Novel thermal treatment model to decontaminate airborne SARS Cov-2 virus for residential and commercial buildings

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Abstract
In the COVID-19 pandemic, control of airborne virus transmission is exceptionally challenging as it is attached to suspended particles in the air and stays for an extended time. Air contaminated with airborne viruses holds a substantial risk for household transmission. In this study, a novel thermal treatment system is modeled based on porous heating for the decontamination of airborne SARS-Cov-2. The model includes an air heating domain, insulated chamber, buffer tank and heat exchanger. The airborne SARS-Cov-2 is decontaminated when passing through a porous heat pipe and the insulated chamber for an anticipated dwelling period of more than 5 min at 105°C and further stored in a buffer tank for natural cooling. The obligatory decontaminated air is allowed in the residential space under ambient conditions passing through a heat exchanger. The numerical investigation of the porous pipe model at different $L/D$ ratios with altered porosities aims to establish the best-performing porous domain. Besides this, the buffer tank is intended to maintain buffer storage of the treated air and significant natural cooling before...
passing to the heat exchanger. A solar PV module is proposed to meet the prerequisite energy requirements of the equipped devices.

**KEYWORDS**
- computational analysis
- mathematical modeling
- porous domain
- SARS-CoV-2
- solar PV
- thermal treatment

## 1 | INTRODUCTION

In the ongoing COVID-19 pandemic, more than 200 countries have critically suffered from medical and financial crises amid the human life loss of 1.75 million till June 2021. COVID-19 was first reported in December 2019 instigated by Severe Acute Respiratory Syndrome Coronavirus 2 (SARS-CoV-2).\(^1\) Evidence-based studies illustrate the spread of the respiratory virus through direct contact, respiratory droplets, and airborne transmission.\(^2\)–\(^4\) Essential control measures are advised such as frequent hand washing and wearing a face mask to diminish the infection of coronavirus through direct contact and respiratory droplets (WHO, ICMR). However, the control of airborne virus transmission is exceptionally challenging as it is attached to suspended particles in the air and stays for a long period (≈3 h).\(^2\) Henceforth, the infection rate through airborne virus transmission is relatively rapid.\(^4\) A study found that the virus spreads through human respiratory activities of sneezing, coughing, and talking in terms of the number of droplets and travel velocity 10,000 (20 m/s), 100–1000 (10 m/s), and 50 (<5 m/s), respectively.\(^4\),\(^5\) Moreover, a lab-based study conducted in Wuhan hospital noticed that two recovering COVID-19 patients aged 71 and 81 years, released 7.35 to 7.77 × 10^4 viruses per hour through breathing.\(^6\)

It is rather challenging to maintain the preventive measures in a dense residential or commercial area. In the countries like India, highly dense residential and commercial buildings deal substantial risk of airborne virus transmission. Hence, the treatment of the surrounding air is essentially needed before entering the living space to mitigate the spread of coronavirus. It is evident from the existing literature that SARS-CoV-2 is extremely sensitive to high temperatures.\(^7\) It is highly stable at 4°C for 14 days, but the possibility of its survival declines at a higher temperature.\(^1\) The active period of the SARS-CoV-2 virus is reported 8–10 days at a temperature of 37°C, but it can be destroyed at a temperature > 90°C in 15 min.\(^8\) Although a few studies\(^8\),\(^9\)–\(^10\) have revealed the experimental data of temperature and time for the inactivation of SARS-CoV-2, the threshold temperature and time to destroy the virus is still the point of concern.

### 1.1 Temperature-dependent inactivation of SARS-CoV-2

A comprehensive study is carried out to ascertain the required temperature and time for the inactivation of SARS-CoV-2 based on available evidential data in the literature. Table 1 illustrates the critical inactivation temperature and time of SARS-CoV-2, which significantly advocates the decontamination of airborne coronavirus through thermal treatment for a certain extent of time. In this context, Yap et al.\(^11\) adopted the dry heating process greater than 70°C
temperature. It was observed that all kinds of viruses, including SARS-CoV-2, were inactivated. Chen et al. originated the thermal inactivation of SARS-CoV-2 RNA using autoclaving at 121°C, boiling at 100°C, and heating at 80°C for 20 min in each and observed no viral RNA after autoclaving and boiling process.

The time-dependent thermal treatment of airborne SARS-CoV-2 is adequately supported in the discussed literature adopting certainly available methodologies. Nevertheless, there exists tremendous scope to address more effective and rapid thermal treatment practices for large spaces. Porous materials (PM) have been used in a wide range of heating applications for the augmentation of fluid’s thermal performance. The PM ensures proper mixing by offering a higher surface-to-volume ratio to enhance the heat transfer between solid and gas phases. However, the pressure drop inside the porous zone demands more pumping power which can be suppressed considering the optimum porosity of PM. PM’s performance and efficacy are critically dependent on thermophysical properties; coefficient of thermal expansion, thermal conductivity, melting point, heat capacity, porosity, pore size, and area density. Researchers have extensively used metal foams, porous sintered metal, and ceramics in porous-based air heating equipment. Abo-Elfadl et al. have investigated the PM-based air heating in a tubular solar air heater at variable mass flow conditions and noticed 8.9% and 8.2% improvement in energy and exergy efficiencies of the tubular air heater. In the same context, Hernandez et al. conducted a numerical analysis of a solar air heating collector with a porous steel matrix, and the air distribution in the collector, and the heat transfer characteristics were improved. Moreover, in an experimental study by Watanabe et al., thermo-physical characteristics of dried air (200°C) were investigated in a sintering fibrous porous tube. The gathered outcomes reveal an increased rate of heat exchange in a porous tube in comparison to a conventional heating tube.

In the milieu of energy sources, solar PV/thermal established their potential utility in the domestic and industrial applications as a replacement of conventional energy with zero carbon formation. Solar-powered installations are drastically growing in India due to adequate solar potential reaching 40.09 GW (March 2021). Solar PV publicized its utility in numerous applications like lighting, pumping, air conditioning, space heating, water heating, and air heating. On the flip side, solar thermal collectors (PTC) are also recommended for air heating systems. However, the nonoperational phase at solar intermittent fluctuation and nighttime is

| Temperature (°C) | Dwelling period (min) | References          |
|------------------|-----------------------|---------------------|
| 55               | 10                    | Wang et al.         |
| 60               | 10.5                  | Yap et al.          |
| 65               | 15                    | Kampf et al.        |
| 70               | 2.5                   | Yap et al.          |
| 80               | ≤1                    | Kampf et al. and Yap et al. |
| 90               | <1                    | Yap et al.          |
| 92               | 15                    | Kushan et al.       |
| 95               | 1                     | Burton et al.       |
| 100              | < 5                   | Kampf et al.        |

### Table 1

Inactivation temperature of SARS-CoV-2 along with the dwelling period.
unbearable in this concern. Besides plenty of literature available on SARS-Cov-2 decontamination, limited literature evidence the thermal treatment of airborne coronavirus\textsuperscript{4,13} for continuous air treatment and supply to large spaces like households and commercial places. Although Burton et al.\textsuperscript{12} have performed the heat treatment of SARS-Cov-2 in a tissue culture medium using an unlidded dry, hot block at the temperature range of 56–95°C. The study noticed that the complete inactivation of the virus at the temperature of 80°C and 95°C takes place for the duration of 90 and 5 min, respectively. Still, no study is found revealing the thermal treatment of surrounding air infected with airborne SARS-Cov-2. Therefore, it is an immense requirement to develop a system for heating the air to inactivate the airborne SARS-Cov-2 and supply the treated air continuously for the living space.

In the present study, a porous-based novel air thermal treatment system is modeled and designed to decontaminate airborne SARS-Cov-2. The surrounding air near containment areas or hospitals gets contaminated with the virus, which is responsible for the virus transmission in the household. The contaminated air with airborne SARS-Cov-2 is passed through the porous pipe and heated to a maximum temperature of 105°C. It is further passed to an insulated chamber, ensuring a minimum anticipated dwelling period of 5 min to destroy the virus. The treated air is passed to the buffer storage tank (BT), which cools the treated air to a certain extent and acts as the reservoir to assure the uninterrupted supply for the living space. The obligatory decontaminated air is provisioned to living space at the atmospheric condition after passing through a heat exchanger (HE). Solar PV is recommended as an energy source for the system’s functioning. However, the direct AC source must be imposed to resume the system’s functioning under the intermittent solar fluctuation of cloudy weather, rain, and nighttime operation.

2 | THERMAL TREATMENT MODEL

A detailed schematic of the model illustrated in Figure 1 comprises a porous domain, insulated chamber, buffer storage tank, heat exchanger, and solar PV module, along with essential medicated appliances. The anticipated model is intended to ensure a comfortable thermal condition $T_{\text{air,out}}$ of decontaminated inflowing air to the house. In the schematic, $T_1$, $T_2$, $T_3$: $T_a$, $T_{\text{air,out}}$, $T^*$ and $P_1$, $P_2$, $P_3$: $P_a$, $P_{\text{air,out}}$, $P^*$, where $T^*$ and $P^*$ symbolize the air decontamination criterion.

2.1 | Process description

The air infected with SARC-Cov-2 around the residential area passes through a thermal treatment system before entering the house, obeying the process loop presented in Figure 2. The requisite ventilation air for residential space is calculated using the equation; $\dot{V}_{\text{tot}} = N\dot{V}_p + A\dot{V}_b$, where $\dot{V}_{\text{tot}}$ (L/s) is the total ventilation rate, $N$ is the number of occupants, and $A$ (m$^2$) is the space area. However, $\dot{V}_p$ (L/s × person) is the airflow rate for diluting emissions from occupants, and $\dot{V}_b$ (L/s × m$^2$) is the airflow rate for diluting emissions due to the building component.\textsuperscript{26} Hence, the present model is proposed for a residential house of 120 m$^2$ of a family (5 persons), considering $\dot{V}_p$ and $\dot{V}_b$ as 2.6 and 0.16, respectively.
Air at elevated pressure $P_3$ (6 bars) moves toward the porous air heater through an air filter and surge tank. An air filter is recommended to eliminate the dust particles and moisture content at the entrance. However, the surge tank is provisioned to regulate the pressure transit to establish uniform flow stability of air in the porous domain. Afterward, the air is impelled toward a cylindrical SiC-based porous domain to decontaminate the virus at a constant, elevated temperature of 105°C. The electrical heaters are wrapped to maintain the consistent desired temperature of the porous domain to ascertain uniform heating of the contaminated air.

**Figure 1** Schematic layout of thermal treatment model.

**Figure 2** Process loop obeying thermal treatment of the contaminated air.
Furthermore, the high-temperature air resides for an explicit period in an insulated chamber before entering the buffer tank. The inclusive travel volume in the porous domain and the insulated tank is assured of achieving the desired dwelling period of ≥5 min for the treated air. However, the air pressure retained sufficiently high to minimize the insulated chamber's volume and improve the heat capacity of residing air. The treated air arrives in the buffer tank depending on stored air pressure \((P_3 > P_2)\) during the system operation. Subsequently, the required mass of treated air arrives in the residential space passing through an air cooler (HE) to obtain comfortable thermal conditions \((P_1 \approx 1\text{ bar} \text{ and } T_1 \leq 30^\circ\text{C})\).

2.2 Specifications: Porous domain and insulated chamber

A cylindrical pipe of a certain length and diameter acts as a porous domain for the thermal treatment of contaminated air, shown in Figure 3. The candidate materials for the cylindrical pipe and porous domain are steel and silicon carbide, respectively. Thermal characterization of the airflow through a porous pipe is carried out at different L/D ratios and porosities, as illustrated in Table 2.

The insulated chamber is intended to accomplish the desired dwelling period for complete decontamination of air, as presented in Figure 4. The prerequisite air travel volume of 1.215 m\(^3\) inside the insulated chamber ascertains the prescribed dwelling period (≈5 min) for complete decontamination. The air travel volume implies an extended circular passage between four circular baffles (S) inside the insulated chamber. The entire chamber is well insulated with the glass fiber sheet and specific air gaps to minimize heat losses. The detailed specifications of the insulated chamber are summarized in Table 3.

2.3 Optimal sizing of the buffer storage tank and heat exchanger

The buffer tank sizing plays a vital role in the modeling and performance of the entire thermal treatment system. The optimal dimensions of the insulated chamber and heat exchanger solely

| Porous pipe | Diameter (m) | Length (m) | (L/D) ratio | Porosity (\(\varepsilon\)) |
|-------------|--------------|------------|-------------|---------------------------|
| Case 1      | 0.12         | 1          | 8.33        | 0.9, 0.8, 0.7, 0.6, 0.5   |
| Case 2      | 0.10         | 1          | 10.0        | 0.9, 0.8, 0.7, 0.6, 0.5   |
| Case 3      | 0.8          | 1          | 12.5        | 0.9, 0.8, 0.7, 0.6, 0.5   |

FIGURE 3 Schematic of porous heating domain.
depend on the size of the buffer tank. Primarily, to subside the tank's size, treated air is considered to be stored at an elevated pressure of 5 bar and allowed to cool till thermal equilibrium is established. However, the buffer air stored in the tank ensures a continuous supply. The buffer tank helps in cooling the stored air by natural convection and reduces the cooling load on the cooler (heat exchanger). Tank sizing is carried out at the optimum aspect ratio ($H/D = 1.37$) to establish maximum cooling. Interestingly, the entire tank’s volume is separated by an inner annular cylindrical strip to avoid the direct flow of hot air toward the heat exchanger.

The air-cooled heat exchanger is intended to acquire the desired cooling of the buffer tank’s outflowing air to comfortable room conditions ($P_1 = 1$ atm, $T_1 \leq 30^\circ$C). The gross surface area of the finned tube heat exchanger is determined based on the steady-state temperature of air outflow (50.05°C) from the buffer tank. Depending on the weather conditions, a natural or forced convection heat exchanger may be adopted to cool the treated air. However, during warmer environmental conditions, a household's air-conditioning system may ensure the required thermal conditions of the treated air. The heat exchanger specifications are particularized in Table 4.

### Table 3: Specifications of insulated chamber.

| Insulated chamber                     |
|--------------------------------------|
| Material                             | Stainless steel |
| Absolute pressure (bars)             | 6              |
| Capacity (m$^3$)                     | 1.21           |
| Dimensions (chamber)                 |
| Diameter (m)                         | 1.12           |
| Height (m)                           | 1.2            |
| Dimensions (round baffles)           |
| Number of circular baffles           | 4              |
| Thickness (m)                        | 0.002          |
| Height (m)                           | 1              |
The present study recommends the solar photovoltaic panel to function in the air thermal treatment system. The inclusive thermal treatment model is equipped with electrically operated air heaters and a pneumatic pump for continuous system operation. The ultimate power requirements for the electrical heaters (H1 and H2) and pneumatic pumps (100 PSI) are approximately 2500 and 500 W, respectively. The indispensable panel area is intended considering the typical operating characteristics of solar PV listed in Table 5.

### Table 4  Specifications: Buffer storage tank and heat exchanger.

| Buffer tank                  |                        |
|------------------------------|------------------------|
| Material                     | Stainless steel        |
| Maximum pressure (bars)      | 5.5                    |
| Air storage capacity, normalized to 1 atm (m³) | 23.65                |
| Dimensions (tank)            |                        |
| Diameter (m)                 | 1.6                    |
| Height (m)                   | 2.2                    |

| Heat exchanger               |                        |
|------------------------------|------------------------|
| Type of cooling              | Natural convection     |
| Material                     | Stainless steel        |
| Maximum \( m_{\text{air}} \) (kg/s) | 0.036               |
| Maximum air pressure (bars)  | 1.5                    |
| Maximum temperature (°C) (fluid in) | 50                  |
| Convective coefficient \( h_1, h_0 \) (W/m²K) | 137, 4.7            |

| Dimensions                   |                        |
|------------------------------|------------------------|
| Tube Surface area (m²)       | 2.98                   |
| Tube length (L) and diameter (D) (m) | 50, and 0.019 |
| Fins Surface area (m²)       | 1.85                   |

### 2.4 Solar PV module

The present study recommends the solar photovoltaic panel to function in the air thermal treatment system. The inclusive thermal treatment model is equipped with electrically operated air heaters and a pneumatic pump for continuous system operation. The ultimate power requirements for the electrical heaters (H1 and H2) and pneumatic pumps (100 PSI) are approximately 2500 and 500 W, respectively. The indispensable panel area is intended considering the typical operating characteristics of solar PV listed in Table 5.

### 3 Mathematical Modeling

An inclusive mathematical model establishes optimum sizing of the entire air thermal treatment system for a five adults residential space. The essential governing equations related to the porous domain, insulated chamber, buffer storage tank, and heat exchanger are explained extensively. However, the optimum sizing and performance analysis of the entire system is carried out on the ground of certain assumptions and as follows:

(a) Air is considered a perfect gas.
(b) The preferred materials are isotropic, homogeneous, and have persistent thermal characteristics.
3.1 | Essential governing equations

3.1.1 | Porous pipe

3.1.1.1 | Mass conservation

\[
\frac{\partial (\rho \varnothing)}{\partial \tau} + \frac{\partial}{\partial x_i}(\rho u_i) = 0. \tag{1}
\]

3.1.1.2 | Momentum conservation

\[
\frac{\partial}{\partial \tau}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = \frac{\partial p}{\partial x_i} + \frac{\partial \sigma'_{ij}}{\partial x_j} + \rho g \delta_{ij} + F_i, \tag{2}
\]

where, in Equation (2), the stress tensor \(\sigma'_{ij}\) is as follows:

\[
\sigma'_{ij} = \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu \left( \frac{\partial u_k}{\partial x_k} \right) \delta_{ij}. \tag{3}
\]

In Equations (1)–(3), the velocity is determined using Dupuit Forchheimer equation \(u = \varnothing V\), where \(V\) stands for intrinsic velocity.\(^{15}\) Equation (2) is characterized in the \(i^{th}\) direction for an inertial reference frame, where \(u_i\) and \(u_j\) are the \(i^{th}\) and \(j^{th}\) components of the instantaneous velocity, respectively. Moreover, \(p\), \(\delta_{ij}\), and \(F_i\) are the static pressure, the Kronecker delta operator, and external body force in the \(i^{th}\) direction. In Equation (3), \(\sigma'_{ij}\), \(g\), and \(\mu\) stand for stress tensor, gravitational body force, and dynamic viscosity.\(^{21}\)
3.1.1.3 | Energy conservation equation

\[
\frac{\partial}{\partial t}(\rho e) + \frac{\partial}{\partial x_i}[u_i(\rho e + p)] = \frac{\partial}{\partial x_i}\left[\lambda_{\text{eff}}\frac{\partial T}{\partial x_i} - \sum_{j}h_{j}J_{j} + u_{j}(\sigma'_{ij})_{\text{eff}}\right] + S_{h}. \tag{4}
\]

In Equation (4), \(e\), \(\varnothing\) and \(h\) are the specific total energy, porosity, and sensible enthalpy. However, \(\lambda_{\text{eff}}\) refers to the effective conductivity, that is, \(\varnothing\lambda_{f} + (1 - \varnothing)\lambda_{s}\). Furthermore, \(J_{j}\) refers to diffusion flux of \(j\)th species, and \(S_{h}\) denotes supplementary volumetric heat source which is deserted in this case.\(^{14,18}\)

3.1.1.4 | Pressure drops in the porous domain

\[
\int_{P_{1}}^{P_{2}}dP = \int_{x=0}^{L}\left[\alpha_{1}\frac{(1 - \varnothing)^{2}\mu}{\varnothing^{2}d_{p}^{2}}(u_{i}) + \alpha_{2}\frac{(1 - \varnothing)\rho}{\varnothing^{2}d_{p}}(u_{i}^{2})\right]dx. \tag{5}
\]

Ergun\(^{29}\) established an empirical relation for the pressure drop (\(\Delta P\)) of a compressible fluid flow through a porous domain as stated in Equation (5), where the constants \(\alpha_{1}\) and \(\alpha_{2}\) are the factors for viscous and form drag portion of pressure drop, respectively. Moreover, \(x\) and \(D_{p}\) are the displacement and mean particle diameter in meters.\(^{32,33}\)

3.1.1.5 | Nondimensional parameters

Reynolds number

\[
Re_{p} = \frac{vd_{p}}{(1 - \varnothing)\nu}. \tag{6}
\]

Nusselt number and Peclet number

\[
Nu_{D} = \frac{hD}{\lambda_{\text{fluid}}} = 1.015Pe_{D}^{1/2}, \tag{7}
\]

\[
Pe_{D} = \frac{v \cdot D}{\alpha_{m}}. \tag{8}
\]

The Reynolds number of airstream in the porous domain is determined through Equation (6) depending on pore diameter (\(d_{p}\)).\(^{32}\) However, the heat convection coefficient is appraised using Equations (7) and (8), where \(Pe_{D}\) stands for Peclet number. In Equation (8), \(\alpha_{m}\) is the mean thermal diffusivity of PM and follow: \(\alpha_{m} = \lambda_{\text{eff}}/(\rho C_{p}).\(^{33}\)

3.1.2 | Insulated chamber

3.1.2.6 | Air travel passage

\[
V_{T} = \dot{V}_{a}\tau_{R}. \tag{9}
\]
The inclusive air travel passage in the insulated chamber is intended concerning the required residence period \( \tau_R (s) \) of the treated air using Equation (9), where \( V_a (m^3/s) \) denotes air volume flow rate.

### 3.1.2.7 Heat losses

\[
\dot{Q}_L = \frac{(T_{O,C} - T_0)}{\sum_{j=1}^{3} R_j}.
\]  
(10)

In Equation (10), \( T_{O,C} \) and \( T_0 \) stand for chamber’s outer periphery temperature (afore insulation) and dead state temperature (300 K), respectively. \( \sum_{j=1}^{4} R_j = R_1 + R_2 + R_3 \), where \( R_1 \), \( R_3 \), and \( R_2 \) are the thermal resistance offered by consecutive layers of insulations AG (0.04 m) and GF (0.05 m) and stainless-steel sleeve (0.002 m), respectively. The conductive thermal resistances are as follows:

\[
R_1 = \frac{\ln \frac{r_2}{r_1}}{(2\pi \cdot w\cdot \lambda_{AG})}, \quad R_2 = \frac{\ln \frac{r_3}{r_2}}{(2\pi \cdot w_{SS}\cdot \lambda_{SS})}, \quad \text{and} \quad R_3 = \frac{\ln \frac{r_4}{r_3}}{(2\pi \cdot w_{GF}\cdot \lambda_{GF})}.
\]

### 3.1.3 Buffer storage tank

#### 3.1.3.8 Generalized mass balance equation

\[
\sum \dot{m}_{air,in} = \sum \dot{m}_{air,out}
\]  
(11)

Equation (11) represents the buffer tank’s mass balance during continual system operation where \( \dot{m}_{air,in} \) and \( \dot{m}_{air,out} \) are the mass flow (kg/s) of incoming and leaving air in the storage tank. However, at the initialization, the \( \dot{m}_{air,out} = 0 \) and the insulated chamber’s treated air is stored until the requisite air pressure is attained.\(^{27}\)

#### 3.1.3.9 Generalized energy balance equation

\[
\sum \dot{Q}_{in} - \sum \dot{Q}_{out} = \sum \dot{Q}_{BT}.
\]  
(12)

The buffer tank’s energy balance is illustrated in Equation (12), where, \( \dot{Q}_{in} \) (W) and \( \dot{Q}_{out} \) (W) signify the energy of inflowing and leaving air in the control volume (BT). However, \( \dot{Q}_{BT} \) (W) refers to the rate of energy accumulated in the BT’s control volume.\(^{14,34}\) The explicit terms are expressed in Equations (13) to (18) and as follows:

\[
\dot{Q}_{in} = \dot{Q}_{air,in} = \int_{T_{\infty}}^{T_{air,in}} \dot{m}_{air,in} C_{p_a} dT,
\]  
(13)

\[
\sum Q_{out} = \dot{Q}_{air,out} + \dot{Q}_{losses},
\]  
(14)

\[
\dot{Q}_{air,out} = \int_{T_{\infty}}^{T_{air,out}} \dot{m}_{air,out} C_{p_a} dT,
\]  
(15)

\[
\dot{Q}_{losses} = (\dot{q}_{Convection} + \dot{q}_{Radiation}) A_{BT} = \left\{ h(T_{BT} - T_{\infty}) + \epsilon\sigma\left( T_s^4 - T_{\infty}^4 \right) \right\} A_{BT},
\]  
(16)
\[
\dot{Q}_{BT} = \frac{\partial}{\partial \tau} Q_{Ac},
\]
(17)

\[
Q_{Ac} = m_{air} C_{p\text{,air}} (\bar{T}_{BT} - T_{BT,i}).
\]
(18)

In Equations (13) and (15), \(T_{\infty}\), \(T_{air,\text{in}}\), and \(T_{air,\text{out}}\) are designated as the ambient temperature, inflowing and outflowing air temperature (°C), respectively. The CV’s leaving energy (\(\dot{Q}_{out}\)) comprises the energy of outflowing air and energy losses from the tank’s bare area \((A_{BT} = 2\pi rh + \pi r^2)\) via natural convection and radiation. In Equations (16) and (18), \(\bar{T}_{BT}\) and \(T_{BT,i}\) are the mean temperature and initial temperature of buffer storage tank (°C), and \(\bar{h}, \bar{\varepsilon}\), and \(\sigma\) are the average convection coefficient \((\text{W m}^{-2} \text{K}^{-1})\), surface emissivity, and Stefan-Boltzmann constant \((5.67 \times 10^{-8} \text{W m}^{-2} \text{K}^{-4})\), respectively. \(Q_{Ac} \text{ (J)}\) is the energy accumulated in the BT’s control volume.

The continuous cooling of the buffer tank is predominantly governed by natural convection heat loss, as illustrated in Equation (16). Furthermore, the heat transfer coefficient for a cylindrical buffer tank is determined through the average Nusselt number \((\bar{h} = \frac{2}{\lambda} \bar{N}u_D)\). Churchill and Chu\(^3\) have anticipated an explicit correlation for the \(\bar{N}u_D\) at \((Ra_D) \leq 10^{12}\) and as follows:

\[
\bar{N}u_D = \left[ 0.60 + \frac{0.387 \ Ra_D^{0.16}}{1 + (0.559/Pr)^{0.562} \sigma^{0.296}} \right]^2,
\]
(19)

\[
Ra_D = \frac{\beta g (T_S - T_{\infty})D^3}{\alpha \nu},
\]
(20)

where, \(Ra_D\) and \(Pr\) are the Rayleigh number and Prandtl number, respectively. The \(Ra_D\) is calculated using the Equation (20) where, \(\beta, \alpha,\) and \(\nu\) stand for expansion coefficient \((\text{K}^{-1})\), thermal diffusivity \((\text{m}^2/\text{s})\) and kinematic viscosity \((\text{m}^2/\text{s})\), respectively.

3.1.4 | Heat exchanger

3.1.4.10 | Effectiveness-NTU approach

\[
\varepsilon = f (NTU, \zeta).
\]
(21)

The heat exchanger’s effectiveness (\(\varepsilon\)) is primarily reliant on a nondimensional parameter \((NTU)\) and heat capacity ratio \((\zeta = \frac{C_{\text{out}}}{C_{\text{in}}}\)). Although a rational definition of HE’s effectiveness is the ratio of actual heat transfer to the maximum possible heat transfer as illustrated in Equation (22).\(^3\)

\[
\varepsilon = \frac{Q}{Q_{\text{Max}}} = \frac{C_h \left( T_{\text{in}} - T_{\text{out}} \right)}{C_{\text{in}} \left( T_{\text{in}} - T_{\text{cin}} \right)},
\]
(22)
\[\text{NTU} = -\ln\left[1 + \left(\frac{1}{\xi}\right)\ln(1 - \varepsilon \xi)\right]. \quad (23)\]

The HE's effectiveness is determined using Equation (22), where, \(C_h\) is the hot fluid's heat capacity. However, \(C_{\text{min}}\) signifies smaller hot or cold heat capacity \(mC_p\). Further, the nondimensional entity NTU is quantified using Equation (23).\(^{35,36}\)

### 3.1.4.11 Gross surface area

\[A_{\text{HE}} = \frac{\text{NTU} C_{\text{min}}}{U}. \quad (24)\]

The heat exchanger's gross surface area is intended through Equation (24) where \(U\) is the overall heat transfer coefficient of exposed surface area and as follows:

\[U_0 = \left[\frac{1}{h_0} + R'_{f_o} + \frac{1}{h_i}\right]^{-1}. \quad (25)\]

In Equation (25), \(h\) and \(R_f\) refer to the heat convection coefficient \((\text{W m}^{-2} \text{K}^{-1})\) and fouling factor \((0.0002) \ (\text{m}^2 \text{KW}^{-1})\) at the outer \((o)\) and inner side \((i)\) of the HE's tube.\(^{36}\) The heat convection coefficients for the inner and outer sides of the heat exchanger tube are determined using the Dittus–Boelter and Hilpert correlations owing to forced convection heat transfer.\(^{37}\)

\[h_i = \frac{k}{d_i}\left[0.023R^{0.8}_{\text{D}} Pr^{0.3}\right] \left(\text{Re}_D \geq 2300\right), \quad (26)\]

\[h_o = \frac{k}{d_o}\left[0.193R^{0.618}_{D} Pr^{1/3}\right], \quad \left(4 \times 10^3 \leq \text{Re}_D \leq 4 \times 10^4\right). \quad (27)\]

At elevated air temperature reliant on the buffer tank's thermal condition or warm climate conditions, the forced convection cooling is recommended, and convection coefficient \((h_o)\) is suitably determined using Equation (27). However, the convection coefficient can be obtained using Equations (19) and (20) for natural convection cooling. In the present study, the heat exchanger's optimum dimensions are calculated considering maximum air outflow temperature from the buffer tank and natural convection cooling.

### 3.1.4.12 Pressure drops and friction factor

\[\Delta P = \int_{x=0}^{L} f \left(\frac{\rho u d_i}{2}\right) dx, \quad (28)\]

\[f = (0.790 \ln \text{Re}_D - 1.64)^{-2} (3 \times 10^3 \leq \text{Re}_D \leq 5 \times 10^6). \quad (29)\]

Equation (28) estimates the major pressure losses in the heat exchanger tube based on the friction factor \((f)\) and \(\text{Re}_D\) for a turbulent flow as stated in Equation (29).\(^{37}\) However, the minor losses are deserted.
3.1.5 | Solar PV modeling

The solar PV modeling aims to figure out the approximate size of the solar PV module to meet the ultimate energy demand of the air thermal treatment system for ongoing operation. The gross energy $E_{Re}$ (W) sums up the energy required for the electrical heaters and pumping system, as illustrated in Equation (30).

3.1.5.13 | Gross energy required

\[ E_{Re} = E_{H1} + E_{H2} + E_p. \]  

(30)

$E_{H}$ and $E_p$ represent obligatory energy demand (W) for electrical heaters and pumps, respectively. However, $E_{H1} \approx F_c \sum Q_{PP}$, signifies energy supplied to the heater ($H1$) for the porous domain’s preheating and incessant air heating during the system operation, where compensation factor ($F_c$) 1.12 encounters energy losses from the porous pipe.

\[ \sum Q_{PP} = \int_{T_{\infty}}^{T^*} \dot{m}_{\text{air,in}} C_p dT \]  

(31)

In Equation (31), $\Delta T$ is the temperature rise ($T^* - T_{\infty}$). However, $T^*$ is the maximum decontamination temperature. The $E_{H2}$ (W) is the fragmented energy required for a substitute heater ($H2$) to compensate for the energy losses from the insulated chamber ($\dot{Q}_l$) as stated in Equation (10).

3.1.5.14 | Available solar energy

\[ E_{\text{solar}} = A_{PV} I_R. \]  

(32)

3.1.5.15 | Solar PV energy output

\[ E_{PV} = F_f (V_{o,c} \times I_{s,c}), \]  

(33)

\[ F_f = \frac{(V_{o,c} \times I_{s,c})_{\text{max}}}{(V_{o,c} \times I_{s,c})_{\text{act}}}. \]  

(34)

3.1.5.16 | Electrical efficiency

\[ \eta_{el} = \frac{F_f (V_{o,c} \times I_{s,c})}{A_{PV} I_R}. \]  

(35)

$A_{PV}$, $F_f$, and $\eta_{el}$ are the gross solar PV area (m$^2$), mean solar radiant energy (W/m$^2$), fill factor, and electrical efficiency, respectively$^{30,31}$ as stated in Equations (32)–(35). However, $V_{o,c}$ and $I_{s,c}$ represent open circuit voltage and short circuit current.

3.1.5.17 | Electrical power

\[ E_{el} = \eta_{inv} \{F_f (V_{o,c} \times I_{s,c})\}. \]  

(36)
The electrical power (W) indicates inverter output energy accessible to meet the gross energy requirement \( E_{Re} \) as illustrated in Equation (36), where, \( \eta_{inv} \) is the specific inverter efficiency.\(^{29}\)

3.1.5.18 | Solar PV panel area

\[
A_{PV} = \frac{E_{Re}}{\eta_{inv} \eta_{d} I_{R}},
\]  

(37)

The indispensable panel area approximated using Equation (37) is predominantly reliant on available solar irradiation at the local condition (Calcutta, India), where the numerator and denominator are in W and W/m².

4 | COMPUTATIONAL SECTION

The geometric modeling and extended numerical analysis of the porous pipe model and buffer storage tank are carried out using a commercial ANSYS 19.2 module. The requisite computational scheme, solution methodology, and stipulated boundary conditions for both models are discussed consecutively.

4.1 | Computational scheme for porous pipe analysis

The 3D model for porous pipe with all three different \( \frac{L}{D} \) ratios (\( \frac{L}{D} \): 8.33, 10, 12.5) are prepared, meshed, and numerically investigated using ANSYS Workbench and Fluent, respectively, as illustrated in Figure 5. The presence of porous media best suits enhanced heat transfer between solid and fluid phases.\(^{38,39}\) In the present study, heating of high-pressure air passing through a porous pipe domain with various porosities (\( \varnothing \)): 0.5, 0.6, 0.7, 0.8, and 0.9, are extensively studied. However, the explicit governing equations of mass, momentum, and energy for a porous domain are revealed in Equations (1)–(3). The optimized or structured tetrahedral mesh was attained by a grid-independent test for the porous pipe model at an \( \frac{L}{D} \) ratio and porosity of 8.33 and 0.9, respectively. Four different meshes (G1–G4) along with flexible element sizes 1 mm to 4 mm are examined. The inspected temperature consequences for the above conditions attained by G3 and G4 are found with an error of 1.23%. Hence, the

FIGURE 5  Three-dimensional model of porous pipe with cross-section view displaying mesh feature
present study considers optimal mesh G4 of 4 mm element size and 814980 cells for the computational analysis.

However, a similar approach continues for the remaining porosities of the particular case and forthcoming cases. The computational study of porous pipe model with different $L/D$ ratios is initiated by establishing requisite models, materials, cell zone conditions, and anticipated boundary conditions. A realizable $k$-$\varepsilon$ turbulence model is considered with amended empirical constant $C_{\varepsilon,1} = 1.6$. Moreover, the porous zone is taken in cell zone conditions considering the porosities ($\varnothing$): 0.5, 0.6, 0.7, 0.8, and 0.9, respectively. The viscous resistance ($1/\alpha''$) and inertial resistance ($\beta''$) in the porous zone are quantified by Equations (38)–(40).

$$\frac{1}{\alpha''} = \left[ \frac{D_p}{160} \times \frac{\varnothing^3}{(1 - \varnothing)^2} \times S^2 \right]^{-1}. \quad (38)$$

The Horkheimer's equation for inertial resistance:

$$-\frac{dP}{dz} = u_k \frac{\mu}{K} + \beta' u_k |u_k|, \quad (39)$$

$$\beta' = \frac{3.5}{D_p} \times \frac{1-\varnothing^2}{\varnothing^3} \times \frac{1}{S}, \quad (40)$$

where $D_p$, $S$, $K$, and $u_k$ refer to pore diameter, sphericity, permeability, and the velocity component along the flow direction, respectively. The viscous resistance (m$^{-2}$), inertial resistance (m$^{-1}$), and heat transfer coefficient (W/m$^2$K) for air are calculated considering 2 mm pore diameter and 0.8 sphericity in the porous zone and listed in Table 6.

### 4.2 Computational scheme for buffer storage tank

A 3D model of the buffer storage tank is prepared in Workbench and Fluent (ANSYS 19.2), shown in Figure 6. A fully unstructured tetrahedral meshing with an adaptive refinement factor of 2 is considered for meshing the buffer tank with enhanced meshing density near the inlet. The nonuniform grids with moderate computational cost are sufficient to capture the grid resolutions of flow parameters in the entire computational domain.

The thermal characterization of airflow in the buffer tank is executed through a computational approach. The explicit governing equations of mass, momentum, and energy

| Porosity ($\varnothing$) | $1/\alpha$ | $\beta$ | $h$ |
|------------------------|-----------|---------|-----|
|                        | L/D: 8.33 | 10      | 12.5|
| 0.5                    | 750,000.0 | 506.78  | 19.18|
| 0.6                    | 370,370.4 | 1538.08 | 19.54|
| 0.7                    | 185,860.1 | 3717.20 | 19.89|
| 0.8                    | 87,890.6  | 8641.97 | 20.25|
| 0.9                    | 32,578.8  | 21,000  | 20.59|
presented in Equations (1)–(3) are adopted considering $\bar{\varnothing} = 1$ (nonporous domain). The grid independence test is performed for five meshes (G1–G5) with the element size varying from 2 to 8 mm by examining the tank’s mean temperature. Consequently, the optimum grid G4 with a 6 mm element size and 8,567,329 cells is preferred for the simulation with an average error of 1.57%. The computational analysis of the buffer storage tank is accomplished by establishing requisite models, materials, cell zone conditions, and anticipated boundary conditions. A realizable $k$–$\varepsilon$ turbulence model is considered with amended empirical constant $C_{\varepsilon 1} = 1.6$.

### 4.3 Solution methodology

The transient numerical analysis of porous pipe and buffer storage tank models are performed by adopting anticipated boundary conditions using a commercial multigrid solver on a high-performance computational facility. For both cases, the Favre averaged differential equations for mass, momentum, and energy are integrated over a control volume for solving a set of equations. However, the infinite volume method is perceived to obtain algebraic equations solved iteratively by stipulated boundary conditions. The SIMPLE method is adapted as a solution method for pressure velocity coupling owing to the turbulent flow of fluids, while the second-order upwind scheme is considered to discretize the convective terms spatially. Moreover, the second-order implicit (SOI) scheme is used for transient formulation. The calculations were performed at a lower time step size of 0.01 s in all cases of the porous pipe model. However, a variable time step size ranging from 0.01 to 0.1 s was adapted for the buffer tank’s simulations. The thermophysical properties of steel, silicon carbide, and air are listed in Table 7.

### Table 7 Thermodynamic properties of candidate materials $^{14,38}$

| Properties                      | Stainless steel | Silicon carbide | Air       |
|---------------------------------|-----------------|-----------------|-----------|
| Density (kg/m$^3$)              | 8000            | 3210            | 1.22      |
| Specific heat (J/kg K)          | 500             | 750             | 1006.43   |
| Thermal conductivity (W/m K)    | 16.5            | 120             | 0.02      |
4.4 | Boundary conditions

The numerical analysis of the porous pipe model at different $L/D$ ratios and buffer storage tanks is accomplished by adopting stipulated boundary conditions, shown in Table 8. The entire porous domain is preheated at zero mass flow rate and isothermal wall condition before allowing the airflow in the porous pipe domain. Subsequently, the air is allowed to flow in the porous domain at a particular mass flow rate.

4.5 | Model validation

The present numerical model is duly validated with the experimental outcomes of Baragh et al. and analytical results calculated through the modified Ergun equation stated in Equation (5). Baragh et al. conducted a series of experiments on airflow through a porous test section with a diameter and length of 10 and 70 cm, respectively. In the test section, PM was

| Boundary Specifications Credentials |
|-----------------------------------|
| **TABLE 8** Stipulated boundary conditions for porous pipe and buffer storage tank model. |

### Porous pipe

**Inlet**
- **Air**
  - Boundary type: Mass flow inlet
  - Mass flow rate (kg/s): 0.036
  - Temperature (°C): 27

**Outlet**
- **Air**
  - Boundary type: Pressure outlet
  - Gauge pressure (Pa): 600,000

**Pipe wall**
- Boundary type: Wall
  - Material: Stainless steel
  - Outer wall: Isothermal
  - Temperature (°C): 160

### Buffer storage tank

**Inlet**
- **Air**
  - Boundary type: Mass flow inlet
  - Mass flow rate (kg/s): 0.036
  - Temperature (°C): 105

**Outlet**
- **Air**
  - Boundary type: Pressure outlet
  - Gauge pressure (Pa): 550,000

**Outer wall**
- Boundary type: Wall
  - Material: Stainless steel
  - Outer wall: Mixed
  - Convective coefficient (W/m² K): 4.1
  - External emissivity: 0.9
  - Free stream temperature (°C): 27
inserted at $X/L$ 0.2 to 0.63, where $X$ and $L$ denote porous channel distance from the inlet and the test section’s gross length, respectively. The aluminum-wired mesh porous media of 0.9 porosity with a different inner and outer diameter configuration is considered for the experiment at the Reynolds numbers 1125, 3400, and 6437. Out of all experimental models, model 4 of filled porous zone with porous media ($X/L$: 0.2–0.63) is considered for the validation at operating parameters: Reynolds No. = 3400, porosity ($\varnothing$) = 0.9, permeability ($K$) $6.99 \times 10^{-6}$ (m$^2$), and constant heat flux ($q''$) = 275 W, respectively.41 A computational model analogous to the referred case is investigated by implementing the prescribed computational methodology and boundary conditions. The numerical consequences of wall temperature (°C) at different $X/L$ are compared and illustrated in Figure 7. The numerical results are validated well with the experimental outcomes for the entire range of $X/L$. However, a moderate deviation is observed at the trailing end of the test section ($X/L > 0.5$). A maximum deviation of 2.32% is noticed between the experimental and numerical outcomes at $X/L = 0.65$.

In addition, the present numerical results of $\Delta p/L$ were also compared with analytical outcomes obtained from the modified Ergun Equation (Equation 5), for the $L/D$ ratio of 8.33, at various porosities ($\varnothing$): 0.5, 0.6, 0.7, 0.8, and 0.9. The numerical and analytical outcomes are validated well, shown in Figure 7B, with the maximum allowable error of 10.12% at the porosity of 0.6. The numerical results slightly differ from experimental and analytical data since the inclusive differences may occur due to the errors in testing devices, calibration of observations, and respected assumptions for organizing the mathematical and numerical model. Hence, the current numerical model was found equitably reliable for an extended investigation of porous pipe domain and buffer storage tank.

5 | RESULTS AND DISCUSSION

In the present study, a comprehensive computational analysis aims to determine the optimal $L/D$ ratio and porosity for significant thermal dispersion of air in the porous domain to abolish SARS-Cov-2 at the desired operating conditions. Furthermore, the transient computational
analysis of the buffer storage tank is also executed to maximize the heat loss to the environment and minimize the air temperature at the exit of the buffer tank. Significant outcomes are obtained from a comprehensive numerical investigation of the porous pipe model and buffer storage tank adopting stipulated boundary conditions.

5.1 | Porous heating domain

5.1.1 | Effect of porosity and $L/D$ ratio on temperature

The temperature variations across the pipe length are illustrated in Figures 8 and 9 for different $L/D$ ratios of 8.33, 10, and 12.5, at various porosities ($\varnothing$): 0.5, 0.6, 0.7, 0.8, and 0.9, respectively. The surface average temperatures are noted across the pipe length at 0m (inlet), 0.25, 0.50, 0.75, and 1m (outlet). The present study extensively investigates the effect of altered porosity and variable pipe $L/D$ ratio on air temperature. The mass flow rate of the air is maintained constant at 0.036 kg/s, and the heating system operates at 6 bars.

In Case 1 ($L/D = 8.33$), a noticeable change in air’s outlet temperature is observed as 102.2°C, 102.8°C, 103.5°C, 103.9°C, and 105.1°C at $\varnothing$: 0.5, 0.6, 0.7, 0.8, and 0.9, respectively. Likewise, the outlet temperature appears 103.1°C, 103.7°C, 104.1°C, 104.8°C, and 103.8°C, for Case 2 ($L/D = 10$) and 103.7°C, 104.4°C, 104.9°C, 104.4°C, and 104.2°C, for Case 3 ($L/D = 12.5$), at $\varnothing$: 0.5, 0.6, 0.7, 0.8 and 0.9, respectively. Hence, for the $L/D$ ratios 8.33, 10, and 12.5, the maximum outlet temperature is observed at the porosities 0.9, 0.8, and 0.7, respectively, which signifies the fluid’s optimum thermal dispersion and stay period in the porous domain.

The computational results witness a polynomial fit between pipe outlet temperature and porosity follows $T_{p,o} = C_1\varnothing^2 + C_2\varnothing + C_3$, where the parametric constants $C_1$, $C_2$, $C_3$: $-28.5$, 51.2, 81 ($L/D = 8.33$, Figure 8A), $-17.14$, 26.4, 94 ($L/D = 10$, Figure 8B), and $-19.2$, 27.5, 94 ($L/D = 12.5$, Figure 9A), respectively.

The cross-sectional temperature contours across the pipe length are established in Figures 10–12 for Case 1 ($L/D = 8.33$), Case 2 ($L/D = 10$), and Case 2 ($L/D = 12.5$), respectively.
In Figure 10, the cross-sectional color contours demonstrate the temperature rise from pipe inlet to outlet at ∅: 0.5, 0.6, 0.7, 0.8, and 0.9. The colored contours evidence progressive temperature escalation toward the pipe outlet at the altered porosities in each case. However, the temperature gradually increases from the pipe center to the wall due to a significant velocity gradient. Therefore, the temperature contours for the $L/D$ ratio and porosity of 8.33 (∅ = 0.9) and 12.5 (∅ = 0.9) reveal a rapid temperature upsurge of air in the porous domain with a maximum outlet temperature of 105.1°C and 104.9°C, respectively.

5.1.2 | Effect of porosity and $L/D$ ratio on pressure drop

Furthermore, the effect of altered porosity on the pressure drop of air across the porous pipe is observed and shown in Figure 13. Extreme pressure drop is witnessed at lower porosity 0.5 compared to higher porosity 0.9 for any $L/D$ ratio of the pipe and found significantly reliant on the pressure variation trends with porosity as stated in Ergun equation Equation (5). Maximum pressure losses across the pipe length are noted as 60.6, 125.5, and 307 kPa for the $L/D$ ratios of 8.33, 10, and 12.5, respectively, at lower porosity of 0.5. However, the pressure drop at higher porosity (0.9) has a marginal difference in all the cases. Consequently, the pressure drop across the pipe length ($\Delta p/L$) reveals an apparent exponential behavior with porosity as $\Delta p/L = C_4 e^{C_5 \varnothing}$, where the explicit parametric constants “$C_4$” and “$C_5$” are 5442 and −8.97 at $L/D$: 8.33, 26.176, and −10.37 at $L/D$:10 and 28,803 and −9.02 at $L/D$:12.5, respectively.

5.1.3 | Nondimensional parametric analysis

The Reynolds number ($Re$) and Nusselt number ($Nu$) for different $L/D$ ratios are critically analyzed at the porosities 0.5 to 0.9 and presented in Figure 14A,B. The $Re$ of the air in the porous domain
dominates at higher porosity and $L/D$ ratio. However, the $Re$ significantly diminishes toward the lower porosity due to a consistent reduction in the seepage velocity. It is observed that when the porosity is varied from 0.5 to 0.9, the $Re$ is increased from 6369 to 11,464; 7643 to 13,757; 9554 to 17,197 for different $L/D$ ratios of 8.33, 10.0, and 12.5, respectively. Moreover, the $Nu$ is quantified based on the Peclet Number ($Pe$) for different porosities and $L/D$ ratios. The trends of $Nu$ are relatively analogous to the $Re$, as illustrated in Figure 14. The $Nu$ dominates at elevated porosity 0.9 as 79.74, 87.35, and 97.66 for different $L/D$ ratios of 8.33, 10, and 12.5, respectively. The trends of obtained results are as follows: $Re \propto \Phi \cdot \frac{L}{D}$, and $Nu \propto \Phi \cdot \frac{L}{D}$. 

**FIGURE 10** Cross-sectional temperature contours of the porous domain for $L/D$ 8.33 at pipe length 0–1 m.
It can be noted from Figures 9B and 13 that the \( L/D \) ratio and porosity of 8.33 and 0.9 retain a maximum pipe outlet temperature of 105.1°C with a marginal pressure drop of 1.6 kPa. Hence, the \( L/D \) ratio and porosity of 8.33 and 0.9 found an optimal case to achieve the maximum thermal dispersion of air in the porous domain.

### 5.2 Buffer storage tank

A buffer storage tank intended for storage and considerable cooling of the treated air is critically analyzed using a time-dependent computational approach. The overall thermal analysis of the buffer storage tank comprises air outflow temperature \( T_{\text{air,out}} \), outer wall temperature \( T_{\text{wall}} \), volume average temperature \( T_{\text{mean}} \), and energy balance in the control volume. The computational analysis commences by assuming the tank occupied the treated air.
at $P_2 \approx 5.5$ bars and $27^\circ C$. Besides, the foremost objective of buffer tank (BT) analysis is to recognize the steady-state air outflow temperature to optimize the sizing of the heat exchanger.

5.2.1 Time-dependent thermal characterization

The time-dependent thermal characterization reveals instantaneous thermal dispersion of the inflowing air inside the buffer tank. A close approximation of the buffer tank's temperature contours is presented in Figure 15 at various time frames. Before the numerical simulation commences, the BT's control volume exists in thermodynamic equilibrium with the surrounding. The entire tank volume is segregated by a circular baffle integrated at the top to avoid direct flow toward the outlet and ensure proper air mixing.
The thin annular zone between the baffle plate and the cylindrical surface helps air cooling before entering the heat exchanger. The inlet temperature of the air in the BT tank is \( \sim 105°C \). The timeframe contours reveal a significant upsurge in \( T_{\text{mean}} \) at an early stage. The treated air gradually disperses in a confined region and spreads throughout the inner core. However, considerable heat dispersal has not been noticed in the separated area escaping direct flow toward the outlet. Afterward, thermal dispersion exaggerates in the separated passage. Here, the sliced graphics establish uniform temperature distribution in the divided region in a critical time lag of 5 h and appear steady beyond this period.

Figure 16A reveals the maximum \( T_{\text{mean}}, T_{\text{air, out}}, \) and \( T_{\text{wall}} \) as 69.45°C, 50.05°C, and 36.09°C, respectively, at 5 h and holds firmly stable beyond this period. However, a lesser \( T_{\text{wall}} \) of 36.09°C
FIGURE 15  Instantaneous temperature contours of buffer tank's cross-section at different time intervals.
signifies magnificent and prerequisite buffer tank cooling. Interestingly, the steady-state air outflow temperature appears moderate. Hence, a sophisticated and small heat exchanger device is sufficient to cool the air further and supply it to the household.

Figure 16B and 17 illustrate time-dependent the energy scenario of buffer tank control volume with steady-state occurrence. The $\dot{Q}_{\text{air,in}}$, $\dot{Q}_{\text{losses}}$, and $\dot{Q}_{\text{air,out}}$ noticed 2265, 1468, and 696 W, respectively, at the end of 5-h and remain steady with continuous system operation. Primarily, $\dot{Q}_{\text{air,in}}$ dominates at the inner core region separated by the baffle plate, resulting in lesser and insignificant heat dispersion in the buffer tank's annular passage. However, $\dot{Q}_{\text{losses}}$ and $\dot{Q}_{\text{air,out}}$ increase with time (hours) and attain maximum values as the steady-state condition is achieved. A higher $\dot{Q}_{\text{losses}}$ witnesses rapid and prerequisite cooling of the buffer storage tank through natural convection as a massive surface area ($13 \text{ m}^2$) is exposed to the environment. Consequently, the maximum $\dot{Q}_{\text{air,out}}$ is not exceeding much due to substantial heat losses from
control volume. Hence, the following consequences significantly support the optimal buffer storage tank modeling. Besides this, the maximum accumulated energy reaches 1359 kJ, corresponding to $T_{mean}$ 50.05°C, as shown in Figure 17.

6 | CONCLUSIONS

In the present study, a porous-based novel air thermal treatment system is modeled for the decontamination of airborne SARS-Cov-2. The airborne SARS Cov-2 can be decontaminated, passing through a porous air heater and an insulated chamber at > 105°C, ensuring the anticipated dwelling period >5 min. The obligatory decontaminated air is provisioned to the residential living space at the atmospheric condition after passing through a cooling system. The geometric modeling and extended numerical analysis of the porous pipe model and buffer storage tank are carried out using a commercial module of ANSYS 19.2. The existing numerical model is duly validated with the experimental outcomes of Baragh et al.\textsuperscript{41} and analytically calculated data using a modified Ergun equation for compressible fluids stated in Equation (5) with acceptable agreements.

A comprehensive computational study aims to obtain optimal $L/D$ ratio and porosity for significant thermal dispersion of air in the porous domain to abolish SARS-Cov-2. The linear temperature variations across the heated porous pipe at different axial locations are observed for various $L/D$ ratios of the porous pipe with different porosities ($\varnothing$) of 0.5, 0.6, 0.7, 0.8, and 0.9. The mass flow rate of the air is maintained constant at 0.0009 kg/s and enters at a temperature and pressure of 27°C and 6 bars. Interestingly, the findings witness a polynomial fit between pipe outlet temperature and porosity follows $T_{p,o} = C_1\varnothing^2 + C_2\varnothing + C_3$. Consequently, the optimal porosity and $L/D$ ratio of 0.9 and 8.33 are observed to obtain the air’s maximum temperature (105.1°C) at the porous pipe outlet. It is observed that the pressure drop across the pipe length increases with increasing the $L/D$ ratio for a constant porosity. The pressure drops across the pipe length ($\Delta p/L$) reveal an apparent exponential behavior with porosity following a relation: $\Delta p/L = C_4 e^{C_5\varnothing}$. The $Re$ of the mean stream velocity in the porous domain increases with increased porosity. However, the trends of $Nu$ are relatively analogous to the $Re$. The trends are as follows: $Re \propto \varnothing, \frac{L}{D}$, and $Nu \propto \varnothing, \frac{L}{D}$.

A buffer storage tank intended for buffer storage and considerable cooling of the treated air is critically analyzed using a time-dependent computational approach. A time-dependent analysis determines the steady-state consequences of BT’s associated temperature and energy balance. The maximum temperature of outflowing air from the buffer tank is observed to obtain an optimal heat exchanger size. A critical period of 5 h characterizes the minimum time lag between system initialization and steady-state condition. The maximum $T_{mean}$, $T_{air,out}$, and $T_{wall}$ were noticed as 69.45°C, 50.05°C, and 36.09°C, respectively, ending 5-h time lag and holds firmly stable beyond this period. However, the lesser $T_{wall}$ and $T_{air,out}$ signifies magnificent and prerequisite buffer tank’s cooling. Besides, the stored air of 23.65 m$^3$ at 1 ATM can meet the fluctuating demand for ventilation air.

An appropriate heat exchanger with a surface area of 4.83 m$^2$ is established between the residential space and the buffer tank. The heat exchanger can cool the tank’s outlet air of ($T_{air,out}$) 50.05°C to room temperature (27°C) by natural convection mode of heat transfer. However, an enhanced buffer tank’s cooling may not necessitate the heat exchanger during rainy and cold weather conditions. A solar PV module with a gross area of 28 m$^2$ is proposed to meet the prerequisite energy requirements of all equipped devices.
NOMENCLATURE

VARIABLES

As  surface area (m$^2$)
C  heat capacity (J/K)
D  diameter (m)
$d_p$  particle diameter (m)
$F_C$  compensation factor
$F_f$  fill factor
$h$  heat convective coefficient (W/m$^2$K)
L  length (m)
P  pressure (bar)
$P_t$  Prandtl number
$\dot{Q}$  heat (W)
$q''$  heat flux (W/m$^2$)
R  thermal resistance (K/W)
T  temperature (°C)
U  overall heat transfer coefficient (W/m$^2$K)

GREEK SYMBOLS

$\rho$  density (kg/m$^3$)
$\eta$  efficiency
$\mu$  dynamic viscosity (N Sec/m$^2$)
$\tau$  time (s)
$\bar{\eta}$  efficiency
$\phi$  porosity
$\varepsilon$  effectiveness
$\alpha$  thermal diffusivity (m$^2$/s)
$\lambda$  thermal conductivity (W/mK)
$\nu$  kinematic viscosity (m$^2$/s)

SUBSCRIPTS/SUPERSCRIPT

*  decontamination
$s$  surface
$\infty$  ambient
$Ac$  accumulated
$el$  electrical
$eff$  effective
$i$  inlet
$o$  outlet

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DATA AVAILABILITY STATEMENT
Data sharing not applicable to this article as no data sets were generated or analyzed during the current study.

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