



# Article Enhancing Air Quality for Embedded Hospital Germicidal Lamps

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Abstract: The indoor air of a hospital is always full of bacteria and viruses due to patients with different diseases. These bacteria and viruses could be highly infectious to the people in the hospital irrespective of their health conditions, and could be hazardous to the patients, their care takers, and hospital staff. Thus, keeping a good hospital air quality is very essential to the operation of the hospital. This study aims at enhancing ventilation of the interior lighting of hospitals with germicidal capabilities. Air disinfection is accomplished by adding the specially designed disinfecting filters and fans to existing embedded lamps in the hospitals. The embedded lamp has a square shape of 601 mm in width and 112 mm in thickness. In the design stage, the air flow inside the embedded lamp with the added filters and fans was investigated by numerical simulation using a computational fluid dynamics (CFD) tool. Three designs, referred to as Types 1, 2, and 3, were evaluated using steady-state CFD flow simulations. The ventilation rate of the Type 1 design was about 251.9 CMH, and 348.3 CMH for the Type 2 design by increasing the fan outlet area. However, even though the ventilation was increased by 34%, the flow field of the Type 2 design was not uniform, resulting in flows being circulated around the side locations. Thus, the Type 3 design further treats this aspect by streamlining the outlet geometry and adding flow guiding vanes to reduce flow resistance and flow unsteadiness; the corresponding air ventilation rate reached 376.3 CMH. Hence, the Type 3 design was fabricated and tested. The test results confirm that the design not only has a higher ventilation rate but also operates under a smaller pressure drop, thus accomplishing the goal of providing good air quality in the hospital environment efficiently. Moreover, the associated flow noise is reduced by about 8 dBA. Hence, both an increase in the air ventilation rate and a reduction of noise are achieved simultaneously by the present method.

Keywords: germicidal embedded lamp; CFD simulation; forced convection; hospital indoor air

# 1. Introduction

In the year 2020, the world was troubled and threatened by the outbreak and spread of SARS-CoV-2 (COVID-19) virus. There is no sign of this threat being diminishing yet. Therefore, there is an urgent need to prevent its further spreading because it has multiple transmission pathways, as pointed out by Kenarkoohi et al. [1]. This is especially critical in hospitals, mainly for two reasons. One is that the hospitals are the places for treating COVID-19 patients while other patients who are typically more vulnerable are also present. The other is that the recirculating air conditioning environments of the hospitals can make matters worse due to the presence of airborne viruses. In other words, preventing the airborne transmission of viruses in hospitals is a necessity based on national and international evidence [1]. By identifying the dynamics of the causative agent of SARS-CoV-2, Masoumbeigi et al. [2] showed the significance of the airborne infection potential of the related epidemics. Their results indicated the importance of the hospital's indoor air quality monitoring system. Typically, a hospital's recirculating environment, if not properly treated, can lead to sick building syndrome symptoms and hospital workers may increase the risk of having those symptoms as shown by Babaoglu et al. [3]. Veysi et al. [4] evaluated

the factors affecting the indoor air quality and their effect on the respiratory health of staff in a busy Iranian hospital. The results showed that, compared to office staff, nurses had a higher risk of respiratory symptoms caused by the accumulated indoor pollutants in the hospital environment due to failure of the hospital ventilation systems. Cabo et al. [5] also pointed out that poor hospital indoor air quality might lead to hospital-acquired infections, sick hospital syndrome, and various occupational hazards. The hospital environment tends to have microbes that can deteriorate air quality and affect people's health due to the co-existence of various kinds of viruses. The deterioration of air quality depends on the working areas and the air conditioning systems of the hospital, as shown by Jung et al. [6]. Moreover, the ventilation efficiency of the air conditioning system is related to the indoor air chemical and microbiological concentration levels in the hospitals, the outdoor air quality, and the relative locations of the air inlets and outlets, as pointed out by Baures et al. [7], Chamseddine et al. [8], and Liu et al. [9], respectively.

Improving the air quality, such as by filtering or sterilization, is essential to mitigate the above mentioned situations. Vijayan et al. [10] showed that air filtering could reduce particulate matter and allergens to prevent disease spreading. Bache et al. [11] revealed that a high-intensity light of 405 nm in wavelength mounted on the ceiling was effective at killing bacteria. The effectiveness of these approaches can be optimized by using the Internet of Things (IoTs), as pointed out by Saini et al. [12], to facilitate real-time air quality monitoring by sensors and network connectivity. This kind of IoTs approach also has the potential for effective energy utilization because the environments can be monitored and tuned more easily.

There is no doubt that controlling hospital air quality is essential and the air flow pattern plays an important role in this regard, in addition to air filtering and air sterilization by light. On the other hand, the renovation of existing hospital ventilation is always costly and time-consuming. Hence, numerical simulations are a viable alternative before the renovation is actually implemented. As examples, Karuchit and Warissarangkul [13] used numerical simulation to investigate the optimum design and operation of germicidal irradiation tuberculosis isolation rooms in Thailand's community hospitals. They showed that the optimum efficiency of 98.19% (16% better than the standard design) could be achieved by installing a UV light of 16 W at a height of 3.3 m above the floor and with an air exchange rate of 6  $m^3/h$ . Zhou et al. [14] demonstrated numerically that improper air ventilation could lead to cross-infection due to air transmission in hospitals. Following these approaches, in this study, germicidal filters were combined with fans and installed in hospital lighting to improve the air quality of hospitals. The effectiveness of three designs was examined using flow simulations to obtain the largest air flow rate under the condition of a fixed fan speed before the actual implementation of the design. Then, the design that showed good performance was implemented and tested. Thus, both air quality and fan energy-saving can be achieved simultaneously and more economically. Details of the simulation approach, results and discussion, and conclusions are presented as follows.

## 2. Simulation Approach

In the following sub-sections, the governing equations, lamp and fan geometries, material properties, and the simulation step are described.

# 2.1. Governing Equations

The governing equations adopted for the numerical simulations are the Reynolds averaged Navier–Stokes momentum equations with k- $\varepsilon$  turbulent modelling [15]. The analysis mode was set to the mean steady-state condition. Briefly, the three-dimensional Navier–Stokes momentum equations of the mean flow in Cartesian coordinates (x, y, and z) are illustrated in Equations (1) to (3).

(a) x-direction momentum equation

$$\rho \frac{\partial U}{\partial t} + \rho U \frac{\partial U}{\partial x} + \rho V \frac{\partial U}{\partial y} + \rho W \frac{\partial U}{\partial z}$$

$$= \frac{\partial P}{\partial x} + \frac{\partial}{\partial x} \left[ 2(\mu + \mu_t) \frac{\partial U}{\partial x} \right] + \frac{\partial}{\partial y} \left[ (\mu + \mu_t) \left( \frac{\partial U}{\partial y} + \frac{\partial V}{\partial x} \right) \right] + \frac{\partial}{\partial z} \left[ (\mu + \mu_t) \left( \frac{\partial U}{\partial z} + \frac{\partial W}{\partial x} \right) \right]$$
(1)

(b) y-direction momentum equation

$$\rho \frac{\partial V}{\partial t} + \rho U \frac{\partial V}{\partial x} + \rho V \frac{\partial V}{\partial y} + \rho W \frac{\partial V}{\partial z}$$

$$= \frac{\partial P}{\partial y} + \frac{\partial}{\partial x} \left[ (\mu + \mu_t) \left( \frac{\partial U}{\partial y} + \frac{\partial V}{\partial x} \right) \right] + \frac{\partial}{\partial y} \left[ 2(\mu + \mu_t) \frac{\partial V}{\partial y} \right] + \frac{\partial}{\partial z} \left[ (\mu + \mu_t) \left( \frac{\partial V}{\partial z} + \frac{\partial W}{\partial y} \right) \right]$$
(2)

(c) z-direction momentum equation

$$\rho \frac{\partial W}{\partial t} + \rho U \frac{\partial W}{\partial x} + \rho V \frac{\partial W}{\partial y} + \rho W \frac{\partial W}{\partial z}$$

$$= \frac{\partial P}{\partial z} + \frac{\partial}{\partial x} \left[ (\mu + \mu_t) \left( \frac{\partial U}{\partial z} + \frac{\partial W}{\partial x} \right) \right] + \frac{\partial}{\partial y} \left[ (\mu + \mu_t) \left( \frac{\partial V}{\partial z} + \frac{\partial W}{\partial y} \right) \right] + \frac{\partial}{\partial z} \left[ 2(\mu + \mu_t) \frac{\partial W}{\partial z} \right]$$
(3)

where *U*, *V*, and *W* are the mean flow velocities in the x, y, and z directions, respectively; *P* is the mean flow pressure. And  $\rho$ ,  $\mu$ , and  $\mu_t$  denote the fluid density, dynamic viscosity, and turbulent eddy viscosity, respectively. The turbulent eddy viscosity was obtained from the two-equation k- $\varepsilon$  turbulent model by Equation (4).

$$\mu_t = C_\mu \rho \frac{k^2}{\varepsilon} \tag{4}$$

That is,  $\mu_t$  is calculated from the turbulent kinetic energy (*k*) and the turbulent energy dissipation rate ( $\varepsilon$ ) which are determined by their corresponding transport equations, as given by Equations (5) and (6).

$$\rho \frac{\partial k}{\partial t} + \rho U \frac{\partial k}{\partial x} + \rho V \frac{\partial k}{\partial y} + \rho W \frac{\partial k}{\partial z} = \frac{\partial}{\partial x} \left[ \left( \frac{\mu_t}{\sigma_K} \right) \frac{\partial k}{\partial x} \right] + \frac{\partial}{\partial y} \left[ \left( \frac{\mu_t}{\sigma_K} \right) \frac{\partial k}{\partial y} \right] + \frac{\partial}{\partial z} \left[ \left( \frac{\mu_t}{\sigma_K} \right) \frac{\partial k}{\partial z} \right] - \rho \varepsilon + \mu_t \left[ 2 \left( \frac{\partial U}{\partial x} \right)^2 + 2 \left( \frac{\partial V}{\partial y} \right)^2 + 2 \left( \frac{\partial W}{\partial z} \right)^2 + \left( \frac{\partial U}{\partial y} + \frac{\partial V}{\partial x} \right)^2 + \left( \frac{\partial U}{\partial z} + \frac{\partial W}{\partial x} \right)^2 + \left( \frac{\partial V}{\partial z} + \frac{\partial W}{\partial y} \right)^2 \right]$$
(5)

$$\rho \frac{\partial \varepsilon}{\partial t} + \rho U \frac{\partial \varepsilon}{\partial x} + \rho V \frac{\partial \varepsilon}{\partial y} + \rho W \frac{\partial \varepsilon}{\partial z} = \frac{\partial}{\partial x} \left[ \left( \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x} \right] + \frac{\partial}{\partial y} \left[ \left( \frac{\mu_t}{\sigma_K} \right) \frac{\partial \varepsilon}{\partial y} \right] + \frac{\partial}{\partial z} \left[ \left( \frac{\mu_t}{\sigma_K} \right) \frac{\partial \varepsilon}{\partial z} \right] - C_2 \rho \frac{\varepsilon^2}{k} + C_1 \mu_t \frac{\varepsilon}{K} \left[ 2 \left( \frac{\partial U}{\partial x} \right)^2 + 2 \left( \frac{\partial V}{\partial y} \right)^2 + 2 \left( \frac{\partial W}{\partial z} \right)^2 + \left( \frac{\partial U}{\partial y} + \frac{\partial V}{\partial x} \right)^2 + \left( \frac{\partial U}{\partial z} + \frac{\partial W}{\partial x} \right)^2 + \left( \frac{\partial V}{\partial z} + \frac{\partial W}{\partial y} \right)^2 \right]$$
(6)

The values of the empirical constants  $C_{\mu}$ ,  $C_1$ , and  $C_2$  and the turbulent Prandtl number adopted were 0.09, 1.44, 1.92 and 1.0, respectively. By coupling with the mean flow continuity equation, given by Equation (7), these seven equations were used to obtain the seven unknowns of U, V, W, P,  $\mu_t$ , k, and  $\varepsilon$ .

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho U}{\partial x} + \frac{\partial \rho V}{\partial y} + \frac{\partial \rho W}{\partial z} = 0$$
(7)

#### 2.2. Lamp and Fan Geometries

The geometry of the embedded hospital germicidal lamp module is illustrated in Figure 1a. The lamp has a square geometry of 601 mm by 601 mm by 112 mm (thickness). Its interior components are shown in Figure 1b which include the front cover, air filters, flat light panel, flow channel, fan, and rear cover. The air filters are installed inside the front cover at the inlets of the air flow.



Figure 1. Embedded hospital germicidal lamp module.

Three types of flow channels, as shown in Figure 2 and represented by Types 1, 2, and 3, were simulated numerically. Type 1 was a standard design; Type 2 was modified from Type 1 by enlarging the channel size with round edges; Type 3 was a modification of Type 2 by further enlarging the channel size and also adding flow deflectors to reduce flow resistance.



Figure 2. Lamp module of three different flow channels.

## 2.3. Material Properties

The material properties of the solid plastic and air needed for the simulation are listed in Table 1. These values are fixed in this study.

# Table 1. Material Property.

| Item          | Unit              | Air                 | Plastic  |
|---------------|-------------------|---------------------|----------|
| Density       | kg/m <sup>3</sup> | 1.2047              | 432.54   |
| Viscosity     | Pa-s              | $1.817	imes10^{-5}$ | —        |
| Conductivity  | W/m-K             | 0.02563             | 0.259843 |
| Specific Heat | J/kg-K            | 1004                | 837      |

## 2.4. Simulation Steps

In the simulation, the environment was set to the temperature of 25 °C and relative humanity of 50%. The operation conditions of the fan were 110 V, 0.58 A, and 1000 rpm for the voltage, current, and rotation speed, respectively. The fan and air filter conditions were taken from the manufacturers' specifications. Steady-state simulations were performed for the three types of channel configurations illustrated in Figure 2. The finite element numerical simulation steps were as follows:

To save storage and computing time, the fan was simplified into a block geometry. The associated 3D drawing was then fed into the CFD software while imposing proper boundary conditions and related flow parameters.

(2) Boundary Conditions

The gauge pressures of the boundary conditions of the air inlets and outlets displayed in Figure 3 were set to zero.



Figure 3. Air inlet and outlet boundaries.

## (3) Fan Performance Curve

The fan performance curve [16] from the manufacturer, shown in Figure 4, was inputted to the CFD software for simulation.



Figure 4. Fan Performance Curve [16].

(4) Filter Resistance Curve

The filter resistance curve of the single layer glass fiber air filter of AR103 from the manufacturer [17], as shown in Figure 5, was also inputted into the CFD software to provide the flow resistance of the filter.



Figure 5. Filter Resistance Curve [17].

## (5) Mesh and convergence test

The technique of boundary mesh enhancement by auto mesh was adopted for the computational mesh. The non-orthogonal mesh with three layers with a mesh size of 0.4 is shown in Figure 6. The convergence test is displayed in Figure 7 by setting the changes due to the mesh size being less than 5% for the last 20% of the total iterations explored for all the mean velocities, pressure, turbulent kinetic energy, and turbulent energy dissipation rate. The latter two also determined the convergence of the turbulent eddy viscosity. Details of finite element mesh and nodes are listed in Table 2, together with the empirical constants adopted in the k- $\varepsilon$  turbulent model.



Figure 6. Non-orthogonal finite element mesh.



Figure 7. Convergence situation.

|  | Constant          |                 | Value   |  |
|--|-------------------|-----------------|---|--|
| k-ε Model  | $C_{\mu}$         |                 | 0.09  |  |
|  | $C_1$             |                 | 1.44  |  |
|  | $C_2$             |                 | 1.92  |  |
|  | $\sigma_K$        |                 | 1.0   |  |
| Boundary conditions                                    | Types 1, 2, and 3 | inlet<br>outlet | Pressure 0 Pa (Gage)  |  |
| Mesh conditions<br>(number of nodes and<br>elements) - | Type 1            | Fluid<br>Solid  | Nodes: 558,526; Elements: 1,618,093<br>Nodes: 39,475; Elements: 484,726 |  |
|  | Type 2            | Fluid<br>Solid  | Nodes: 355,004; Elements: 1,060,911<br>Nodes: 28,135; Elements: 304,958 |  |
|  | Type 3            | Fluid<br>Solid  | Nodes: 472,322; Elements: 1,520,467<br>Nodes: 31,239; Elements: 353,267 |  |

Table 2. Summary of mesh and boundary conditions.

#### 3. Results and Discussion

The simulation results of the three types of design are as follows.

#### 3.1. Velocity Distribution in the Flow Channel (Z-Plane)

The velocity distributions of the three types of flow channels are portrayed in Figure 8 for the bird-eye view (left) and the mid-plane (right) perpendicular to the Z-axis (i.e., the vertical direction at the height of 56 mm from the bottom) under a rotation speed of 1000 rpm. The air is drawn in from the inlets to the lamp center and then exits to the environment from the outlets through the air channels. Because of the swirl and non-uniform behaviors of the fan flow and the difference in flow resistance, the flow fields of the three types show different features. For Type 1, flow swirl at the lamp center is obvious and the velocity distribution is extremely skewed toward one direction (the diagonal from the upper-left corner to the lower-right corner), as shown in Figure 8a; i.e., the velocity distribution is very non-uniform. By enlarging the flow channel and using round edges for the Type 2 design, both the large swirl and skewness exhibited in the Type 1 design were reduced, but the non-uniformity of the flow field in the channels was still obvious, as shown in Figure 8b. Further modification of the flow channel by adding flow guides resulted in a more uniform velocity distribution, as evident from Figure 8c for the Type 3 design for which both the flow swirl and skewness were greatly reduced; the effectiveness of flow guides adopted in the Type 3 design is clear.

#### 3.2. Inlet/Outlet Velocity Distributions

The inlet/outlet velocity distributions for the three designs are shown in Figure 9. The inlet flows of the original design (Type 1), shown in Figure 9a, are uniform. However, the outlet flows are not uniform; one has larger velocities and the other has reverse flows. The difference between the maximum and minimum velocities is about 9.4 m/s. For Type 2, the outlet flow is better but still not very uniform, as shown in Figure 9b. The difference between the maximum and minimum velocities is about 5.8 m/s. For Type 3, both inlet and outlet flows are more uniform, as illustrated in Figure 9c, with the difference between the maximum and minimum velocities being about 2.5 m/s. For comparison, the overall inlet and outlet volumetric flow rates (ventilation rates) are summarized in Table 3. Three trends can be observed. Firstly, the inlet flow rates are uniform, irrespective of the design types, while Type 3 has a larger inlet flow rate compared to those of the other two. Secondly, similar to the uniformity of the outlet velocity distribution, the uniformity of the outlet flow rates is better for Type 3. The differences between the maximum and minimum flow rates are 146.3 m<sup>3</sup>/h (CMH), 66 CMH, and 12.8 CMH for Types 1, 2, and 3, respectively. Lastly, the volumetric outlet flow rates of Type 3 were increased by nearly 149.4%, taking those of Type 1 as a reference. For indoor ventilation, the uniformity of the flow field at the



outlets is essential. Hence, the design of Type 3 performs better in both flow uniformity and flow rate.

Figure 8. Velocity distributions of the lamp module with different flow channel.



Figure 9. Inlet and outlet velocity distributions of the lamp module.

| Item   |               | Type 1 | Type 2 | Type 3 |
|--------|---------------|--------|--------|--------|
| Inlet  | Inlet 1       | 64     | 87.3   | 94.8   |
|        | Inlet 2       | 63.7   | 87.3   | 93.5   |
|        | Inlet 3       | 62.1   | 86.9   | 95.5   |
|        | Inlet 4       | 62.1   | 86.8   | 92.5   |
|        | Sum           | 251.9  | 348.3  | 376.3  |
|        | Increase rate | 100%   | 138.3% | 149.4% |
| Outlet | Outlet 1      | 135    | 50     | 98.7   |
|        | Outlet 2      | -10.4  | 116    | 86.4   |
|        | Outlet 3      | 135.5  | 68.5   | 97.5   |
|        | Outlet 4      | -10.8  | 110.1  | 85.9   |
|        | Sum           | 249.3  | 344.6  | 368.5  |
|        | Increase rate | 100%   | 138.2% | 147.8% |

**Table 3.** Inlet/outlet volumetric flow rates (m<sup>3</sup>/h, CMH).

#### 3.3. Lamp Module Operation Point

The lamp module operation points are 252 CMH/264 Pa, 348 CMH/192 Pa, and 376.3 CMH/167 Pa, for Types 1, 2, and 3, respectively. That is, the corresponding pressure drops are 264 Pa, 192 Pa, and 167 Pa. In other words, the Type 3 design performs best with both a smaller pressure loss (about 63% of the original standard design) and a larger volumetric flow rate (about 149.4% of the original standard design, as shown in Figure 10 and Table 3. Combined with the results shown in Figure 9, the non-uniform and highly skewed air flow of the original standard design (Type 1) due to the fan and flow resistance are not only mitigated by the proper design of the flow guides (Type 3), but also increased volumetric flow rates at both inlets and outlets while achieving a smaller pressure drop. Namely, the Type 3 design performs more efficiently while achieving a larger volumetric flow rate. Therefore, Type 3 was fabricated and tested. The tested results confirmed the design with a smaller pressure drop.

As the Type 3 design is more energy efficient, it was expected that its noise level should also be lower. Hence, the fabricated Type 3 design was tested in a semi-anechoic chamber for its noise level. The results obtained at the location 1 m from the lamp are drawn in Figure 11. The noise at the operation point (376.3 CMH/167 Pa) is about 58 dBA, which is about 8 dBA smaller than that of the Type 1 design, confirming the expectation. It can also be observed that the noise level increases as the flow rate increases, as expected. Hence, if the noise level is of great concern, it is possible to use multiple fans (i.e., multiple

lamps) while running the fans at a smaller flow rate so that both the smaller noise level and the needed flow rate requirement can be accomplished simultaneously.



Figure 10. System operation point of the lamp module with various designs.





In addition to the indoor air quality of hospitals, which motivated this study, sickbuilding syndromes due to air quality can also arise in retirement homes, day care centers, classrooms, and shopping malls, just to name a few. For example, Kim et al. [18] examined the indoor air quality and its relationship with the symptoms of sick-building syndrome in various store types in underground shopping centers. They showed the occurrence of skin, irritation, and respiratory symptoms due to poor air quality. In other words, the sick-building syndrome is quite prevailing in modern society. The present approach of numerical simulations followed by actual validation can be extended readily to improve the indoor air quality of non-hospital environments. This is because the data needed for simulation are the material properties, fan P-Q (pressure-flow rate) curves, and filter resistance, which are all known in advance.

#### 4. Conclusions

In this study, steady-state finite element numerical simulations were conducted to examine the performance of three designs of the embedded hospital germicidal lamp module with air filters and various fan configurations. The governing equations were the Reynolds averaged Navier–Stokes momentum equations with the k- $\varepsilon$  turbulent model. In addition to the numerical simulations, the module that performed better was fabricated and tested. Key observations are as follows.

Firstly, the outlet flow velocity distributions of the original standard design (Type 1) are not uniform and are highly skewed due to the swirl flow of the fan and the larger flow resistance. Thus, the resulting outlet volumetric flow rate is smaller. Secondly, by proper design of flow guides (Type 3), the volumetric flow rates at both inlets and outlets can be increased by about 149.4%, in addition to achieving a more uniform flow velocity

distribution. Lastly, the improved design (Type 3) can be operated with a smaller pressure loss of about 63% while having a larger flow rate. Therefore, it can be operated more energy efficiently and also has a noise level of about 8dBA lower than that of the original design.

It can also be noted that the present approach can be readily extended to other nonhospital air environments of concern. This is because the data needed for simulation are all known in advance.

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## References

- Kenarkoohi, A.; Noorimotlagh, Z.; Falahi, S.; Amarloei, A.; Mirzaee, S.A.; Pakzad, I. Hospital indoor air quality monitoring for the detection of SARS-CoV-2 (COVID-19) virus. *Sci. Total Environ.* 2020, 748, 141324. [CrossRef]
- 2. Masoumbeigi, H.; Ghanizadeh, G.; Yousefi Arfaei, R.; Heydari, S.; Goodarzi, H.; Dorostkar Sari, R. Investigation of hospital indoor air quality for the presence of SARS-Cov-2. *J. Environ. Health Sci. Eng.* **2020**, *18*, 1259–1263. [CrossRef]
- 3. Babaoglu, U.T.; Milletli Sezgin, F.; Yag, F. Sick building symptoms among hospital workers associated with indoor air quality and personal factors. *Indoor Built Environ.* **2019**, *29*, 645–655. [CrossRef]
- 4. Veysi, R.; Heibati, B.; Jahangiri, M.; Kumar, P.; Latif, M.T.; Karimi, A. Indoor air quality-induced respiratory symptoms of a hospital staff in Iran. *Environ. Monit. Assess.* **2019**, *191*, 50. [CrossRef] [PubMed]
- 5. Cabo Verde, S.; Almeida, S.M.; Matos, J.; Guerreiro, D.; Meneses, M.; Faria, T. Microbiological assessment of indoor air quality at different hospital sites. *Res. Microbiol.* **2015**, *166*, 557–563. [CrossRef]
- 6. Jung, C.-C.; Wu, P.-C.; Tseng, C.-H.; Su, H.-J. Indoor air quality varies with ventilation types and working areas in hospitals. *Build. Environ.* **2015**, *85*, 190–195. [CrossRef]
- Baures, E.; Blanchard, O.; Mercier, F.; Surget, E.; Canna, P.; Rivier, A.; Gangneux, J.-P.; Florentin, A. Indoor air quality in two French hospitals: Measurement of chemical and microbiological contaminants. *Sci. Total Environ.* 2018, 642, 168–179. [CrossRef] [PubMed]
- 8. Chamseddinea, A.; Alameddinea, I.; Hatzopouloub, M.; El-Fadela, M. Seasonal variation of air quality in hospitals with indoor–outdoor correlations. *Built Environ.* **2019**, *148*, 689–700. [CrossRef]
- 9. Liu, J.; Wang, H.; Wen, W. Numerical Simulation on a Horizontal Airflow for Airborne Particles Control in Hospital Operating Room. *Built Environ.* **2009**, *44*, 2284–2289. [CrossRef]
- 10. Vijayan, V.K.; Paramesh, H.; Salvi, S.S.; Dalal, A.A. Enhancing indoor air quality: The air filter advantage. *Lung India* **2016**, *33*, 705. [CrossRef] [PubMed]
- 11. Bache, S.E.; Maclean, M.; Gettinby, G.; Anderson, J.G.; MacGregor, S.J.; Taggart, I. Universal decontamination of hospital surfaces in an occupied inpatient room with a continuous 405 nm light source. *J. Hosp. Infect.* **2018**, *98*, 67–73. [CrossRef] [PubMed]
- 12. Saini, J.; Dutta, M.; Marques, G. Indoor Air Quality Monitoring Systems Based on Internet of Things: A Systematic Review. *Int. J. Environ. Res. Public Health* 2020, *17*, 4942. [CrossRef] [PubMed]
- 13. Karuchit, S.; Warissarangkul, W. Optimum Indoor Air Uv Germicidal Irradiation System for Application in Community Hospitals. *Suranaree J. Sci. Technol.* **2016**, *23*, 251–260.
- 14. Zhou, Q.; Qian, H.; Liu, L. Numerical Investigation of Airborne Infection in Naturally Ventilated Hospital Wards With Central-Corridor Type. *Indoor Built Environ.* **2016**, 27, 59–69. [CrossRef]
- 15. Schiestel, R. *Modeling and Simulation of Turbulent Flows;* ISTE Ltd and John Wiley & Sons, Inc.: Hoboken, NJ, USA, 2008; pp. 293–327.
- 16. San, J. SW175HA2-72 Product Specification Approval Sheet; SAN JU Electric Machinery CO., LTD.: San Ju, China, 2011; pp. 1–4.
- 17. AIRREX. AR103 Fiberglass Mat Panel Filter; AIRREX CO., LTD.: Changzhou, China, 2018; pp. 1–2.
- Kim, J.; Jang, M.; Choi, K.; Kim, K. Perception of indoor air quality (IAQ) by workers in underground shopping centers in relation to sick-building syndrome (SBS) and store type: A cross-sectional study in Korea. *BMC Public Health* 2019, 19, 632–639. [CrossRef] [PubMed]