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Highlights

- Local cooling load (LCL) is proposed for general non-uniform indoor environment
- LCL values obtained by Procedure CAV and Procedure VAV are almost the same
- LCL can be less than, equal to, or greater than the conventional space cooling load
- Ventilation system, heat source distribution and target zone have great impact on LCL

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Cooling load for the design of air terminals in a general non-uniform indoor environment oriented to local requirements

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Abstract

The indoor environment is usually non-uniform and only local air zones need to be controlled. Although the specific calculation method of cooling load for occupied zones has been proposed, no general definition and calculation method of cooling load exists for non-uniform indoor environment oriented to the local air zone. Thus, a general cooling load oriented to the local air zone, called local cooling load (LCL), was proposed for the design of air terminals in a general non-uniform indoor environment. Two calculation procedures of LCL, namely Procedure CAV (constant air volume) and Procedure VAV (variable air volume), were introduced. Experiments were conducted to illustrate LCL in a climate chamber measuring $4.12\text{ m} \times 2.89\text{ m} \times 4.2\text{ m}$. Twenty cases were numerically investigated to analyze the influences of different ventilation systems, heat source locations, and target zones on the LCL. The results show that the two calculation procedures lead to almost the same values of LCL. The LCL can be less than, equal to, or even greater than the conventional cooling load in the uniform indoor environment, depending on the three influencing factors. The concept of LCL can be used for the air volume and temperature difference design of terminals in non-uniform indoor environments oriented to local requirements.

Keywords

Local cooling load; Non-uniform indoor environment; Ventilation; Local air conditioning; Design of air terminal

Nomenclature

h_{set}	specific enthalpy of air in the entire room with set temperature and humidity, kJ/kg
h_{local_set}	specific enthalpy of air in the local air zone with set temperature and humidity, kJ/kg
h_s	specific enthalpy of supply air, kJ/kg
h_r	specific enthalpy of return air, kJ/kg
h_f	specific enthalpy of fresh air, kJ/kg
m_s	mass flow rate of supply air, kg/s
m_r	mass flow rate of return air, kg/s
m_f	mass flow rate of fresh air, kg/s
Q_{sys}	system cooling load, kW
Q_{fresh}	fresh air cooling load, kW
Q_{space}	conventional space cooling load in uniform indoor environment, kW
Q_{local}	local cooling load (LCL) in non-uniform indoor environment, kW
$Q_{local-s}$	sensible LCL in non-uniform indoor environment, kW
T_{local_set}	set temperature of air in the local air zone, °C
T_s	temperature of supply air, °C
T_e	temperature of exhaust air, °C
V_s	volume flow rate of supply air, m ³ /s

Abbreviations

<i>CAV</i>	constant air volume
<i>CFD</i>	computational fluid dynamics
<i>DV</i>	displacement ventilation
<i>HVAC</i>	heating, ventilation, and air conditioning
<i>LCL</i>	local cooling load
<i>MV</i>	mixing ventilation
<i>PV</i>	personalized ventilation
<i>SAC</i>	stratified air conditioning system
<i>UFAD</i>	underfloor air distribution system
<i>VAV</i>	variable air volume

1 Introduction

Energy production and consumption are a major global concern ^[1]. Especially in China, building energy consumption has been increasing rapidly in recent years owing to the rapid growth of floor space and energy use intensity for all building types ^[2]. Building energy consumption accounts for 30% of the global energy consumption ^[3], in which heating, ventilation, and air conditioning (HVAC) systems play a major role^[4-5]. As a significant part of HVAC system, ventilation system needs to be designed properly to save energy.

An effective use of different ventilation systems relies on the proper consideration, quantification, and treatment of the space cooling load in the indoor environment. Conventionally, the indoor environment created by mixing ventilation (MV) is treated as uniform. The space cooling load in a uniform indoor environment is the rate of energy removal required to maintain the entire room at a desired temperature and humidity condition ^[6]. However, with the development of displacement ventilation (DV) ^[7-9], underfloor air distribution system (UFAD) ^[10-11], stratified air conditioning system (SAC) ^[12-13], etc., increasing attention is being paid to the occupied zone instead of the entire room. Because only the heat entering the occupied zone needs to be considered in these ventilation systems, the space cooling load is reduced. In addition to the conventional space cooling load calculation methods, some new methods have been proposed, for example, by Yuan et al. ^[14-15] and Lau et al. ^[16] for DV systems, Loudermilk ^[17], Bauman et al. ^[18], and Xu et al. ^[19] for UFAD systems, and Miyagawa et al. ^[20] and Zou et al. ^[21] for SAC systems. These

calculation methods can be summarized as a two-zone model method, as indicated in Eq. (1)^[19]. It divides the room into two zones, i.e., the occupied zone and unoccupied zone, and correspondingly, the cooling load can be artificially distributed into $Q_{occupied}$ and $Q_{unoccupied}$. The point then is to calculate which part of the heat gain is effective in the occupied zone to form the space cooling load. It can be estimated empirically or regressed from a database with adequate computational fluid dynamics (CFD) cases, and it varies with different researchers.

$$Q_{occupied} = ECLF_{wall} Q_{wall} + ECLF_{occupant} Q_{occupant} + ECLF_{lighting} Q_{lighting} + ECLF_{equipment} Q_{equipment} \quad (1)$$

where Q_{wall} , $Q_{occupant}$, $Q_{lighting}$, and $Q_{equipment}$ are the heat gain from walls, occupants, lighting, and equipment, respectively, in W; $ECLF_{wall}$, $ECLF_{occupant}$, $ECLF_{lighting}$, and $ECLF_{equipment}$ are the effective space cooling load factor (ECLF) for walls, occupants, lighting, and equipment, respectively.

In general, there is a need for a paradigm shift from uniform indoor environment to non-uniform indoor environment accommodating individual preferences^[22]. The target zones that need to be controlled are mostly only one or a few local air zones instead of the occupied zone or the entire room. In industrial buildings, only the zone around the products or workers needs to be protected from pollution and thermal stress^[23-24]. Based on this, local exhaust ventilation^[25-26] and push-pull ventilation^[27] have been widely used to directly remove the contaminant and heat. In operating rooms and hospital wards, only the local indoor environment for patients and medical personnel needs to be secured^[28]. Local bed ventilation^[29-30] and natural personalized

ventilation ^[31] were proposed to meet the individual indoor air quality requirements and avoid cross-infection. In office buildings, only the area consisting of the breathing zone and personnel working zone, where occupants spend most of their time, needs to be controlled ^[32-33]. Personalized ventilation (PV) ^[34], ductless PV ^[35], ceiling-mounted PV ^[36], and protected occupied zone ventilation ^[37] were proposed to address the individual requirements of indoor air quality and thermal comfort. In residential buildings, bedside personalized ventilation ^[38] was applied for sleeping people to create a local environment that achieves human sleep comfort and quality. Although the distinction between occupied zone and unoccupied zone as mentioned in the previous paragraph can be regarded as a type of non-uniform indoor environment, the general non-uniform indoor environment means that the target zone can be any part and any shape of the ventilated space. If only the local air zone is controlled, the real requirement in the non-uniform indoor environment can be satisfied and the space cooling load will be different.

Currently, researchers mostly pay attention to the local air quality and local thermal comfort, but express little concern on the local cooling load in a general non-uniform indoor environment where only the local air zone is controlled. Few researchers ^[39-40] utilized a modified heat removal effectiveness to evaluate the energy performance of ventilation system in a non-uniform indoor environment oriented to the local requirements. Some issues about the cooling load in a general non-uniform indoor environment are still not well addressed: How much cooling capacity should be input to the room in a general non-uniform indoor environment? Is the space

cooling load in a general non-uniform environment always smaller than that in a uniform indoor environment? Which factors will influence the space cooling load in a general non-uniform environment? Due to these existing issues, the air terminals and ventilation system will not be designed properly, the actual indoor requirement will not be met, and the energy will be wasted instead in the general non-uniform indoor environment.

In order to address the above mentioned issues, a general cooling load oriented to the local zone, which is called local cooling load (LCL), is proposed in this study. The calculation method based on the experiment or CFD simulation is introduced. Experiments are conducted to obtain and preliminarily study the LCL in a climate chamber measuring $4.12\text{ m} \times 2.89\text{ m} \times 4.2\text{ m}$ at Technical University of Denmark. Finally, the influence of the ventilation system, the heat source location, and the location and size of the target zone on the LCL is comprehensively analyzed by CFD simulation.

2 Concept and calculation method of LCL

2.1 Definition of LCL

Cooling load results from the unsteady heat transfer processes of conduction, convection, and radiation through the building envelope, internal sources and system components. In the cooling load calculation, the time delay effect is a critical point and has been satisfactorily solved with the heat balance method and radiant time series method as explained in ASHARE standard ^[6]. Actually, the unsteady heat

transfer process can be simplified to multiple quasi-steady heat transfer processes, each of which provides the specified heat gains that consider the time delay effect. Therefore, the cooling load in the steady state is mainly analyzed in this study.

In general, the centralized air conditioning system as shown in Fig. 1, is used to create the conventional uniform indoor environment.

Figure 1. System cooling load and space cooling load in conventional uniform indoor environment created by a centralized air conditioning system

The system cooling load is used to determine the size of the air conditioning system that influences the energy consumption and operation cost. It focuses on the air handling unit as shown in Fig. 1(a), and it can be demonstrated with the line segment between point M and point S in Fig. 1(b) and expressed by Eq. (2.1).

$$Q_{sys} = m_r h_r + m_f h_f - m_s h_s \quad (2.1)$$

$$Q_{sys} = m_s (h_{set} - h_s) + m_f (h_f - h_{set}) = Q_{space} + Q_{fresh} \quad (2.2)$$

where m_r , m_f and m_s are the mass flow rates of return air, fresh air and supply air, respectively, kg/s; h_r , h_f , h_s and h_{set} are the specific enthalpy of return air, fresh air, supply air and the entire room air with set temperature and humidity, respectively, kJ/kg; Q_{fresh} , Q_{space} and Q_{sys} are the fresh air cooling load, the space cooling load

and the system cooling load, respectively, kW.

In the conventional uniform indoor environment, Q_{sys} is equal to the sum of Q_{space} and Q_{fresh} when reheating, heat leakage, and fan heat are ignored, as indicated in Eq. (2.2). Different from Q_{sys} , Q_{space} is used to decide the supply air parameters to design the air terminals. It focuses on the ventilated space of the room, as shown in Fig.1 (a). In this study, the space cooling load is mainly investigated.

In the conventional uniform indoor environment, the space cooling load Q_{space} is demonstrated as the line segment between point *Set* and point *S* in Fig. 1(b), and could be routinely calculated by using the steady-state heat balance equation as shown in Eq. (3).

$$Q_{space} = m_s (h_{set} - h_s) = Q_{gain} \quad (3)$$

where Q_{gain} is the total heat gain in the room, kW. In steady state, Q_{space} is equal to Q_{gain} in the uniform indoor environment.

The validity of Eq. (3) relies on the assumption that the room air is fully mixed. However, in the general non-uniform indoor environment where only the local air zone is controlled, the room air is not fully mixed so that the Eq. (3) does not work. Three diagrams are used to further clarify the difference of space cooling load between the conventional uniform indoor environment and the general non-uniform indoor environment, as shown in Fig. 2.

(a) Uniform indoor environment

Figure 2. Space cooling load in uniform and non-uniform indoor environment

In Figs. 2 (a-c), the size of the room and the amount of heat gain are all the same (but the heat source location is different). In Fig. 2(a), a uniform environment with a temperature of $26 \text{ }^{\circ}\text{C}$ is maintained by MV, and the corresponding supply air parameters are $16 \text{ }^{\circ}\text{C}$ and $180 \text{ m}^3/\text{h}$. In Fig. 2(b-c), a non-uniform indoor environment oriented to the local air zone is maintained by DV, and the supply air parameters are higher $20 \text{ }^{\circ}\text{C}$ and $180 \text{ m}^3/\text{h}$ (or $16 \text{ }^{\circ}\text{C}$ and lower $120 \text{ m}^3/\text{h}$) because the local zone is close to the supply air diffuser. The supply air parameters in the non-uniform indoor environment are significantly different from those in the uniform indoor environment. Therefore, the space cooling load in the non-uniform environment when only the local air zone is controlled will be different and needs to be analyzed.

When applying Eq. (3) in the non-uniform indoor environment, i.e., replacing the set value in the entire room h_{set} with the set value in the local air zone h_{local_set} , the LCL in the non-uniform environment can be calculated using Eq. (4).

$$Q_{local} = m_s (h_{local_set} - h_s) \neq Q_{gain} \quad (4)$$

where h_{local_set} (kJ/kg) is the specific enthalpy of air in the local air zone with the set temperature and humidity, and Q_{local} (kW) is the LCL when only the local air zone needs to be controlled instead of the entire room or occupied zone.

If only the transfer of sensible heat occurs or the temperature is the main

concern, Eq. (4) can be expressed as Eq. (5):

$$Q_{local-s} = c_p m_s (T_{local_set} - T_s) \neq Q_{gain-s} \quad (5)$$

where c_p is the specific heat capacity of air, kJ/(kg·°C); T_s is the supply air temperature, °C; T_{local_set} is the set air temperature in the local air zone, °C; $Q_{local-s}$ is the sensible LCL when only the average temperature in the local air zone needs to be controlled, kW.

As shown in Eq. (5), LCL is defined based on the supply air temperature and supply airflow rate of the entire space, and the average temperature of the target zone. It indicates the cooling amount we provide through air terminals to the space to maintain the temperature of the target zone. Actually, LCL reflects the influence of all the heat sources on the target zone under the given flow pattern. According to the theoretical derivation in the PhD dissertation^[41] and the similar conclusion about contaminant in the paper of Roomvent & Ventilation 2018 conference^[42], the LCL is decided not by the supply air temperature and the average temperature of target zone, but by the intensities and positions of all heat sources and the flow pattern. Therefore, the LCL proposed in this study can be used to get further understanding of the cooling load in a general non-uniform indoor environment, which may instruct us to optimize the design of air terminals.

By comparing Eq. (3) with Eq. (4), it can be observed that h_{set} is equal to h_{local_set} so that the LCL is equal to the conventional space cooling load in the uniform indoor environment. It indicates that LCL is generic and can be applied in the both uniform and non-uniform indoor environment. In other words, the LCL in the

non-uniform indoor environment is an extension of the conventional space cooling load; the conventional space cooling load is a special case of LCL.

In addition, the set temperature in the target zone T_{local_set} is usually not equal to the return air temperature T_r in the non-uniform environment (for example, $26\text{ }^{\circ}\text{C} \neq 30\text{ }^{\circ}\text{C}$ in Fig. 2(b)). Therefore, the LCL in the non-uniform environment is not equal to the heat gain in the steady state, as indicated in Eq. (4-5). It causes that the calculation method of LCL in the non-uniform indoor environment cannot be similar to the conventional space cooling load in the uniform indoor environment.

2.2 Calculation method of LCL

Since temperature is the main parameter in the indoor thermal environment, a calculation method for sensible LCL which doesn't consider the humidity is introduced. In the general non-uniform indoor environment, only the temperature in the local air zone is required to be the set value ($26\text{ }^{\circ}\text{C}$ here), while the temperature in other zones is unknown. Thus, the influence of the other zones on the target zone is not clear so that LCL cannot be obtained directly. Obtaining the spatial temperature distribution by experimental measurement or CFD simulation^[43] is the first step in making an exact assessment of LCL. Similar to the description in Fig. 2(b) and Fig. 2(c), there are two procedures for calculating LCL, namely Procedure CAV (constant air volume) and Procedure VAV (variable air volume). The detailed calculation procedures are shown in Fig. 3.

In Procedure CAV, the flow rate of supply air is fixed. First, we set a supposed temperature of supply air. Second, with the fixed flow rate and assumed temperature

of supply air, the indoor air temperature distribution is obtained by experimental measurement or CFD simulation, and the average temperature in the local air zone can be computed. Third, multiple rounds of iterations are performed until the average temperature in the local air zone is close to the set value (for example, 26 °C here) by modifying the temperature of supply air. Finally, with the final temperature and fixed flow rate of supply air, the LCL can be calculated using Eq. (4). The detailed flow chart is shown in Fig. 3(a).

In Procedure VAV, the supply air temperature is fixed. First, we set a supposed flow rate of supply air. Second, with the fixed temperature and assumed flow rate of supply air, the indoor air temperature distribution is obtained by experimental measurement or CFD simulation, and the average temperature in the local air zone can be computed. Third, multiple rounds of iterations are performed until the average temperature in the local air zone is close to the set value by modifying the flow rate of supply air. Finally, with the final flow rate and fixed temperature of supply air, the LCL can be calculated using Eq. (4). The detailed flow chart is shown in Fig. 3(b).

Figure 3. Calculation procedures of LCL

3 Experimental study

3.1 Experimental setup

In order to obtain and preliminarily study the LCL, the experiments were conducted in a full-scale climate chamber at Technical University of Denmark. The chamber was located in the laboratory hall; therefore, the surrounding temperature was fairly stable all the time. The chamber, with dimensions of $4.12\text{ m} \times 2.89\text{ m} \times 4.2\text{ m}$ (length $\times$ height $\times$ width), was ventilated via a linear slot diffuser in the middle of ceiling and two exhaust diffusers in the two corners of ceiling, as shown in Fig.4 (a-b). The supply air diffuser had two slots with a flow deflection vane to obtain a smooth flow and provide Coanda effect, and the size of each slot was $0.47\text{ m} \times 0.01\text{ m}$. The two exhaust diffusers with size of $0.1\text{ m} \times 0.1\text{ m}$ were connected to an exhaust fan via ducts. It should be noted that an all-fresh air cooling system was adopted in this experiment. The fresh air with constant temperature of $26\text{ }^{\circ}\text{C}$ was cooled by a heat exchanger, then was supplied to the room by slot diffusers, and all was exhausted finally. The heat sources in the chamber were simulated by three gray cylindrical boxes, as shown in Fig. 4(c). Each box was heated at 140 W , its diameter was 0.4 m , and the height was converted to 0.65 m . The chamber was illuminated by two fluorescent lights, each of which consumed 50 W . The space below 1.4 m in the chamber was divided evenly into nine local air zones and numbered accordingly, as shown in Fig. 4(d). Zone D was the target zone, in which the average temperature needs to be kept at $26\text{ }^{\circ}\text{C}$. The three heat sources were located in Zone E or Zone F.

Figure 4. Climate chamber

Each wall was divided into three parts in the vertical direction owing to different materials, the heat transfer coefficient of which is given in Table 1.

Table 1. Heat transfer coefficient of the envelope

	South-bottom wall	South-middle wall	South-top wall	North-bottom wall	North-middle wall	North-top wall	Ceiling
Heat transfer coefficient $W/(m^2 \cdot ^\circ C)$	0.6	0.8	0.6	0.6	1.8	0.6	0.5
	West-bottom wall	West-middle wall	West-top wall	East-bottom wall	East-middle wall	East-top wall	Floor
Heat transfer coefficient $W/(m^2 \cdot ^\circ C)$	0.8	1.5	1.5	0.6	1.8	0.6	0

Thirty-six Crafttemp Astra thermistor sensors with a precision of $\pm 0.2^\circ C$ were

used to measure the internal surface temperature of the four walls, floor, and ceiling. Seven RS Pro Class A PT100 sensors with a precision of ± 0.15 °C at the heights of 0.1 m, 0.3 m, 0.6 m, 1.1 m, 1.7 m, 2.1 m, and 2.6 m were used to measure the indoor air temperature. Eight low-velocity wireless thermal anemometers AirDistSys 5000 with a precision of ± 0.02 m/s were used for the air velocity measurement at heights of 0.1 m, 0.3 m, 0.6 m, 1.1 m, 1.4 m, 1.7 m, 2.1 m, and 2.6 m. During measurements, the air temperatures of four points, namely D1, D2, D3, and D4, at a height of 0.7 m were monitored in real-time by four PT100 sensors (precision is ± 0.15 °C) and Keysight 34970A, and they were used to represent the average temperature of the target zone. Additionally, the supply air temperature, exhaust air temperature, and ambient air temperature were all monitored simultaneously.

As shown in Fig. 3, there are two procedures to obtain the LCL, i.e., Procedure CAV and Procedure VAV. Based on this, four experimental cases were designed, as summarized in Table 2.

Table 2. Experimental cases

Case No.	Heat sources location	Calculation procedure	Supply air parameter	Target zone
Case E-CAV	Zone E	Procedure CAV	23.5 L/s	Zone D, average temperature should be controlled as 26 °C
Case E-VAV		Procedure VAV	16 °C	
Case F-CAV	Zone F	Procedure CAV	23.5 L/s	
Case F-VAV		Procedure VAV	16 °C	

The three heat sources with 420 W were located in Zone E for Case E-CAV and Case E-VAV, but in Zone F for Case F-CAV and Case F-VAV. In Case E-CAV and Case F-CAV, the supply airflow rate was constant at 23.5 L/s, which was monitored by an FCO510 micromanometer (precision is 0.25% of reading). In Case E-VAV and Case F-VAV, the supply air temperature was constant at 16 °C, which was monitored

by the PT100 sensor.

3.2 Test procedure

By adjusting the exhaust fan, we can attain the same supply and exhaust airflow rate and hence, a balanced condition in the chamber. For Case E-CAV and Case F-CAV, we keep the supply airflow rate at 23.5 L/s, and then regulate the temperature of the supply air for 3-5 times to attain a 26 °C average temperature of the four points in Zone D. For Case E-VAV and Case F-VAV, we keep the supply air temperature at 16 °C, and then regulate the damper of supply air duct for 3-5 times to attain a 26 °C average temperature of the four points in Zone D. When the air temperature of the chamber is basically unchanged and the air velocity is under a small fluctuation, a steady state is considered to be achieved and the measurement is then conducted. Owing to the limited number of sensors, the indoor air temperature and velocity in the chamber were measured line by line by shifting the location of a probe-holding steel stand attached with seven PT100 sensors and eight anemometer sensors. The steel stand is shown in Fig. 4(c), and it was located at the center of each zone as shown in Fig. 4(d). A waiting time of 30 min was provided for the chamber to reach steady-state condition again after relocation of the steel stand.

3.3 Experimental results

When the average temperature of the four points in Zone D was 26 °C, the final supply air parameters were obtained, as listed in Table 3 below.

Table 3. Final supply air parameters in four experimental cases

	Final supply airflow rate V_s (L/s)	Final supply air temperature T_s (°C)
Case E-CAV	23.5	15.0
Case E-VAV	26.3	16.0
Case F-CAV	23.5	17.0
Case F-VAV	21.0	16.0

In Case E-CAV and Case F-CAV, the final supply air temperature is 15.0 °C and 17.0 °C respectively. According to Eq. (5), the LCL in Case E-CAV can be obtained and the result is 311.8 W with an uncertainty of ± 5.6 W, and the LCL in Case F-CAV can be obtained and the result is 255.1 W with an uncertainty of ± 4.9 W. In Case E-VAV and Case F-VAV, the final supply airflow rate is 26.3 L/s and 21.0 L/s respectively. According to Eq. (5), the LCL in Case E-VAV can be obtained and the result is 317.2 W with an uncertainty of ± 5.6 W, and the LCL in Case F-VAV can be obtained and the result is 253.3 W with an uncertainty of ± 4.6 W. The detailed experimental results are shown in Fig. 5. By the way, the conventional space cooling load in Fig. 5 is 281 W with an uncertainty of ± 9.7 W, which is calculated by summing the heat gain of the two lights, three heat sources, and the envelope in the ideal uniform indoor environment with 26 °C everywhere.

(a) When heat sources are located in Zone E

(b) When heat sources are located in Zone F

Figure 5. LCL in the four experiment cases

By comparing Case E-CAV and Case E-VAV as shown in Fig. 5(a), the LCL

calculated with Procedure CAV and Procedure VAV are 311.8 W and 317.2 W, respectively. The difference is less than 3%, which indicates that they could be considered as identical with the allowable deviation. A similar conclusion could be drawn by comparing Case F-CAV and Case F-VAV as shown in Fig. 5(b).

Based on the existing knowledge as shown in Eq. (1), the local cooling load in a non-uniform indoor environment should be smaller than the conventional cooling load, because only the part of heat sources involved in the local air zone is considered. In Fig. 5(b), the experimental results in Case F-CAV and Case F-VAV show that the LCL is smaller than the conventional cooling load, which is consistent with the existing knowledge. However, in Fig. 5(a), the experimental results in Case E-CAV and Case E-VAV show that the LCL is greater than the conventional cooling load instead, which subverts the existing knowledge. It indicates that more cooling capacity should be supplied to a room in a non-uniform indoor environment when only the local air zone is controlled. This is because the three heat sources are very close to the target zone in Case E-CAV and Case E-VAV. The contribution of three heat sources to heating the target zone is very large so that the target zone is comparatively difficult to be cooled.

In comparison with the conventional uniform environment, the air with lower temperature (or higher flow rate) was supplied to the room in the non-uniform indoor environment when only Zone D is controlled in Case E-CAV (or Case E-VAV). Thus, the indoor air temperature and exhaust air temperature was relatively low, as shown in Fig. 6(a). On the contrary, the indoor air temperature and exhaust air temperature in

Case F-CAV (or Case F-VAV) was relatively high, as shown in Fig. 6(b).

(a) When heat sources are located in Zone E (b) When heat sources are located in Zone F

Figure 6. Exhaust air temperature in the four experiment cases

4 Numerical study

Due to the limited experimental conditions, it is unable to conduct a large number of experimental measurement in the climate chamber to comprehensively illustrate the LCL. Therefore, the numerical simulation based on CFD was adopted to further analyze the influencing factors of LCL including the ventilation system, heat source location, and target zone.

4.1 CFD model validation

In the steady-state CFD simulation with Airpak 3.0 software, Boussinesq approximation was used to model the buoyancy force. The RNG κ - ϵ turbulence model, which is widely used in mixing ventilation^[44-45], underfloor air distribution^[46-47] and displacement ventilation^[48-49], was employed to consider the turbulent flow.

The body-force-weighted scheme for pressure was adopted, which computes the face pressure by assuming that the normal gradient of the difference between pressure and body forces is constant. The second-order upwind scheme for momentum, energy and

others were adopted, which computes the quantities at cell faces by using a multidimensional linear reconstruction approach. The discrete equations were solved under the imposed boundary conditions with an iterative procedure and the SIMPLE algorithm. In addition, the discrete ordinates radiation model was used in the CFD model. Because of the nonlinearity of the equation set, under-relaxation factors which reduce the update of computed variables at each iteration was applied to ensure convergence. Convergence was assumed to be attained when all the scaled residuals leveled off and minima of 10^{-6} for energy and 10^{-3} for others were reached.

Based on the experimental results, a CFD model validation is conducted. The geometry and boundary conditions in the CFD model are the same as the measurement data in experimental Case E-CAV. Figs. 7–8 compare the measured and simulated air velocity and temperature profiles at different heights in eight zones of the chamber. The absolute difference between the simulated value and the experimental value is less than 0.04 m/s for air velocity and 0.3 °C for air temperature. The coefficient of determination R^2 is 0.8193 for air velocity and 0.9371 for air temperature, which indicates a good correlation. Consequently, it implies that the CFD model is valid and can be used to simulate the room airflow.

Figure 7. Comparison of simulated and measured velocity profiles in the eight zones (measuring uncertainty is ± 0.02 m/s)

Figure 8. Comparison of simulated and measured temperature profiles in the eight zones (measuring uncertainty is ± 0.2 °C)

4.2 Simulated cases

4.2.1 Geometry and grid

The geometry for the CFD model was identical to the climate chamber, as shown in Fig. 4 and Fig. 9. Three different types of ventilation systems were selected: MV, UFAD, and DV. For MV, as shown in Fig. 4(a), the details have been described in Section 3.1. For UFAD, as shown in Fig. 9(a), the inlet with size of $0.235 \text{ m} \times 0.1 \text{ m}$ was located on the middle of the floor. For DV, as shown in Fig. 9(b), the inlet with size of $0.47 \text{ m} \times 0.25 \text{ m}$ was located on the west wall at a height of 0.5 m . The exhaust diffusers in the three ventilation systems are the same.

(a) Underfloor air distribution system (UFAD) (b) Displacement ventilation system (DV)

Figure 9. Geometry of CFD model

Three target zones with different locations were selected, as shown in Fig. 4(d). Four target zones with different sizes were selected, all of which were located in the middle of the floor as shown in Fig. 10. They are the entire room measuring $4.12 \text{ m} \times 2.89 \text{ m} \times 4.2 \text{ m}$, occupied zone measuring $4.12 \text{ m} \times 1.4 \text{ m} \times 4.2 \text{ m}$, big Zone E measuring $2.5 \text{ m} \times 1.4 \text{ m} \times 2.5 \text{ m}$, and Zone E measuring $1.38 \text{ m} \times 1.4 \text{ m} \times 1.4 \text{ m}$.

(a) Isometric view of the four target zones

(b) Top view of the four target zones

Figure 10. The four target zones with different sizes

The grid of computational domain comprised $50 \times 50 \times 50$ nodes, respectively in the X, Y, and Z directions. The first node was positioned at 0.01 m from the wall, the non-dimensional wall distance (y^+) ranged from 37 to 105, and the average y^+ was approximately 60, which can ensure a good approximation near the wall [50-51]. A maximum stretching ratio of 1.2 controlled the cells located in the surroundings of three heat sources, two lights, and walls. 20 and 4 cells were used along the length and height for each supply air opening respectively, and 10 and 6 cells were used along the length and height for each exhaust air opening respectively. The grid had approximately 576,000 hexahedral cells (differentiating slightly with different ventilation systems), which resulted from a grid sensitivity analysis based on three grids: a coarse grid with 284,105 cells, a basic grid with 576,042 cells (reference case), and a fine grid with 1,122,575 cells.

4.2.2 Boundary conditions

Twenty cases were designed to illustrate the LCL and analyze its influencing factors in a non-uniform indoor environment when only the local air zone is controlled, as listed in Table 4.

Table 4. Simulated cases

Case No.	Calculation procedure	Ventilation system	Heat source location	Target zone	
Case A-MV-1-1	Procedure CAV $V_s = 23.5$ L/s	MV	Location 1	Zone D	
Case A-MV-1-2				Zone E	
Case A-MV-1-3				Zone F	
Case A-MV-1-4				Big Zone E	
Case A-MV-1-5				Occupied zone	
Case A-MV-1-6				Entire room	
Case A-MV-2-1			Location 2	Zone D	
Case A-MV-3-1			Location 3		
Case A-UFAD-1-1			UFAD		Location 1
Case A-DV-1-1			DV		
Case B-MV-1-1	Procedure VAV $T_s = 16$ °C	MV	Location 1	Zone D	
Case B-MV-1-2				Zone E	
Case B-MV-1-3				Zone F	
Case B-MV-1-4				Big Zone E	
Case B-MV-1-5				Occupied zone	
Case B-MV-1-6				Entire room	
Case B-MV-2-1			Location 2	Zone D	
Case B-MV-3-1			Location 3		
Case B-UFAD-1-1			UFAD		Location 1
Case B-DV-1-1	DV				

In Table 4, case number is named as “Case X (calculation procedure)-X (ventilation system)-X (heat source location)-X (target zone)”. The total heat gain from three heat sources are the same at 420 W, while their locations are different, i.e., Location 1, Location 2, and Location 3, which means that the three heat sources are located in Zone E, Zone C, and Zone B respectively. The “target zone” is the local air zone that should be maintained at 26 °C. In order to analyze its effect, three target zones with different locations (i.e. Zone D, Zone E, and Zone F) and four target zones

with different sizes (i.e. Zone E, Big Zone E, occupied zone and entire room) are investigated.

All the 20 cases are divided into two series, namely series A and series B. In series A, Procedure CAV is adopted and the supply airflow rate is constant at 23.5 L/s. In series B, Procedure VAV is adopted and the supply air temperature is constant at 16.0 °C. By comparing the two series, the difference between Procedure CAV and Procedure VAV could be analyzed further. Case A-MV-1-1 is the basic case, which is compared with the other cases. Comparison of Case A-MV-1-1, Case A-MV-1-2, and Case A-MV-1-3 is intended to analyze the effect of target zone's location, comparison of Case A-MV-1-2, Case A-MV-1-4, Case A-MV-1-5, and Case A-MV-1-6 is intended to analyze the effect of target zone's size, comparison of Case A-MV-1-1, Case A-MV-2-1, and Case A-MV-3-1 is to analyze the effect of heat source location, and comparison of Case A-MV-1-1, Case A-UFAD-1-1, and Case A-DV-1-1 is to analyze the effect of ventilation system.

The ambient air temperature surrounding the chamber was measured as steady at 22.5 °C. The measured ambient air temperature and heat transfer coefficient (as given in Table 1) were used as the boundary conditions of the envelope in the CFD model.

The inlets for MV, UFAD, and DV were set as 2.5 m/s ($\pm Z$ direction), 1.0 m/s (+Y direction), and 0.2 m/s (+X direction) respectively.

4.3 Simulated results

4.3.1 Final supply air parameters

Given the temperature of each mesh node in the CFD model, the average temperature in the target zone could be determined. After 3-5 rounds of iterations, the average temperature in the target zone in all 20 cases was adjusted to 26 °C, as shown in Figs. 11–13. Because of limited space, only the flow field and temperature distribution at $X = 2.06$ m in Case A-MV-1-1, Case A-UFAD-1-1, and Case A-DV-1-1 are displayed.

Figure 11. Airflow pattern and temperature contour in Case A-MV-1-1 with MV system

Figure 12. Airflow pattern and temperature contour in Case A-UFAD-1-1 with UFAD system

Figure 13. Airflow pattern and temperature contour in Case A-DV-1-1 with DV system

The final supply air parameters (supply air temperature and supply airflow rate) were obtained, as listed in Table 5.

Table 5. Final supply air parameters in 20 simulated cases

Final parameter s	Case A-MV-1-1	Case A-MV-1-2	Case A-MV-1-3	Case A-MV-1-4	Case A-MV-1-5	Case A-MV-1-6	Case A-MV-2-1	Case A-MV-3-1	Case A-UFAD-1-1	Case A-DV-1-1
Supply airflow rate (L/s)	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5
Supply air temperature (°C)	15.0	9.5	15.4	13.0	15.3	15.6	19.0	18.0	19.6	24.0
Final parameter s	Case B-MV-1-1	Case B-MV-1-2	Case B-MV-1-3	Case B-MV-1-4	Case B-MV-1-5	Case B-MV-1-6	Case B-MV-2-1	Case B-MV-3-1	Case B-UFAD-1-1	Case B-DV-1-1
Supply airflow rate (L/s)	26.3	40.0	24.2	31.2	24.8	24.2	16.8	19.0	15.5	4.6
Supply air temperature (°C)	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0

Using Eq. (5), the LCL values in the 20 cases were obtained with the final supply air parameters, as shown in Fig. 14.

Figure 14. LCL with two different calculation procedures by CFD simulation

It can be observed in Fig. 14 that the LCL values calculated using Procedure CAV and Procedure VAV are almost the same, and the difference is less than 3.2% in the 20 cases. In combination with the experimental data shown in Fig. 5, the results further prove that the LCL values using the two calculation procedures could be considered as identical. This is because that the change of supply airflow rate is usually not so much in the area of indoor ventilation so that the difference of flow pattern by using Procedure VAV and Procedure CAV is not so big. Therefore, the LCL by using Procedure VAV and Procedure CAV is almost the same, and only the LCL calculated using Procedure CAV by CFD simulation is used in the following part.

4.3.2 Influencing factors of LCL

A quantitative analysis on the influencing factors (i.e., the ventilation system,

heat source distribution, and target zone) of LCL was conducted, and the results are shown in Figs. 15–18.

Figure 15. LCL with different ventilation systems

Figure 16. LCL with different heat source locations

Figure 17. LCL with different locations of the target zones

Figure 18. LCL with different sizes of the target zones

The conventional cooling load means the cooling load in the ideal uniform indoor environment. It was obtained by summing the heat gain of two lights, three gray boxes, and the envelope in the ideal uniform indoor environment with 26 °C everywhere. The conventional cooling load for MV, UFAD and DV was all calculated by the above method, which was the same 281 W as shown in the white bars of Figs. 15-18. The LCL in the ideal uniform indoor environment is equal to the conventional cooling load. Therefore, the conventional cooling load was used as a reference value for comparing with the LCL in the other cases.

Fig. 15 illustrates the LCL in Case A-MV-1-1, Case A-UFAD-1-1, and Case A-DV-1-1 with three different ventilation systems (the target zone and heat source location are the same). It is evident that the ventilation system has a huge impact on the LCL. In Case A-DV-1-1 with DV, the inlet is closest to the target zone (Zone D); hence, its contribution to cooling the target zone is the highest. Therefore, the LCL in Case A-DV-1-1 is the lowest at 56.7 W (79.8% lower than the conventional cooling load), which represents a large potential for cooling amount reduction and energy saving if there is no recirculation of return air. In Case A-MV-1-1 with MV, the supplied cold airflow is far away from the target zone; hence, its contribution to cooling the target zone is the smallest. Therefore, the LCL in Case A-MV-1-1 is the highest at 311.8 W, which is 10.9% higher than the conventional cooling load. This implies that more effort is required for cooling the target zone in comparison with that in the uniform indoor environment. The LCL value in Case A-UFAD-1-1 is in the middle between Case A-MV-1-1 and Case A-DV-1-1.

Fig. 16 illustrates the LCL in Case A-MV-1-1, Case A-MV-2-1, and Case A-MV-3-1 with three different heat source distributions (the ventilation system and target zone are the same). In Case A-MV-2-1 with Location 2, the three heat sources are located in Zone C, which is close to the exhaust diffuser; thus, the heat from the heat sources can be exhausted quickly. Therefore, the LCL in Case A-MV-2-1 is the lowest at 198.4 W (29.4% lower than the conventional cooling load). In Case A-MV-1-1 with Location 1, the three heat sources are located in Zone E, which is close to the target zone; thus, they are most capable of heating the target zone. Therefore, the LCL in

Case A-MV-1-1 is the highest at 311.8 W (10.9% higher than the conventional cooling load). In Case A-MV-3-1 with Location 3, the three heat sources are located in Zone B, which is in the influenced region of supply cold air jet; thus, it is not easy to heat the target zone (Zone D). Therefore, the LCL is 226.7 W, which is 19.3% lower than the conventional cooling load.

Fig. 17 illustrates the LCL in Case A-MV-1-1, Case A-MV-1-2, and Case A-MV-1-3 with three different locations of target zones (the ventilation system and heat source distribution are the same). In Case A-MV-1-3, Zone F (target zone) is at the back of the heat sources and close to the exhaust diffusers; hence, it is relatively easy to be cooled. Therefore, the LCL in Case A-MV-1-3 is the lowest at 300.4 W (6.9% higher than the conventional cooling load). In Case A-MV-1-2, the three heat sources are located in the target zone (Zone E). The heat released by the heat sources increases the surrounding air temperature rapidly so that it is very easy to heat the target zone. Therefore, the LCL in Case A-MV-1-2 is the highest at 467.6 W (66.4% higher than the conventional cooling load). The LCL value in Case A-MV-1-1 is in the middle between Case A-MV-1-2 and Case A-MV-1-3.

Fig. 18 shows that the LCL increases with decreasing size of target zone. The LCL in Case A-MV-1-2 with Zone E is the highest at 467.6 W (66.4% higher than the conventional cooling load), because the air temperature around the heat sources is very high due to the convection heat (as for radiation, it will also form the convection heat finally). The LCL in Case A-MV-1-6 with the entire room is the smallest at 294.7W, which is slightly higher than the conventional cooling load. If the heat source

is also distributed uniformly in this case (the entire room as the target zone), the indoor environment will be uniform and the LCL will be equal to the conventional cooling load.

5 Conclusions

In this study, a general cooling load oriented to local requirements, local cooling load (LCL), was proposed in a non-uniform indoor environment when only the local air zone needs to be controlled. The concept and calculation method of LCL was introduced, and then the influences of differentiating ventilation systems, heat source locations, and target zones on the LCL were analyzed by experiment and simulation. The following primary conclusions could be drawn:

(1) The LCL values obtained by using Procedure CAV and Procedure VAV are almost the same. Therefore, the concept of LCL can be used to get further understanding of the cooling load and design the air terminals in a general non-uniform indoor environment.

(2) The LCL in the non-uniform indoor environment oriented to local zone can be less than, equal to, or greater than the conventional cooling load in the traditional uniform indoor environment, which subverts the existing knowledge of cooling load in the non-uniform environment oriented to occupied zone (UFAD, DV and so on) that it must be less than the conventional cooling load owing to the smaller target zone.

(3) The ventilation system, heat source location, and target zone have a large

impact on LCL. The differences resulting from differentiating ventilation systems, heat source locations, and target zones were as much as 79.8%, 29.4%, and 66.4%, respectively, in the simulated 20 cases.

The concept and results of LCL will benefit the design of ventilation systems and air terminals in a non-uniform indoor environment oriented to local requirements.

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