justifications for the present placement of the outlet grille, further parametric studies could be pursued in this direction.

Case 7 (with no retrofit) results in a lower MAA distribution, implying that the stale air is purged off more efficiently than the other cases. This is also reflected in Fig. 13 where Case 7 yields lower values of DAV mean MAA as well as DAV maximum MAA. Therefore, strictly in this aspect, envelope retrofitting options do not endow favourable gains. Variation in MAA for different hours occurs due to the changes in the thermo-fluid characteristics of indoor air domain as functions of transient outdoor boundary conditions. The general tendency of stale air accumulation near the central roof is however consistent at all the hours for all the cases (not shown due to manuscript space limitation).

The coordinates of the point having maximum MAA can be extracted from the CFD postprocessor and its distance from Reference Point A (labelled in Fig. 6) in front of the patient’s face can be calculated for each design case. As the location of the maximum MAA point changes at different hours, the diurnal average of the location is computed for each case and presented in Fig. 15. Since the location of this point is sensitive to indoor airflow patterns, heat gain, temperature distribution, rate/temperature of air supply, turbulent eddy formation and other thermo-fluid interactions within the indoor atmosphere, it is difficult to predict the motion of this point (across different cases and at different hours) by correlating with any of the influencing variables. Fig. 15 shows that the diurnal average distance between maximum MAA point and Reference Point A is largest for Case 11 which is indicative of a safer design. In contrast, the distance is minimum in Case 1 which signifies a less safe indoor condition. The difference in the distances for the two extreme cases in this context is 0.55 m which indeed is significant for the given ward dimensions.

Pointwise values of MAA levels near the patient (at Reference Points A, B and C labelled in Fig. 6) are extracted for different cases to have an
estimate of the freshness of air in the proximity of patient. Fig. 16 plots the diurnal averages of the pointwise MAA values for each case.

Ideally, cases with lowest possible values of MAA at all of these points and low disparities among the pointwise MAA values should be selected. Fig. 16 shows that for the points of interest, Cases 1, 5, 9, 13, 17 and 21 bear these favourable characteristics. These are the cases involving 12 ACH inlet air supply rate. Therefore, favourable pointwise MAA distribution near the patient is cardinally regulated by the fresh air supply rate which is the same trend observed for volumetric mean MAA values (Fig. 13).

### 3.5. Optimal design selection

In order to select an optimal design out of the various cases investigated, the different performance attributes pertaining to the ward envelope and indoor atmosphere need to be considered for their respective optimality. These performance attributes do not vary coherently towards their favourable values across different design cases. Hence, the optimal design is selected by implementing TOPSIS which is a popular multicriteria decision making methodology.

Table 5 summarizes the diurnal average/total values of key performance attributes for the design cases considered (Initial data matrix for TOPSIS).

| Case no. | Criteria 1 | Criteria 2 | Criteria 3 | Criteria 4 | Criteria 5 | Criteria 6 |
|----------|------------|------------|------------|------------|------------|------------|
|          | Diurnal average of volumetric mean MAA (s) | Diurnal average of volumetric max. MAA (s) | Average (diurnal) distance of max. MAA point from point 1 (m) | Average (diurnal) deviation from thermal comfort temperature (°C) | Total diurnal indoor heat gain (kJ/day) | Total diurnal cooling energy supplied (kJ/day) |
| 1        | 129.17     | 486.53     | 2.27       | 1.14       | 27281.05   | 93976.54   |
| 2        | 155.22     | 506.70     | 2.37       | 1.05       | 25128.43   | 78263.42   |
| 3        | 163.37     | 373.27     | 2.69       | 0.91       | 21761.89   | 62701.39   |
| 4        | 214.70     | 469.63     | 2.57       | 0.72       | 19601.93   | 46988.27   |
| 5        | 126.08     | 429.39     | 2.35       | 0.69       | 21196.87   | 72372.74   |
| 6        | 153.50     | 488.74     | 2.31       | 0.75       | 19470.24   | 60271.83   |
| 7        | 164.96     | 351.08     | 2.70       | 0.88       | 17273.41   | 48287.28   |
| 8        | 212.90     | 442.58     | 2.61       | 1.02       | 15488.57   | 36186.37   |
| 9        | 127.87     | 462.99     | 2.33       | 1.15       | 21067.88   | 93976.54   |
| 10       | 153.11     | 492.69     | 2.31       | 1.05       | 19361.56   | 78263.42   |
| 11       | 177.90     | 484.35     | 2.82       | 0.95       | 16663.25   | 62701.39   |
| 12       | 220.51     | 553.38     | 2.74       | 0.81       | 15173.78   | 46988.27   |
| 13       | 125.56     | 426.73     | 2.41       | 0.69       | 16413.48   | 72372.74   |
| 14       | 152.04     | 484.00     | 2.33       | 0.74       | 15103.76   | 60271.83   |
| 15       | 172.28     | 418.14     | 2.73       | 0.83       | 12695.26   | 48287.28   |
| 16       | 214.60     | 496.68     | 2.67       | 0.94       | 11252.58   | 36186.37   |
| 17       | 132.86     | 425.56     | 2.37       | 1.25       | 16262.68   | 93976.54   |
| 18       | 147.15     | 513.42     | 2.29       | 1.18       | 15234.62   | 78263.42   |
| 19       | 169.95     | 488.77     | 2.81       | 1.07       | 13088.09   | 62701.39   |
| 20       | 213.87     | 518.24     | 2.69       | 0.94       | 12029.26   | 46988.27   |
| 21       | 121.42     | 381.25     | 2.44       | 0.61       | 12675.85   | 72372.74   |
| 22       | 145.86     | 457.77     | 2.44       | 0.65       | 31815.13   | 60271.83   |
| 23       | 165.03     | 377.66     | 2.69       | 0.75       | 10301.44   | 48287.28   |
| 24       | 209.25     | 449.73     | 2.59       | 0.85       | 9253.91    | 36186.37   |

Fig. 16. Diurnal average of pointwise MAA variations in the proximity of patient for different design cases.
### Table 6
Weight normalized decision matrix and ranking of design cases using TOPSIS.

| Case no. | Diurnal avg. of volumetric mean MAA (s) | Diurnal avg. of volumetric max. MAA (s) | Avg. (diurnal) distance of max. MAA point from point 1 (m) | Avg. (diurnal) deviation from thermal comfort temperature (°C) | Total diurnal indoor heat gain (kJ/day) | Total diurnal cooling energy supplied (kJ/day) | Euclidian dist. from ideal solution (S+ + S-) | Euclidian dist. from anti-ideal solution (S-) | Performance scores (S-) / [(S+ + S-) + S-] | Rank |
|----------|----------------------------------------|----------------------------------------|-----------------------------------------------------------|-----------------------------------------------------|----------------------------------------|---------------------------------------------|------------------------------------------|-------------------------------------------|------------------------------------------|------|
| 1        | 0.0242                                 | 0.0106                                 | 0.0038                                                    | 0.0397                                              | 0.0974                                 | 0.0946                                      | 0.0888                                   | 0.0176                                    | 0.1064                                   | 0.1654 | 24   |
| 2        | 0.0291                                 | 0.0110                                 | 0.0040                                                    | 0.0365                                              | 0.0897                                 | 0.0788                                      | 0.0728                                   | 0.0226                                    | 0.0953                                   | 0.2367 | 23   |
| 3        | 0.0306                                 | 0.0081                                 | 0.0045                                                    | 0.0319                                              | 0.0777                                 | 0.0631                                      | 0.0537                                   | 0.0406                                    | 0.0943                                   | 0.4303 | 18   |
| 4        | 0.0402                                 | 0.0102                                 | 0.0043                                                    | 0.0251                                              | 0.0700                                 | 0.0473                                      | 0.0426                                   | 0.0577                                    | 0.1003                                   | 0.5756 | 13   |
| 5        | 0.0236                                 | 0.0094                                 | 0.0040                                                    | 0.0241                                              | 0.0757                                 | 0.0728                                      | 0.0562                                   | 0.0405                                    | 0.0967                                   | 0.4188 | 19   |
| 6        | 0.0288                                 | 0.0107                                 | 0.0039                                                    | 0.0262                                              | 0.0695                                 | 0.0607                                      | 0.0446                                   | 0.0489                                    | 0.0935                                   | 0.5228 | 16   |
| 7        | 0.0309                                 | 0.0077                                 | 0.0046                                                    | 0.0307                                              | 0.0617                                 | 0.0486                                      | 0.0335                                   | 0.0607                                    | 0.0942                                   | 0.6441 | 9    |
| 8        | 0.0309                                 | 0.0096                                 | 0.0044                                                    | 0.0356                                              | 0.0553                                 | 0.0364                                      | 0.0316                                   | 0.0723                                    | 0.1039                                   | 0.6957 | 7    |
| 9        | 0.0240                                 | 0.0101                                 | 0.0039                                                    | 0.0400                                              | 0.0752                                 | 0.0946                                      | 0.0743                                   | 0.0285                                    | 0.1028                                   | 0.2770 | 22   |
| 10       | 0.0287                                 | 0.0107                                 | 0.0039                                                    | 0.0367                                              | 0.0692                                 | 0.0788                                      | 0.0582                                   | 0.0355                                    | 0.0936                                   | 0.3788 | 21   |
| 11       | 0.0334                                 | 0.0106                                 | 0.0048                                                    | 0.0333                                              | 0.0595                                 | 0.0631                                      | 0.0410                                   | 0.0510                                    | 0.0920                                   | 0.5545 | 14   |
| 12       | 0.0413                                 | 0.0121                                 | 0.0046                                                    | 0.0282                                              | 0.0542                                 | 0.0473                                      | 0.0313                                   | 0.0659                                    | 0.0972                                   | 0.6781 | 8    |
| 13       | 0.0235                                 | 0.0093                                 | 0.0041                                                    | 0.0239                                              | 0.0586                                 | 0.0728                                      | 0.0446                                   | 0.0518                                    | 0.0965                                   | 0.5274 | 15   |
| 14       | 0.0285                                 | 0.0106                                 | 0.0039                                                    | 0.0258                                              | 0.0539                                 | 0.0607                                      | 0.0330                                   | 0.0594                                    | 0.0923                                   | 0.6428 | 10   |
| 15       | 0.0323                                 | 0.0091                                 | 0.0046                                                    | 0.0291                                              | 0.0453                                 | 0.0486                                      | 0.0213                                   | 0.0716                                    | 0.0929                                   | 0.7705 | 4    |
| 16       | 0.0402                                 | 0.0108                                 | 0.0045                                                    | 0.0330                                              | 0.0401                                 | 0.0364                                      | 0.0224                                   | 0.0824                                    | 0.1048                                   | 0.7858 | 3    |
| 17       | 0.0230                                 | 0.0093                                 | 0.0040                                                    | 0.0435                                              | 0.0581                                 | 0.0946                                      | 0.0672                                   | 0.0435                                    | 0.1106                                   | 0.3931 | 20   |
| 18       | 0.0276                                 | 0.0112                                 | 0.0039                                                    | 0.0410                                              | 0.0544                                 | 0.0788                                      | 0.0518                                   | 0.0479                                    | 0.0997                                   | 0.4808 | 17   |
| 19       | 0.0319                                 | 0.0098                                 | 0.0048                                                    | 0.0372                                              | 0.0467                                 | 0.0631                                      | 0.0353                                   | 0.0608                                    | 0.0961                                   | 0.6329 | 11   |
| 20       | 0.0401                                 | 0.0113                                 | 0.0045                                                    | 0.0328                                              | 0.0430                                 | 0.0473                                      | 0.0258                                   | 0.0729                                    | 0.0987                                   | 0.7390 | 5    |
| 21       | 0.0228                                 | 0.0083                                 | 0.0041                                                    | 0.0212                                              | 0.0453                                 | 0.0728                                      | 0.0384                                   | 0.0636                                    | 0.1021                                   | 0.6235 | 12   |
| 22       | 0.0273                                 | 0.0100                                 | 0.0041                                                    | 0.0227                                              | 0.0422                                 | 0.0607                                      | 0.0265                                   | 0.0695                                    | 0.0960                                   | 0.7243 | 6    |
| 23       | 0.0309                                 | 0.0082                                 | 0.0045                                                    | 0.0263                                              | 0.0368                                 | 0.0486                                      | 0.0160                                   | 0.0788                                    | 0.0948                                   | 0.8315 | 1    |
| 24       | 0.0392                                 | 0.0098                                 | 0.0044                                                    | 0.0298                                              | 0.0331                                 | 0.0364                                      | 0.0187                                   | 0.0879                                    | 0.1066                                   | 0.8245 | 2    |
| Ideal    |                                        |                                        |                                                           |                                                     |                                        |                                            |                                          |                                          |                                          |                                          |      |
| soln.:   |                                        |                                        |                                                           |                                                     |                                        |                                            |                                          |                                          |                                          |                                          |      |
| Anti-    | 0.0413                                 | 0.0121                                 | 0.0038                                                    | 0.0435                                              | 0.0974                                 | 0.0946                                      |                                          |                                          |                                          |      |
| ideal    |                                        |                                        |                                                           |                                                     |                                        |                                            |                                          |                                          |                                          |      |
| soln.:   |                                        |                                        |                                                           |                                                     |                                        |                                            |                                          |                                          |                                          |      |


4. Conclusions

This study endeavoured to design a resource and energy conservative modular healthcare ward wherein occupant’s health safety and thermal comfort could be adequately maintained. Densely populated hot climatic regions that are prone to mass outbreaks of airborne diseases will benefit from the proposed conservative healthcare infrastructure. We emphasized designing the ward according to mean age of air and base airflow pattern so that the ward could be used for accommodating patients with a wide range of infection conditions. This could also mean any possible future variant of SARS-CoV-2 or similar pathogens whose rate of release from sources, active infectious particle sizes and lifespan are largely uncertain. To tackle such unforeseen scenarios swiftly with impromptu arrangements, we believe a ward with reduced air stagnation in the interior would be a reasonable way to proceed rather than designing for a particular species of pathogen. Further, the resource conservative measures integrated into the present design will curtail the repercussions on the environment while prioritizing human safety.

From the heat transfer and fluid flow perspectives, the transient conjugate heat transfer through a finite thickness envelope was studied in terms of changes in the indoor air characteristics. This analysis bridged the gap between two distinct groups of studies where either indoor airflow modelling is prioritized with simplified wall boundary conditions or heat transfer through the envelope is investigated with simplified indoor air conditions. The outcomes of the present analysis indicated that while certain indoor environment parameters such as DAV mean MAA and indoor thermal comfort at high ACH values were marginally affected by changes introduced in the envelope heat flow dynamics through retrofitting, there were other key parameters such as DAV maximum MAA, distance of maximum MAA point from patient’s face and indoor heat gain, which were considerably influenced by changes in the envelope heat transfer characteristics. These insights would be instrumental in prospective research where the indoor domain could be modelled with simplified or detailed wall boundary conditions (by incorporating/neglecting transient conjugate heat transfer through the envelope) depending upon which particular attributes of the indoor environment are targeted for control.

The comprehensive analysis of diverse design strategies revealed the following important findings pertaining to the healthcare ward design.

- Air curtain created by the floor mounted central diffuser prevents the movement of air from one occupant to the other, thereby mitigating the infection cross-transmission risk.
- The previously recommended high ACH values for similar infrastructure could be relaxed to some extent by combining with other suitable design interventions.
- Increasing the rate of fresh air supply (or ACH) does not always ensure a reduction in the stale air (with high MAA value) accumulation in all the regions of a ward. In fact, the reverse trend can be noticed for inadequately planned designs.
- Stale air tends to accumulate near the central roof region for all the retrofitting options considered. Placement of the outlet grille near the roof along the roof central line is an effective measure to vent out the accumulated stale air. Placement of the patient’s head towards the rear wall with outlet grille is a good practice as stale air tends to accumulate over the legs of patient but not directly over the patient’s head owing to the presence of the outlet grille on the rear wall.
- Effect of envelope retrofitting is marginal on indoor thermal comfort at high air supply rates. For all the retrofitting strategies, the air accumulated near the roof tends to exceed the indoor thermal comfort limit. However, the central core region of the interior domain near the resting patients shows acceptable thermal comfort conditions.
- High cooling energy supply (by high supply air ACH and/or low supply air temperature) to the ward could lead to higher amount of indoor heat gain, thus leading to a process which is more energy expensive and less productive in terms of useful energy content of air for possible cogeneration purposes.
- Deploying polystyrene as thermal insulation on ward envelope could be a lucrative retrofitting measure to minimize indoor heat gain. The indoor heat gain for a polystyrene insulation embedded envelope for the given ward is less by an amount of 6971.97 kJ/day relative to an envelope without retrofit and 2393.82 kJ/day relative to an envelope with ZNH PCM retrofit (Table 5). Also, for the final optimal case, the use of a lower ACH value (=8 ACH) than the maximum limit considered (=12 ACH) needs to be stressed upon. Furthermore, out of the two inlet air temperatures considered near the human thermal comfort limits, the higher value (−26 °C) is found to be more suitable, thereby reducing the cooling energy requirement for the ward. The overall shrink size of the ward can curtail the land requirement for setting up new infrastructure and lead to a commensurate reduction in construction material, labour, cost and power requirements. This design can therefore be adopted for the studied climatic condition as a resource cum energy efficient solution to develop modular healthcare wards while maintaining occupant’s health safety and indoor thermal comfort.

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Appendix A. Supplementary data

Supplementary data to this article can be found online at https://doi.org/10.1016/j.jobe.2021.103296.

Nomenclature

| Symbol | Description |
|--------|-------------|
| $\text{abs}$ | Solar radiation absorptivity of material |
| $c_p$ | Generalized specific heat (J/kg-K) |
| $c_{ps}$ | Specific heat of solid (J/kg-K) |
| $c_{pl}$ | Specific heat of liquid (J/kg-K) |
| $C_1, C_2$ | Model constant ($C_1 = 1.44$, $C_2 = 1.92$) |
| $E$ | Total energy (J) |
| $\text{ems}$ | Long wavelength emissivity of material |
| $G_b$ | Buoyancy generated turbulent kinetic energy (W/m$^3$) |
| $G_k$ | Generation of turbulent kinetic energy by mean velocity gradient (W/m$^3$) |
| $g$ | Acceleration due to gravity (m/s$^2$) |
| $H$ | Enthalpy (J) |
| $h_b$ | Latent heat of fusion (J/kg) |
| $I$ | Ambient temperature (K) |
| $k$ | Inlet air temperature ($T_{\text{inlet}}$) |
| $k_{\text{eff}}$ | Liquidus temperature (K) |
| $LH$ | Solidus temperature (K) |
| $m_a$ | Sol-air temperature (K) |
| $\Delta t_{\text{diurnal cycle}}$ | Duration of diurnal cycle (86400 s) |
| $\mu_t$ | Eddy viscosity (m$^2$/s) |
| $\nu$ | Velocity (m/s) |
| $Y_M$ | Contribution of fluctuating dilatation in compressible turbulence towards overall dissipation rate (W/m$^3$) |
| $\beta$ | Thermal expansion coefficient of air (K$^{-1}$) |
| $\epsilon$ | Turbulent dissipation rate (J/kg-s) |
| $\rho$ | Density (kg/m$^3$) |
| $\rho_0$ | Reference density for the Boussinesq approximation (kg/m$^3$) |
| $\sigma_k, \sigma_\epsilon$ | Turbulent Prandtl number for $k$ and $\epsilon$ respectively |
| $\overline{\tau}$ | Stress tensor (Pa) |
| $\overline{\tau}_{\text{dev}}$ | Deviatoric stress tensor (Pa) |

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