

Experimental Investigation of Ventilation Efficiency in a Dentistry Surgical Room

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Abstract. As a response to the need to provide an acceptable thermal comfort and air quality in indoor environments, various ventilation performance indicators were developed over the years. These metrics are mainly geared towards air distribution, heat and pollutant removals. Evidence exists of influencing factors on these indicators as centered on ventilation design and operations. Unlike other indoor environments, health care environment requires better performance of ventilation system to prevent an incidence of nosocomial and other hospital acquired illnesses. This study investigates, using in-situ experiments, the ventilation efficiency in a dentistry surgical room. Thermal and hygric parameters were monitored on the air terminal devices and occupied zone over a period of one week covering both occupied and unoccupied hours. The resulting time-series parameters were used to evaluate the room's ventilation effectiveness. Also, the obtained parameters were benchmarked against ASHRAE 170 (2013) and MS1525 (2014) requirements for ventilation in health care environment and building energy efficiency respectively. The results show that the mean daily operative conditions failed to satisfy the provisions of both standards. Regarding effectiveness, the findings reveal that the surgical room ventilation is ineffective with ventilation efficiency values ranging between 0 and 0.5 indicating air distribution short-circuiting. These results suggest further investigations, through numerical simulation, on the effect of this short-circuiting on thermal comfort, infection risk assessments and possible design improvements, an endeavour that forms our next line of research inquiries.

1 Introduction

In its optimal performance, the building ventilation system is tasked with the provision of an acceptable thermal comfort and perceived air quality through fresh air supply, heat removal and pollutant dilutions and/or removal. In health care buildings, ventilation systems are expected to preventing cross infection risks in addition to maintaining adequate thermal comfort to the patients, the caregivers and visitors. On the contrary, despite the high energy requirements [1, 2], the HVAC contributes to making building unhealthy [3] as several cases of nosocomial or hospital acquired illness (HAI) had been previously reported [4-6]. These problems, amongst others, had hitherto led to the emergence of building performance diagnostics. Documented evidence exists on building ventilation performance diagnosis for thermal comfort and indoor air quality [1, 7: Ai, 2014 #617], hygrothermal performance and ventilation energy efficiency [8-10]. They are equally focused on design optimisation [11-13] and performance assessment of new building component design [14, 15].

There are various performance metrics of assessing indoor ventilation based on the ventilation system's tasks: fresh air supply, air mixing, contaminant removal, air exchange, heat removal and occupant's protection [16]. Ventilation performance indices can therefore be grouped based on the expected functions of the system such as fresh air supply (e.g. air change rate, air change efficiency, mean age of air, local air change index, etc.), air mixing and/or diffusion (e.g. air diffusion performance index), pollutant removal (e.g. contaminant removal effectiveness, local air quality index, etc.). Specific indices are also available for thermal comfort (e.g. predicted mean vote, predicted percent dissatisfied, effective draft temperature, and percent dissatisfied due to draft, etc.). More details on the different ventilation performance metrics are available in the literature. Specifically, the readers are referred to Mundt *et al.* [17], Brouns *et al.* [18], and Liddament *et al.* [19].

Practical applications of ventilations performance metrics had been previously reported. The suitability of both air change efficiency and contaminant removal effectiveness formed the focus of a study carried out by Novoselac *et al.* [20] for different indoor spaces,

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ventilation strategies, and contaminant sources. The study revealed that the air change efficiency is adapted to evaluation of ventilation strategies in an indoor environment with unknown contamination sources. On the other hand, the contaminant removal effectiveness performs better for removal of contaminants with known position and generation rates. Nevertheless, certain correlation was found between the two metrics. Contaminant removal metrics had been applied to the ability of air distribution systems to remove unwanted indoor parameters that affect either thermal comfort and/or perceived air quality.

For instance, in a study on the ventilation performance of room air distribution systems, Gan [21] defined two contaminant removal metrics: thermal and ventilation efficiency with the former expressed for heat removal and the later for contaminant removal. Similarly, Irwin *et al.* [22] evaluated, using a scaled experiments, the air distribution performance of different ventilation schemes with the use of contaminant removal performance for heat and gaseous pollutants. In the study heat removal metrics was referred to as thermal energy removal effectiveness [22]. An analogous concept of heat removal as an index of ventilation performance metrics of air distribution system was adopted by Mohammed [23] and Aziz *et al.* [24] where the effectiveness of different air distribution systems was investigated.

Also, as observed by Ali *et al.* [25], the indoor thermal and hygric parameters are treated as contaminants in hygrothermal performance assessments. Hence, both indoor temperature and relative humidity parameters can be used in evaluating the performance of air distribution systems. Both parameters, in addition to

being environmental factors in human thermal comfort [26], they are also found to be important for the preservation of archival materials [27-29].

In this paper, we experiment the ventilation efficiency of an air distribution system using the thermal and hygric removal effectiveness metrics. A case studied dental surgical room is selected part of a larger study on the performance of the air distribution systems in the entire building that housed the surgical room. The aim is to understand the impact of various design parameters on the ventilation performance during operation. The study carried out in-situ experiments in a full-scale room for the evaluation of the ventilation performance. Although of all the available methods for building performance assessments (analytical, field experiments, numerical simulations, etc.), experimental studies provide data for analytical and numerical experiments, instrumentation, setup errors and data collection process greatly influence the data from field experiments [30]. Hence, it is important to assess the uncertainties in measured data from field experiments as good uncertainty clarifications improves results interpretation and reliability.

2 Methodology

2.1 Experimental Setup and Measurement Procedures

The surgical room, 5.3 m × 3.8 m × 3.4 m high, is air-conditioned and ventilated by a central constant air volume (CAV) air handling unit (AHU). The AHU controls the airflow, thermal and hygric distribution

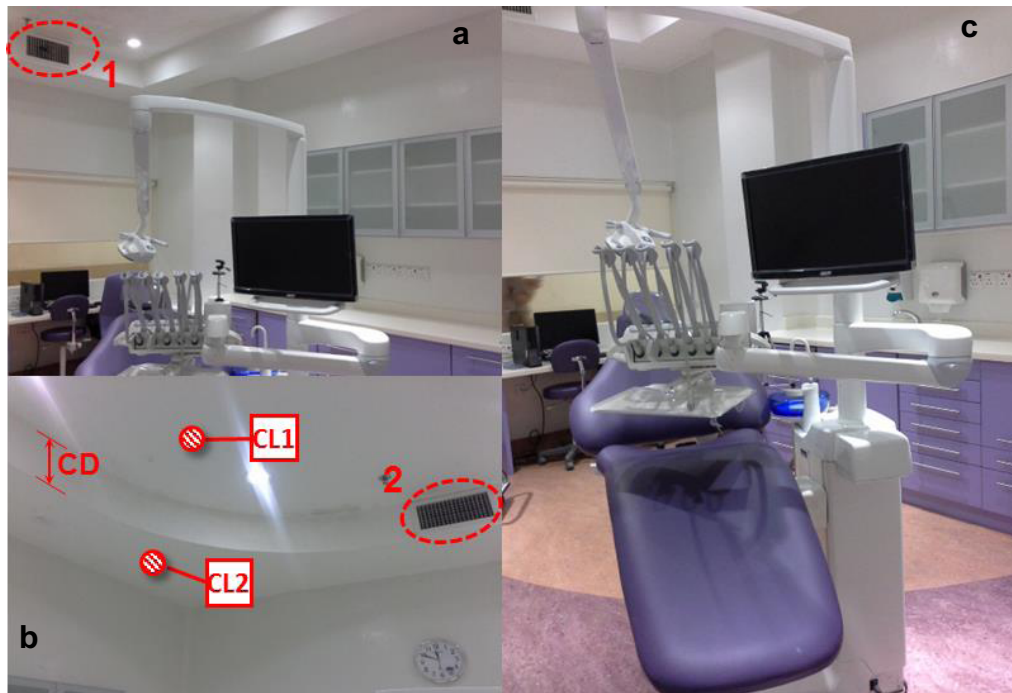


Figure 1. Pictorial representation of case-studied room showing: (a & b) The two-level ceiling design; (c) Room occupied zone showing the dental surgical (Dashed circles #1 & #2 are supply and exhaust outlets; CL1 & CL2 are ceiling levels; and CD is the ceiling drop).

within the room. The air terminal devices in the room are arranged to fit into the architectural design of the two-levelled ceiling (Figure 1).

The air distribution system in the room consist of rectangular supply and exhaust grilles (size 450 mm x 250 mm) vertically mounted on the drop of the architectural ceiling. The facility housing the surgical room operates between 9:00 am and 5:00 pm daily. Figure 2 shows details of the space layouts with positions of supply, exhaust and the measurement locations in the case study room. The air supply to the room is through duct network while the return is through the ceiling plenum as shown the figure.

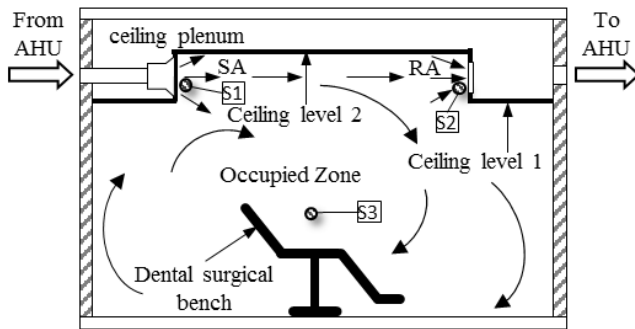


Figure 2. Vertical cross section through the case-studied dental surgical room with measurement locations (S1-S3 are temperature and humidity at supply inlet, return outlet and occupied zone respectively).

Data collection involves thermal and hygienic measurement with data loggers mounted at the supply diffuser, exhaust grille, and the occupied zone over a period of one week from 21/5/2015 to 27/5/2015. The sensors were EL-USB-2-LCD+ Data Logger with LCD from LASCAR Electronics. The uncertainty of the measured parameters is in accordance with the manufacturer's specifications. The measurement accuracy and precision are $\pm 0.5^\circ\text{C}/0.5^\circ\text{C}$, and $\pm 2.5\%/0.5\%$, for temperature and relative humidity respectively. As results of field measurements are susceptible to experimental uncertainties, the measured data are further evaluated for uncertainty assessment using the methodology described in a previous study by one of the authors [31]. The method is briefly described in the following section.

When making field observations of a variable, an acceptable level of uncertainty in the results is first determined as the level of confidence. As according to Vincent *et al.* [32], the commonly applied level of confidence in statistical investigations is 95% ($p < 0.05$). Therefore, we adopted a 95% level of confidence. To evaluate the measurement uncertainty, we applied Equation (1) for the confidence interval.

$$CI = \pm 1.96 \times SE_M \quad (1)$$

Where CI is the confidence interval and SEM is the standard error of the mean. The SEM is obtained from Equation (2) as follows:

$$SE_M = \frac{\sigma}{\sqrt{n}} \quad (2)$$

Where σ is the standard deviation and n is the number of observed time-series data. The *CI* in equation (1) is an interval, hence the positive “+” and negative “-” signs in the equation. The calculated *CI* value is compared with the equipment accuracy provided by the manufacturer. As described in Oladokun [31], the values of *CI* below or at most equal to the equipment accuracy indicates certainty in the obtained parameters at 95% level of confidence. The author further submitted that using this approach, where variation occurs in measurement at various locations, such variations can be attributed to causes other than the measuring equipment. On the other hand, when *CI* exceeds the equipment accuracy, equipment error is suggested and may call for calibration of such measuring tool.

2.2 Ventilation Performance Assessment

The study adopts thermal and moisture removal effectiveness as the ventilation performance indices. The measured temperature and humidity data are used to evaluate the thermal and moisture performance metrics. In this paper, we assessed these metrics from the model shown in Equations (3) and (4). The heat removal effectiveness is estimated as:

$$\varepsilon^{C_T} = \frac{\theta_e - \theta_s}{\theta_{oz} - \theta_s} \quad (3)$$

where ε^{C_T} is the heat removal effectiveness, θ_e is the exhaust temperature, θ_s is the supply temperature, and θ_{oz} is the temperature in the occupied zone. Similarly, the moisture removal effectiveness is calculated as:

$$\varepsilon^{C_H} = \frac{\varphi_e - \varphi_s}{\varphi_{oz} - \varphi_s} \quad (4)$$

where ε^{C_H} is the moisture removal effectiveness, φ_e is the exhaust humidity, φ_s is the supply humidity, and φ_{oz} is the humidity in the occupied zone.

The mean temperature and humidity data obtained from the in-situ measurements were used to calculate the heat and moisture removal effectiveness. To further determine the efficiency of the air distribution system, we apply the equations similar to the one described in Brouns *et al.* [18] as shown in Equations (5) and (6).

$$\eta^{C_T} = \frac{\varepsilon^{C_T}}{1 + \varepsilon^{C_T}} \quad (5)$$

and

$$\eta^{C_H} = \frac{\varepsilon^{C_H}}{1 + \varepsilon^{C_H}} \quad (6)$$

where η^{C_T} and η^{C_H} are the heat and moisture removal efficiency respectively.

According to Brouns *et al.* [18], the values of ventilation efficiency for different types of ventilations are as shown in Equation (7).

$$VE = \begin{cases} \eta^{C_{T,H}} = 0.5 : \text{Complete Mixing} \\ 0.5 \leq \eta^{C_{T,H}} \leq 1, \text{ Piston Flow} \\ 0 \leq \eta^{C_{T,H}} \leq 0.5, \text{ Short - circuiting} \end{cases} \quad (7)$$

where VE is the ventilation efficiency defined in Equations (5) and (6).

It is important to point out that testing the ventilation performance in a hospital environment with gaseous contaminant, using tracer gas, would provide better insight. Such method appears difficult in the current experiment as the use of simulated gaseous contaminants was not permitted. Nonetheless, such deterrent is typical of field experiments as limitations in controlling influencing parameters and measurement restrictions to few sampling points are established factors hindering on-site experiments [33].

3 Results and Discussions

3.1 Measurement Uncertainty Assessments

Measured data from sensors placed at the supply inlet, return outlet and occupied zone is examined for uncertainty using Equations (1) and (2). Table 1 compares the results of uncertainty assessment with the manufacturer’s specifications.

Table 1. Comparison of measuring equipment uncertainty between measured and manufacturer specified values.

Sensors	Measured Uncertainty		Manufacturer Specification	
	θ	ϕ	θ	ϕ
S1	$\pm 0.2 \text{ }^\circ\text{C}$	$\pm 0.3\%$	$\pm 0.5 \text{ }^\circ\text{C}$	$\pm 2.5\%$
S2	$\pm 0.1 \text{ }^\circ\text{C}$	$\pm 0.4\%$	$\pm 0.5 \text{ }^\circ\text{C}$	$\pm 2.5\%$
S3	$\pm 0.1 \text{ }^\circ\text{C}$	$\pm 0.4\%$	$\pm 0.5 \text{ }^\circ\text{C}$	$\pm 2.5\%$

$p < 0.05$

As shown in the table, the measurement uncertainty for temperature ranges from ± 0.1 to ± 0.2 while that of humidity varies between $\pm 0.3\%$ and $\pm 0.4\%$. It is worthy to note an observable difference in the sensor placed at the supply inlet with varying uncertainties from the ones at the exhaust outlet and occupied zone respectively. This variation can be attributed to the colder temperature at the

supply and higher velocity than the other two places. Overall, the estimated equipment uncertainty falls below the manufacturer’s specified accuracy for the sensors. It is, therefore, sufficient to conclude with 95% confidence that the measurements are accurate and variations in between measurement points will be due to reasons other than those related to the measuring equipment uncertainties. Nevertheless, there is a 5% chance that the results are inaccurate, the value that is statistically acceptable.

3.2 Thermal and Hygic Operative Conditions

The results of thermal and hygic operative conditions in the room showed the mean profiles as: S1 ($\theta = 15.3 \pm 0.2^\circ\text{C}$, $\phi = 86.5 \pm 0.3\%$), S2 ($\theta = 16.3 \pm 0.1^\circ\text{C}$, $\phi = 83.0 \pm 0.4\%$), and S3 ($\theta = 17.1 \pm 0.1^\circ\text{C}$, $\phi = 75.8 \pm 0.4\%$).

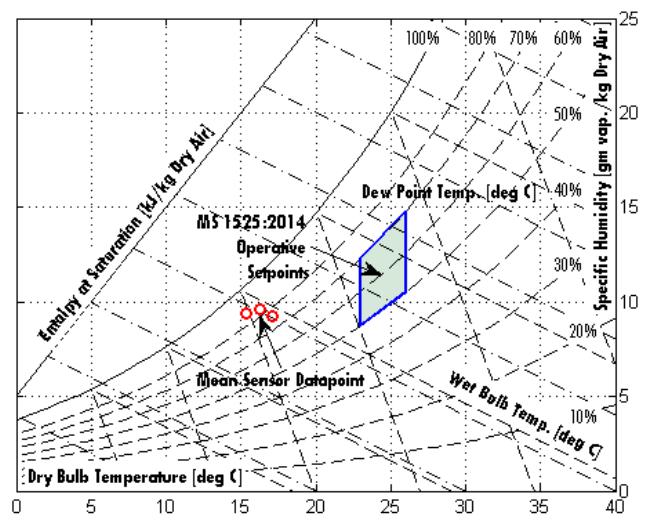


Figure 3. Psychrometric chart showing the mean operative conditions in the surgical room over the experimental duration.

The average temperature and humidity for the measurement locations S1-S3 are plotted on psychrometric charts together with MS 1525 specifications (Figure 3). The plot in Figure 3 revealed an over-cooled operative condition as all points fall below the specified values MS 1525 [34] where thresholds of temperature (θ) and humidity (ϕ) are respectively specified as $\theta = 23 - 26^\circ\text{C}$ and $\phi = 50 - 70\%$. A similar but more stringent set-point is often required by ASHRAE [35] with $\theta = 20 - 24^\circ\text{C}$ and $\phi = 20 - 60\%$. Evidently, the operative conditions, therefore, followed similar submissions of Sekhar [36] that buildings in the hot and humid climates are overcooled.

3.3 Ventilation Efficiency

The mean operative temperature and humidity were used to evaluate Equations (3) to (6). The results of dimensionless ventilation efficiency for heat and moisture removal are presented in Table 2.

Table 2: Estimated ventilation efficiency.

Parameters	Experimental values
heat removal efficiency, η^{C_r}	0.4
moisture removal efficiency, η^{C_H}	0.2

Applying the conditions in Equation (7), we found that the air distribution systems in the surgical room provide short-circuiting flow. Such air distribution system would be ineffective in removing heat and moisture from the occupied zone as the majority of the ventilation flow is directly exhausted without passing through the occupied zone. Most of the air in the occupied zone will be recirculated flow and not fresh air as shown with curved arrows in Figure 1. The observed situations would lead to, in addition to comfort and air quality issues, energy inefficiency of the ventilation system as most of the energy utilized by the ventilation system is taken up to move air without getting to where it is needed most, the occupied zone.

3.4 Discussions

In an indoor environment, the airflow characteristics are greatly influenced by the air jet which is maintained by the size, location of the supply outlet together with the supply temperature and flow velocity. This underscores the importance of the inlet air jets as the main driving

force of indoor air flow. The sizes and locations of the return outlets also play an important role in the indoor air flow trajectory. As the air moves from the supply to exhaust, the air is set into continuous motion thereby separating the room air into primary and secondary air zones. The location of exhaust openings, in relation to the supply, determine which of the primary (direct from supply outlet) and secondary (recirculated or mixed) air dominates the room air flow. For clarity purpose, the air flow pattern in dominant total-volume room ventilation schemes is described and compared with that of the case-studied room.

In mixing ventilation (Figure 4a), the supply and exhaust openings are located on the ceiling level, the room air is well mixed with neither of the primary or secondary air dominating the air flow. As shown in Figure 4a, the primary air dominates the occupied zone. Although there is the likelihood of air short-circuit (direct exhaust of supply air), such is limited as the general flow is dominated by primary air. Equally, there could be regions of flow recirculation in the room under mixing air flow. Nonetheless, mixing air distribution are found to produce acceptable ventilation efficiency [17, 19]. Similarly, in displacement type ventilation (Figure 4b), air is supplied with supply inlet close to the floor and exhaust on the ceiling or opposite wall thereby dividing the room air into colder (lower) and warmer (upper) stratified layers. As shown in Figure 4b, the air within the occupied zone is dominated with fresh primary air with **recirculated air restricted to the upper unoccupied**

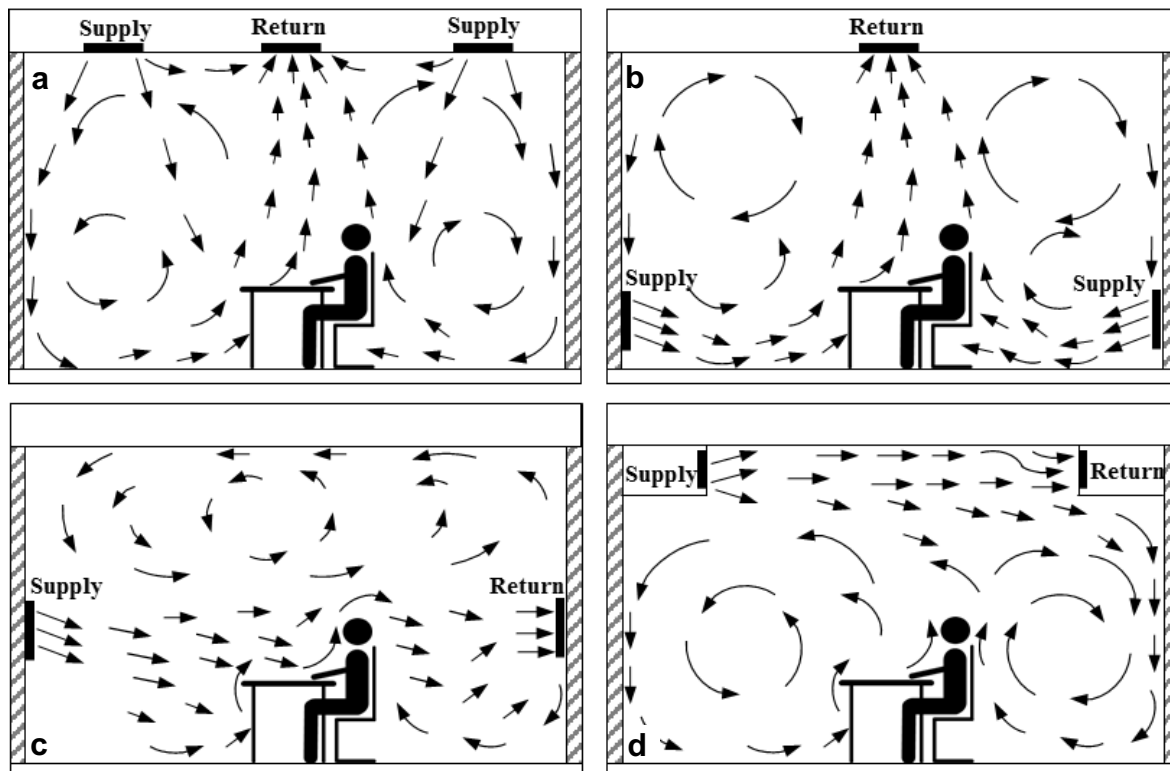


Figure 4. Room air distribution with varying supply and return locations (a) Mixing ventilation with ceiling supply and ceiling return; (b) Displacement ventilation with low-level supply and ceiling return; (c) Stratum ventilation with mid-level wall supply and mid-level wall return; and (d) Air distribution in the case-studied room with high-level ceiling supply and return.

zones. Displacement ventilation is better in performance than the traditional mixing ventilation with higher efficiency [17, 37]. The need to further improve the performance of mixing and displacement ventilation had resulted into a more recently invented stratum ventilation [38] shown in Figure 4c. In this mode of room air ventilation, the air is supplied and returned in the middle of walls thereby results in the provision of primary fresh air not just within the occupied zone but at the occupant's head level. Stratum ventilation provides higher efficiency than mixing and displacement ventilation [39].

In a non-isothermal wall jet close to the ceiling, the air streamline attaches to the ceiling surface by Coanda effects until the air reaches a point where it detaches from the ceiling surfaces. In air distribution where the return opening is located along the path dominated by Coanda effect, the major part of the primary air is exhausted from the room thereby resulting in flow short-circuiting. In such cases, the room airflow is dominated by the secondary recirculated air. Figure 5d shows a case similar to that found in the investigated room where the supply and return outlets are located near the ceiling. As shown in Figure 4d, the major primary air is restricted to the ceiling level with the room airflow dominated by secondary recirculated air. In this case, the room airflow is short-circuited and as such, the occupied zone is deprived of direct fresh air from the supply outlet. This further explains why the ventilation efficiency was found to be below 0.5 in the case-studied room. This finding reiterates the importance of combinations of inlet and outlet locations as influencing factors in the short-circuiting of supply air. Therefore, such arrangement where supply and return openings are fitted into the ceiling drops to suit architectural ceiling design should be avoided for better ventilation efficiency. Where such arrangement must be adopted, adequate care must be given to optimise the efficiency with respect to locations of the air terminal devices.

As the room airflow diffusion is affected by the supply and return outlets on one hand and the indoor heat and contaminants removal related to the room airflow, it is imperative to give utmost consideration to the supply and return air outlets. Reported evidence exist on the effect of various air distribution on indoor air pollutant and human exposure (health risk). An earlier study [40] on indoor contaminant dispersion with mixing and displacement ventilation systems revealed that room airflow pattern has a strong influence on the room contaminants distribution with the return air outlet location having a significant effect on human exposure level in the room. In a study on hospital ventilation performance with respect to airborne contagion, Li *et al.* [5] examined the role of air distribution in SARS transmission during 2003 outbreak. The results showed an imbalance in the airflow pattern from the supply diffusers and returned outlets, which resulted in high infection risks with recorded high fatality in the hospital. The findings from the present study is, therefore, an indication of potential danger in the use of the ventilation under the current design for health care environment as

poor efficiency increases the resident time of pollutant in the room thereby increasing the infection risks between the patient and the caregivers.

In this study, the ventilation effectiveness of air distribution system in a dental surgical room has been examined using in-situ experiments. The ventilation design and operative conditions are both found ineffective regarding compliance with energy efficiency benchmark and ventilation efficiency. While the energy inefficiency might be of little immediate concern, the ventilation inefficiency is a serious one especially in the health care environment where the risk of nosocomial is on the rise. We further discuss some of the study limitations for future research in the following sections. On the measurements, few data points were considered as the room is in operation during the measurements. Installing many sensors in the occupied zone would disrupt the operations. Similarly, as mentioned earlier, ventilation performance assessment with gaseous pollutants could provide additional information on the air distribution efficiency. Again, such is prohibitive under the operative conditions of the room. In our next stage of the study, numerical simulations would be employed to consider most of the constraints in the field experiments and to optimise the system for improved performance.

4 Conclusions

A study on the ventilation efficiency in the dental surgical room has been presented in this paper. The time-series measurement of thermal and hygric operative parameters were used to evaluate the ventilation efficiency and the compliance of operative conditions with the standard benchmark. The experimental data were verified for instrumentation uncertainty using an earlier developed procedure. A good overall accuracy was obtained between the measured parameters and the manufacturer's specifications. The operative conditions of the room showed over-cooling conditions against MS 1525:2014 on one hand and failure to comply with ASHRAE 170-2013 on provisions for health care facility on the other. The ventilation efficiency revealed that the air distribution system, under the current design, produces short-circuiting flow with values less than 0.5 for both thermal and moisture removal. The efficiency assessment demonstrated the need for caution in placing air terminal devices to follow the interior design of ceiling especially in the health care environment which require adequate protection for patients, caregivers, and visitors against nosocomial infections. Also, findings on the over-cooling conditions would suggest the need for more studies to decipher the reasons behind the recurring conditions of overcooling in the hot and humid climate. The ventilation efficiency results suggest further investigations, through numerical simulation, on the effect of this short-circuiting on thermal comfort, infection risk assessments, and possible design improvements. These would be considered in our future research endeavours.

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