Diffuse ceiling ventilation for buildings: A review of fundamental theories and research methodologies

Wentao Wu a, b, Nari Yoon b, Zheming Tong a, c, *, Yujiao Chen b, Yang Lv d, Torbjørn Ærenlund e, Jingru Benner f, **

a State Key Laboratory of Fluid Power and Mechatronic Systems, Zhejiang University, Hangzhou 310027, China
b Harvard Graduate School of Design, Harvard University, Cambridge, 02138 MA, USA
c School of Mechanical Engineering, Zhejiang University, Hangzhou 310027, China
d School of Civil Engineering, Dalian University of Technology, Dalian 116024, China
e Noveco Marine and Offshore A/S, 4600 Koege, Denmark
f College of Engineering, Western New England University, Springfield, 01119 MA, USA

* Corresponding author.
** Corresponding author.
E-mail addresses: tzm@zju.edu.cn (Z. Tong), Jingru.benner@wne.edu (J. Benner).

ABSTRACT

Buildings consume more than 40% of global energy use and ventilation is one of the largest source of energy consumption. Sustainable design requires choosing energy efficient ventilation strategies. Diffuse ceiling ventilation (DCV) has a great energy saving potential due to the low pressure drop (~2 Pa) through the ceiling panel. A DCV system has three components: plenum, suspended ceiling and ventilated room. Conditioned air is supplied to plenum, then diffuses into the ventilated room through the suspended ceiling made of porous materials. The system can be designed to handle high cooling loads without inducing thermal discomfort. This review references research articles on DCV published from 2008 to 2018 to highlight the research outcomes and to identify the research gaps. One major objective of this review paper is to document simplified theoretical modelling methods for proposing quick DCV system design tool. The flow in the plenum can be described as impingement jet over porous materials. A design procedure is proposed to determine the size and number of nozzles. The heat transfer in the porous ceiling is treated as two-phase energy transport. Buoyancy force generated by the heat sources in the room has been identified to drive the air flow circulation, which motivates the thoughtful review of fundamental theories of thermal plumes in a stratified environment. The major task on thermal plumes is to calculate the height and induced volume flow rate, which are summarized according to the type of heat sources, e.g., point or area sources. The principle behind heating efficiency of DCV might be explained by theories of turbulent fountains. The rising height of warm air coming out of diffuse ceiling is determined by the source Froude number. The popular research methods to study DCV system are full-scale experiments and CFD modelling. Full-scale experiments are often used to evaluate the performance of the DCV system based on thermal comfort, indoor air quality and energy efficiency. On the other hand, CFD modelling is used for parametric analysis to improve the design of the DCV system. Finally, future research on DCV is discussed.

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<td>$F_0$</td>
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<tr>
<td>The Rossland mean extinction coefficient</td>
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<td>The thermal conductivity of the air</td>
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<td>The thermal conductivity of the solid phase</td>
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**1. Introduction**

To achieve sustainable development goals (Olawumi and Chan, 2018), the impacts of buildings on natural environment and people must be addressed. As buildings consume more than 40% of global energy use (Gamarra et al., 2018) and account for about 33% of global greenhouse emissions (Zhan et al., 2018). One of the largest sources of energy consumption in buildings is the HVAC (Heating, Ventilation and Air Conditioning) systems (Sun et al., 2018). Although the diversity of buildings and their distinct functions require different energy conservation strategies, one efficient approach is to develop new technologies for HVAC, especially ventilation. Sustainable building design requires choosing energy efficient ventilation strategies (Chenari et al., 2016). Innovative ventilation systems are able to reduce fan energy use and improve wellbeing and good health of indoor occupants. Wellbeing is an important goal of sustainable development (Balaban and Oliveira, 2017). The context of wellbeing in indoor environment often
refers to thermal comfort and indoor air quality (IAQ). The two factors must be taken into consideration when a ventilation system is designed for a building. The thermal comfort can be rated by the level of cold draught due to air motion (Fanger and Christensen, 1986), asymmetrical thermal radiation (Olesen, 1985), PMV/PPD (Fanger, 1972) and many other environmental indices such as operative temperature. The IAQ is usually indicated by the concentration of indoor air pollutants, such as organic ingredients (vegetation pollens, etc.), inorganic components (asbestos, sulfate, nitrate, etc.), and microorganisms (fungi, bacteria, mites, etc.) (Lv et al., 2018; Tong et al., 2016a & b; Dounis and Caraiscos, 2009).

The IAQ can be evaluated by air exchange effectiveness and age of air. One way to maintain thermal comfort and IAQ is to distribute the air in favorable flow patterns. Mixing ventilation (MV) is the most widely used air distribution method to control indoor airflow directions (Shan et al., 2016; Cao et al., 2014). MV supplies air with high momentum from ceiling or side wall to mix with room air and thus to dilute the contaminant concentration. This ventilation method can be used to cool or heat the entire occupied space with large cooling or heating load. However, the high momentum in MV can lead to cold draught in occupied zone (Griefahn et al., 2002). Another disadvantage of the system is that the most polluted area is within the occupied zone because air is supplied from ceiling level. In order to achieve higher IAQ in the occupied zone, displacement ventilation (DV) can be designed by supplying air from lower part of room directly to the occupied zone (Novoselac and Srebric, 2002). The contaminant in the occupied zone is displaced by fresh air. Compared to MV, momentum in DV is smaller and the flow circulation in the room is driven by buoyancy force. The temperature difference between supply and return should not be too large for room with large cooling loads. The increased high momentum can cause cold draught at lower legs (Melikov et al., 2005) and discomfort in the near zones. To satisfy room thermal comfort, the maximum cooling load that a single DV can cope with is up to 30 W m$^{-2}$ (Behne, 1999). To increase the ability of DV to handle higher cooling loads, it can be combined with a radiant cooling ceiling. Even so, the combined system can only deal with cooling loads up to 100 W m$^{-2}$ without causing cold draught (Behne, 1995). The cooling load of a room with large glazed area and internal heat gains can exceed 200 W m$^{-2}$. Restaurants and entertainment rooms on newly designed cruise vessels - AIDAPRIMA (Novenco, 2016) have cooling loads varying from 100 to 226 W m$^{-2}$. In addition to the cold draught problem in MV and DV, larger cooling loads result in larger airflow rates, which require larger diffusers and ductworks. The pressure drop through diffusers is also increased thus energy consumption would be considerably increased for fan operation. The inefficiency of MV and DV in handling high cooling loads calls for energy efficient ventilation systems, which, meanwhile, are capable of keeping indoor thermal comfort and IAQ.

A new ventilation system - diffuse ceiling ventilation (DCV) draws much attention in recent years as it is featured with low energy consumption of fan without comprising on thermal comfort and IAQ (Nielsen and Jakubowska, 2009). A DCV system has three major components (Fig. 1): plenum, suspended ceiling and ventilated room. The plenum is the space between the structural (main) ceiling and the suspended ceiling. There are mainly three types of dropped ceilings (Zhang et al., 2016c) (Fig. 1). The first type is perforated ceiling plate. The second type is to supply air through connection slots between ceiling tiles. The third type is porous material, made of large particles with diameter varying between 1 and 2 mm, such as mineral wool. A DCV system firstly supplies fresh or conditioned air into the plenum. Due to the pressure difference between the plenum and the ventilated room, the air is diffused through the entire porous ceiling into the ventilated room. The air exhaust vent is usually located in the ceiling or on the wall below the ceiling. The principle of DCV is to diffuse the supplied air through a porous ceiling into a room with low momentum and to use a large portion of the ceiling as a single air terminal device. An expected advantage of DCV over MV and DV is that the large ceiling inlet permits to supply high volumes of air without causing draught. Meanwhile, the low resistance of large ceiling inlet would reduce the fan energy use. These merits have been reported by the research summarized in Table 1. Other advantages of DCV include low cost on ductworks and diffusers by using plenum and suspended ceiling to distribute air and low noise level by using acoustic ceiling materials. Despite of the reported advantages of DCV, the system is not widely applied nowadays. The main reason is that the fundamental theories behind such a system are not well understood and documented. There exists a comprehensive review on DCV (Zhang et al., 2014), which focuses on the thermal comfort and the design chart for DCV. Much new research (Table 1) on DCV have been published since 2014, which have not been summarized. Therefore, the first objective of this review is to document the underlying physics determining the air motion in a room with a DCV system. The theoretical knowledge includes impingement jet in plenum, heat transfer in porous ceiling materials, thermal plumes, and turbulent
<table>
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<td>The pressure drop across the ceiling is (continued on next page)</td>
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<tr>
<td>Hvid and Terkildsen (2012) (Conference)</td>
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<td>Hvid and Svendsen (2013) (Journal)</td>
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<td>Full-scale measurement; CFD</td>
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<td>Reference (Paper type)</td>
<td>Building type</td>
<td>Building size (m × m × m)</td>
<td>Location</td>
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<td>Function</td>
<td>Main method</td>
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<tr>
<td>Zhang et al., (2014) (Journal)</td>
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<tr>
<td>Zhang et al., (2015a) (Conference)</td>
<td>office</td>
<td>4.8 × 3.3 × 2.72</td>
<td>Denmark</td>
<td>III</td>
<td>28.4</td>
<td>Cooling</td>
<td>Full-scale measurement; CFD</td>
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<tr>
<td>Zhang et al., (2015b) (Conference)</td>
<td>office</td>
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<td>Denmark</td>
<td>III</td>
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<td>Cooling</td>
<td>Full-scale measurement</td>
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<td>Zhang et al., (2015c) (Conference)</td>
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<td>4.8 × 3.3 × 2.72</td>
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<td>III</td>
<td>28.4</td>
<td>Heating</td>
<td>Full-scale measurement</td>
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cooling and heating to an office all year around. Mikeska and Fan (2015) (Journal) test room 6.05 × 3.25 × 2.65 Denmark II 46 Cooling Full-scale measurement; CFD Conduct research on DCV without crack flow using acoustic ceiling gypsum boards with airtight connections.

Yu et al, (2016) (Journal) office 4.8 × 3.3 × 2.72 Denmark III 30 Cooling Full-scale measurement Propose control strategy for a novel system integrating natural ventilation, DCV and TABS.

Zhang et al, (2016a) (Journal) classroom 4.8 × 3.3 × 2.72 Denmark III 140–154 Cooling Full-scale measurement; CFD Investigate the influence of ceiling opening area, cooling loads, heat source locations and room height on the performance of DCV.

Zhang et al., (2016b) (Journal) office 4.8 × 3.3 × 2.72 Denmark III 28.4 Cooling Full-scale measurement; CFD Evaluate the performance of DCV integrated with radiant ceiling system using CFD modelling.

Zhang et al., (2016c) (Report) – – I; II; III – – Review Provide a design guide on how to design DCV and how to integrate DCV with natural ventilation and TABS.

Zukowska-Tejsen et al., 2016 (Conference) office 16.8 × 5.9 × 2.6 Denmark II 6–116 Cooling Full-scale measurement Test the performance of DCV (1.2–17.9 h⁻¹) in a real office environment with regard to thermal comfort and limitations in cooling capacity.

Hviid and Lessing (2016) (Conference) classroom 12 × 6.3 × 3.6 Denmark II – Cooling Full-scale measurement Characterize the mean heat transfer coefficients of the plenum in a building with DCV.

Kristensen et al. (2017) classroom 7.8 × 6.3 × 2.98 Denmark III – Cooling Full-scale measurement Measure and calculate CO₂ concentration and ventilation efficiency in a Danish classroom with an exhaust-driven mechanical DCV system (1–1.4 h⁻¹).

Lestinen et al. (2018a) test room 5.5 × 3.8 × 3.2 Finland I 40–80 Cooling Full-scale measurement Investigate the airflow and turbulence characteristics in a test room with DCV.

Lestinen et al. (2018b) test room 5.5 × 3.8 × 3.2 Finland I 40–80 Cooling Full-scale measurement Extend the study of Lestinen et al. (2018a) to a test room with asymmetrical distribution of thermal loads.

Note: I — Perforated plates; II — Tiles (slot between them as inlet); III — Porous material (mineral wool).
fountains in ventilated room. The second objective of the review is to untangle the research states and to summarize the research methodologies for DCV. This review intends to offer insight into such a system and to promote the development of simple analytical models to design a DCV system.

2. Methods

This review is based on a systematic and comprehensive literature search using the following academic databases: Web of knowledge, Scopus and ScienceDirect, ResearchGate, Google Scholar, and CambridgeCore. Published research on DCV, turbulent jets, plumes and fountains, and heat transfer in porous material is the focus of the literature search. Articles on DCV are examined from 1960 to 2018 and most of these papers are published from 2008 to 2018 (Table 1). Since most of the research on DCV has been conducted in Scandinavian countries, the personal websites of major researchers are visited to include conference papers, technical reports and academic theses. Fundamental theories such as thermal plumes have already been established since 1950s but remains a hot research theme nowadays. Therefore, the search for this part covers the articles from 1950s to 2015, mainly peer-reviewed papers, books and PhD theses. However, scarce research has linked these theories to DCV. Since this topic is still under development, works that potentially provide research methods for DCV are also considered.

This review is sectioned as follows. The paper starts to tell the history of DCV (Table 1) in Section 3. The propose of DCV can date back to 1960s. The development since 1960s can help to reveal the research trend and direct innovations. Then research gaps on DCV based on the current publications are discussed in Section 4 and listed in Table 1. Furthermore, thermal plumes are identified to be the driving force for air circulation in a ventilated room with DCV systems (Nielsen et al., 2010). The airflow patterns in the plenum depends on the configurations of nozzles or diffusers and can be described by impingement or wall jet theories. These fundamental fluid dynamic theories for DCV are documented in Section 5, which also presents the heat transfer process in porous ceiling. Section 6 discusses the full-scale experiments and computational fluid dynamics (CFD) as research tools to understand DCV. Last but not least, potential future research directions are given in Section 7.

3. Thermal comfort, IAQ and energy saving potential of DCV

Most of the published literature (Table 1) recognized that the DCV system has been used for agricultural buildings in Denmark since 1990s (Jacobsen, 2008). It is due to the fact that more than 60% of the finishing pig housing in Denmark is installed with DCV systems (Jacobsen, 2008). However, the concept of DCV can date back to 1960s when Professor John Rydberg proposed the idea of injecting cooling air through perforated ceilings (Rydberg, 1962a, 1962b, 1963, 1965, 1966). These articles have been either written in Swedish or published in Swedish journals, and hence have not been recognized broadly. In early 1990s, injection of air through perforated ceiling has been applied to cool and heat public rooms (restaurants, lounges, etc.) on marine ships, such as MS Prinsesse Ragnhild (Novenco, 1991) and Princess Cruises (Novenco, 1995). The system was designed by marine and offshore HVAC supplier – Novenco (2018) and was called ‘Raindrop ceiling’. Today it is often referred to DCV.

3.1. Thermal comfort

The first comprehensive study of DCV on thermal comfort was presented by Jacobsen (2008). The use of DCV in livestock buildings is to provide acceptable climatic conditions to improve animal welfare. As piglets are very sensitive to air temperature. Jacobsen (2008) used full-scale, wind tunnel experiments and numerical methods to discuss the performance and principles of DCV in pig stables. The DCV system was able to maintain a stable and comfort micro-environment in the animal occupied zone irrespective of changes in ambient temperature and air exchange rate. Jacobs et al. (2008) applied the DCV concept to residential buildings – a Dutch class room and their laboratory experiments showed that the DCV system was draught free. Nielsen and Jakubowska (2009) carried out experiments in an office and proposed design charts (relation of temperature difference between return and supply with supply airflow rate) to confirm that DCV was able to handle high thermal loads without causing draught. Yang (2011) investigated the thermal comfort of a DCV system using slot between ceiling tiles as inlet installed in an office room. DCV can provide a high level of thermal comfort for both high (5.1 h⁻¹) and low (3.5 h⁻¹) air changes per hour (ACH) in terms of ventilation effectiveness, vertical air temperature gradient, draught rating, and PMV/PPD. Hvitt and Terkildsen (2012) measured the thermal comfort of DCV in a real classroom and concluded that DCV provided a draught free environment and created a uniform temperature distribution in the occupied zone. They also observed negligible discomfort caused by the radiant asymmetry of a warm or cool ceiling. Fan et al. (2013) created design charts for DCV based on validated numerical modelling and further investigated the influence of ACH and heat sources on thermal comfort. Their study showed that only 5% and 8% of people felt discomfort for an ACH of 3.5 h⁻¹ and 5.0 h⁻¹, respectively, based on calculated PMV and PPD. However, Mikeska and Fan (2015) and Zhang et al. (2016b) reported high risk of draught near ankle level and recommended to use DCV in low ceiling room. Lestinen et al. (2018a, b) studied the airflow characteristics (mean flow and turbulence) in an occupied zone with symmetrical or asymmetrical heat load distribution. They monitored a large-scale circulating airflow from the heated window to the corridor resulting in a higher draught rate in the corridor side. In general, DCV provides high thermal comfort even for high air flow rates. Draught could occur near ankle level due to asymmetrical load distributions.

3.2. IAQ

Jacobs et al. (2008) also conducted a pilot full-scale experiment in a classroom with a perforated dropped ceiling. The pilot study showed that DCV can significantly reduce the peak CO₂ concentration in the classroom from 2700 ppm to 1200 ppm without causing cold draught by using high supply airflow rate. The tracer gas measurements of Fan et al. (2013) showed that the ventilation effectiveness by DCV is 0.9–1 in the breathing zone, which means DCV can generate good air mixing in the occupied zone. Mikeska and Fan (2015) carried out experiments on DCV using acoustic gypsum boards with airtight connections to prevent undesirable crack flow. Their tracer gas measurement showed that the average ventilation efficiency was as high as 0.84 without draught issues. Zhang et al. (2016c) summarized the previous works on DCV and found that no stagnant zones or shortcut air circulations were reported. Kristensen et al. (2017) carried out a field study in classroom with DCV under real operating conditions. They found that DCV had a tendency toward displacement ventilation within the occupied zone, which indicates good IAQ in the occupied zone. However, Kristensen et al. (2017) also reported that contaminated air might recirculate inside the room due to the bi-directional flow. Good IAQ can be achieved by DCV in terms of low CO₂ concentration and high ventilation efficiency. Caution should be taken to prevent contaminant room air circulating back to the room.
into the plenum.

3.3. Energy saving potential

Jacobs and Knoll (2009) presented the energy saving potential of DCV systems in schools. The electricity use is 5–10 kW/m³ in Dutch classrooms with conventional AC systems. With DCV in similar classrooms, electricity consumption is lowered to 0.04–0.5 kW/m³. One of the reasons for low energy consumption is due to the low pressure nature of DCV systems. Hviid and Svendsen (2013) studied two types of perforated dropped ceilings as diffuse ventilation inlet. The pressure drop of DCV was smaller than 2 Pa. The low-pressure drop allowed a reduction in fan power (Terkildsen and Svendsen, 2011). The low-pressure drop of DCV is attributed to the reduced ductwork and increased inlet area. The use of the plenum reduces the need for ductwork and air terminals in the room and the large volume of the plenum reduces the pressure loss of the air flow. Consequently, the pressure difference required to deliver air by DCV is much lower than that by a conventional ventilation system. On the other hand, the entire ceiling used as air inlet further reduces the pressure drop compared with conventional air terminals such as ceiling diffusers. The major factors influencing the pressure drop are the ceiling type and the opening area. The pressure drop for three types of ceilings and three opening areas is reproduced in Fig. 4, which shows that the pressure drop is usually very small. Zhang et al. (2016b) investigated the influence of the ceiling opening area (100%, 50% and 18%) on the performance of DCV based on experiments and the influence of heat locations and room

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**Fig. 2.** (a) Impingement jet theory; (b) design of nozzle configurations in a plenum.
height on the performance of DCV based on CFD simulations. Even an 18% opening area can handle high cooling load of 140–154 W m⁻² with very high flow rate of 10.5 h⁻¹. Zukowska-Tejsen et al. (2016) confirmed the good performance of DCV by increasing cooling load up to 116 W m⁻². Another reason for low energy consumption is the radiant cooling potential. Hviid and Svendsen (2013) showed that the ceiling surface temperature was in the range of 14.3–17.7 °C. The low surface temperature increased radiant heat transfer between the ceiling and other warm surfaces in the room. Chodor and Taradajko (2013) showed that 25% of the heat load was removed by radiation and confirmed the radiation cooling potential of the diffuse ceiling due to the large supply area and the low ceiling temperature.

In part of the spring, fall or winter, if the outdoor air is directly supplied to cool the rooms with conventional AC systems, it is often pre-heated by hot water or electrical heating coils to prevent cold draught. Preheating can increase up to 10% of energy use (Labs21, 2005). Zhang et al. (2015a) found that the cold draught did not occur even with an extremely low supply air temperature –6 °C in a room with DCV. This indicates that the outdoor air might be directly supplied to rooms through the diffuse ceiling for cooling even in winter period. Without using reheating, a significant amount of energy use can be reduced. Zhang et al. (2015a) infers that DCV has a high night cooling potential. The heat stored in a building structure during the day can be purged out in the night by DCV like conventional ventilation systems. However, the porous ceiling material (Fig. 1) directly exposing to the supplied outdoor air has a capacity to store the night cooling and improves the efficiency of night cooling. Hviid and Lessing (2016) reported that the thermal convective resistance at the surfaces in the plenum can be reduced to half of the value in a room without DCV, which showed the free cooling potential by using ceiling thermal mass. Therefore, the DCV system has a big energy saving potential due to the possibility of supplying air at extremely low temperature and the thermal mass of diffuse ceiling to improve night cooling efficiency.

Another approach to increase the energy saving potential of DCV is to integrate with other building systems. Yu et al. (2015a, b, 2016) and Zhang et al. (2015a, c) carried out experimental investigation of cooling performance of DCV coupled with thermally activated
building system (TABS) and natural ventilation. The integration of DCV with TABS and natural ventilation reduced the risk of draught. DCV increased the heat transfer coefficient of TABS under heating conditions but reduces the coefficient under cooling conditions. However, the integration of the three systems still had large energy saving potential compared with other systems. Compared with traditional air-based heating and cooling systems, the integration of the three systems has less annual primary energy use when the internal heat load is above 30 W m$^{-2}$, and the energy saving increases with the increase of internal heat load. When the internal heat load is 65 W m$^{-2}$, the integrated system can reduce the annual primary energy use from 63.5 to 5.5 kWh m$^{-2}$.

4. Research gap

Buoyancy force generated by the heat sources in the room has been identified to drive the airflow circulation (Nielsen et al., 2010; Petersen et al., 2014). The uprising thermal plumes from heat sources such as people and electrical appliances eventually meet the downward airflow from the diffuse ceiling. Concerns arise on the risk of a reverse flow from the room to the plenum due to the strong thermal plumes from heat sources and the low momentum of the supplied airflow (Kristensen et al., 2017). Meanwhile, the thermal plumes between different heat sources also interact with each other. The plume interaction makes the prediction of the airflow pattern rather complicated. In addition, the distribution of heat sources in the room is dynamic, e.g., people walk around in the room and might favour the formation of unstable airflow patterns. All above reasons make it difficult to propose simplified physical equations for quick system design.

Current research focuses on the characteristics of the airflow in the ventilated space. Limited attention has been paid to the design of plenum to generate a uniform airflow and temperature distribution over the ceiling. Nowadays, CFD is mainly employed to look into the influence of plenum height, inlet location and configurations. Theoretical calculations based on simplified governing equations have not been adequately discussed in literature.

The investigations made so far focus mainly on the cooling performance of DCV. The heating efficiency under winter condition is also crucial to extend the application of DCV. For instance, when DCV is designed for restaurants on marine ships with large heat loads, the system has to run for both heating and cooling functions. As water-based radiators are usually not designed on ships. When warm and light air is supplied from the ceiling, it opposes the gravity direction and therefore, the heat might accumulate near the ceiling and cannot be effectively advected to the occupied zone. The research on heating performance of DCV including the theoretical principles behind it is currently missing. The influence of radiant heating on thermal comfort is also not well understood.

Due to the above-mentioned problems, a review on the theoretical background of DCV is in urgent need in order to promote the use of DCV in broader climate zones and other types of buildings (besides offices and classrooms) with high heat loads.

5. Simplified theoretical modelling

The airflow patterns and temperature distribution in a room with DCV are described by Navier-Stokes and energy transport equations, as in any other fluid flow problems. However, the non-linearity of the governing equations and the turbulent nature make it impossible to find mathematical solutions of closed form nowadays. An alternative is to get numerical solutions of the transport equations, namely, CFD modelling, which will be discussed in the next section. CFD requires well-trained and specialized engineers and might be still difficult for designers to use. For most applications in building physics, the computation can take up to several days, and is usually too time tight for design purpose. Therefore, researchers should turn to propose assumptions that allow considerable simplification of governing equations so that simple and quick solutions can be provided to design engineers. In Section 5.1, simplified theoretical models for DCV is discussed in detail. The flow and heat transfer involved in DCV is discussed based on the three components: plenum, porous ceiling and ventilated room (Fig. 1). In Section 5.1, we discuss the airflow in the plenum based on the impingement jet theory. In Section 5.2, the heat transfer principle in the porous materials is presented. Thermal plumes from heat sources not only determine the thermal stratification but also set the pattern of air movement in the room (Hunt, 2011). Therefore, the theories of thermal plumes are reviewed in Section 5.3. If DCV is designed for heating, the warm air drops from the ceiling might be considered as a turbulent fountain (Hunt and Burridge, 2015), which is briefly presented in Section 5.4.

5.1. Impingement jet in plenum

In the plenum, air is usually supplied through nozzles by means of high momentum, which creates a turbulent jet flow. If nozzles are mounted at position A (Fig. 1), the turbulent jet impinges on the porous ceiling. There has been some research on the mechanism of the penetration of jets through porous materials (Weiner, 1993; Walklate et al., 1996; Cant et al., 2002). The jet behaviour from the nozzle to the porous wall is between free jet and impinging jet on a solid surface depending on the porosity. For low porosity ($\phi \leq 0.41$), the jet structure is very similar to jet impinging on a solid surface. For high porosity ($\phi \geq 0.65$), the flow type may be described by free jet. For DCV, the dropped ceiling can have a porosity up to 0.65 (Zhang et al., 2016a). Therefore, the theory for free and impinging circular jets is a promising tool to design the nozzle configurations and to determine the optimum height of the plenum. For impinging jet (Fig. 2a), former research (Beltaos and Rajaratnam, 1973) has recognized three distinct flow regions: a free jet region (I), an impingement region (II) and a wall jet region (III). The main objective in the free jet region is to decide its extent. In the impingement region, the static pressure is generally higher than the ambient. The pressure on the wall and the velocity fields should be determined in this region. For the wall jet region, the primary interest is to know the velocity distribution and the jet thickness. The parameters to describe the characteristics of different regions are given in Fig. 2a. The coordinate system ($r, x$) is used for region I and II, and ($r_1, x_1$) is used for region III.

The dimensionless profiles of the time-averaged velocity $u$ in $x$ direction in region I and II show self-similarity and are well described by Gaussian functions for $x/H$ less than 0.95 (Beltaos and Rajaratnam, 1974):

$$\frac{u}{u_m} = e^{-0.693(r/b_u)^2}$$

(1)

where, $H$ is the impingement height, $u_m$ is the maximum value of $u$ at $x$, and $b_u$ is the value of $r$ at which $u$ equals 0.5 $u_m$.

In region I, for $x/H$ up to 0.86, $u_m$ and $b_u$ are described by the equations of the free jet:

$$u_m = \frac{6.3}{U_0} \frac{x}{d}$$

(2)

$$b_u = 0.097x$$

(3)

where $U_0$ is the jet velocity at nozzle outlet, and $d$ is the diameter of the nozzle.
In region II, the following empirical formula can be used to calculate $u_m$:

$$
\frac{u_m}{U_0} = 19.53 \frac{d}{H} \sqrt{1 - \frac{x}{H}}
$$

(4)

As above-mentioned, the static pressure in the impingement region is larger than the ambient pressure. The pressure distribution is also of Gaussian profile:

$$
\frac{p}{p_m} = e^{-0.693(r/b_p)^2}
$$

(5)

where, $b_p$ is the value of $r$ at which $p$ equals 0.5$p_m$ and $p_m$ is the maximum value on the axis.

The pressure on the impingement wall surface is of the form:

$$
\frac{p_w}{p_s} = e^{-114(r/H)\lambda^2}
$$

(6)

where, $p_w$ is the wall surface pressure, and $p_s$ is the stagnation pressure.

The stagnation pressure is given by:

$$
\frac{p_s}{\rho U_0^2/2} = \frac{50}{(H/d)^2}
$$

(7)

where, $\rho$ is the air density. According to the experimental results of Beltaos and Rajaratnam (1974), $x/H = 0.86$ is the end of region I and the beginning of region II. $p_w$ approaches ambient value at $r = 0.22H$, which could be taken as the end of the impingement region.

After impingement, assuming the total pressure remains constant and equal to $p_s$.

$$
p_{m1} = p_s - \rho u_{m1}^2
$$

(8)

where, subscript 1 denotes coordinate system ($r1, x1$).

For $x1/H$ greater than 0.92, $p_{m1}$ is given by:

$$
p_{m1} = 10 \frac{x1}{H} \quad 9.0
$$

(9)

Using Eqs. (6)–(8), the equation to describe $u_{m1}$ is of the form:

$$
\frac{u_{m1}}{U_0} = 7.07 \frac{d}{H} \sqrt{1 - e^{-114(r/H)^2}}
$$

(10)

The variation of the wall shear stress in the impingement region is predicted as $\rho u_1^2$, and $u_1$ is the shear velocity, which is given by

$$
u_{1} = \frac{d}{H} \sqrt{0.029 \left(1 - e^{-114(r/H)^2}\right)(r/H) - 1.51(r/H)e^{-114(r/H)^2}}
$$

(11)

In region III, the velocity distribution is given by the well-known power law (Gauntner et al., 1970).

$$
\frac{u_1}{U_0} = \left(\frac{u_{m1}}{U_0}\right)^{1/n}
$$

(12)

where, $n$ is between 7.5 and 15.

For $x1$ larger than 0.22$H$ and smooth wall, $u_{m1}$ is given by

$$
\frac{u_{m1}}{U_0} = \frac{1.4}{(x1/d)^{1.12}}
$$

(13)

The spread of the jet, half-velocity of the wall jet can be represented by

$$
\frac{b_1}{d} = C_{b1}(\frac{x1}{d})^{0.95}
$$

(14)

$$
C_{b1} = 0.0081 \frac{H}{d} + 0.0864
$$

(15)

Fig. 3a shows the good agreement between the dimensionless velocity profiles from experiments and Eq. (1). Fig. 3b shows that Eqs. (2) and (4) can well predict the air velocities at the centreline of Region I and II.

The important task here is to design the size and number of nozzles (Fig. 2b) to achieve more uniform flow in the plenum as well as to calculate pressure loss from nozzles. When the impingement jet theory is applied, the crucial design criteria are the given ranges of $Y_s, U_1$ and $U_2$ in Fig. 2a. The criteria are defined according to the requirement of comfort and IAQ. The criteria are currently missing for plenum design. Providing that the required ranges for $Y_s, U_1$ and $U_2$ are given by full-scale experiments, a design procedure for nozzle configurations might be given as follow:

1. If nozzles are mounted at position A in Fig. 1, then go to (2). If nozzles are installed at position B in Fig. 1, then go to (5).
2. Calculate the porosity $\varphi$ of diffuse ceiling. For $\varphi \geq 0.65$, go to (3). For $\varphi \leq 0.41$, go to (4).
3. Based on defined ranges of $Y_s, U_1$, using Eq. (2) to decide $U_0$. The nozzle size can be determined based on the noise and the maximum pressure loss requirement. The half diagonal distance ($Y$) (Fig. 2b) between two nozzles can be calculated by $Y = 2b_w$, which is given by eq. (3). The number of nozzles can thus be decided by $Y$ and the dimension of the plenum.
4. Based on defined ranges of $Y_s, U_1$, using Eq. (2) or Eq. (4) to decide $U_0$. Eq. (10) can be used to calculate $U_2$, the range of which must be checked. The distance $Y$ can be dimensioned as follow:

$$
Y = \frac{1.22U_0 d}{U_2}
$$

(16)

5. For nozzles arranged at position B, the jet structure is confined wall jet, which is the same as region III. Eq. (9)(10)(12)(14) to design nozzles or plenum depth.

5.2. Heat transfer in porous ceiling

The possible thermal heterogeneity in the air through the porous ceiling material (Khalid et al., 2018a; Khalid et al., 2018b; Aman et al., 2016) is a major concern in this part. This is due to the fact that heat transfer in solid is different from that in the void of the porous ceiling. The presence of the porous matrix of materials also converts part of the conductive and convective heat transfer into radiative heat transfer (Mishra et al., 2015). Chen and Sutton (2005) and Khan (2006) used two coupled equations to study the two-phase heat transfer in the porous materials. The steady energy equations for the gas and solid phases are written as (Talukdar et al., 2004):

For the gas phase,

$$
\varphi \rho c_p U_1 \frac{\partial T}{\partial x_1} + (1 - \varphi) A h_c (T - T_s) = \varphi \frac{\partial}{\partial x_1} \left( \frac{\lambda}{\partial T} \right)
$$

(17)

where, $c_p$ is the specific heat of air, $T$ is the air temperature, $A$ is the
surface area per unit volume of solid, $h_s$ is the heat transfer coefficient, $T_s$ is the solid material temperature, $\lambda$ is the thermal conductivity of air.

For the solid phase,

$$\frac{\partial q_{R,i}}{\partial x_i} = (1 - \varphi) A h_s (T - T_s) + (1 - \varphi) \frac{\partial}{\partial x_i} \left( \lambda_x \frac{\partial T_s}{\partial x_i} \right)$$  \hspace{1cm} (18)

where, $\lambda_x$ is the thermal conductivity of the solid phase, $\frac{\partial q_{R,i}}{\partial x_i}$ represents the volumetric radiative source term. The radiative heat flux can be predicted by Rosseland approximation (Singh et al., 2011).

$$q_{R,i} = -\frac{4\sigma}{\pi^2} \frac{\partial T^4}{\partial x_i}$$  \hspace{1cm} (19)

where $\sigma$ is the Stefan-Boltzmann constant, $5.67 \times 10^{-8}$ W m$^{-2}$ K$^{-4}$ and $k^*$ is the Rossland mean extinction coefficient, which is given (Hsu and Howell, 1992)

$$k^* = -\frac{3}{D} (1 - \varphi)$$  \hspace{1cm} (20)

where, $D$ is the pore diameter.

The flow field $u_i$, in above equation can be evaluated by adding a pressure drop source term to Navier-Stokes equations. The pressure drop source term is given by

$$\left( \frac{v}{\varphi} + C \right) q_R$$  \hspace{1cm} (21)

where, $v$ is the kinematic viscosity of air, $U$ is the magnitude of $u_i$. $C$ is the inertia coefficient. $C$ can be easily fitted from pressure drop data given in Fig. 4, which is retrieved from Hviid and Svendsen (2013) and Zhang et al. (2016b).

5.3. Turbulent plumes in the room space

Much research in Table 1 has concluded that the airflow patterns in room with DCV are determined by the buoyancy force generated from heat sources. The heat sources in buildings include point sources (e.g., lamp) and area sources (e.g., large piece of equipment). The cold flow from the suspended ceiling might be also considered as micro-plumes from a large area source. The turbulent flow from a heat source expands cone-like outline (see Fig. 5a), which is generally called a turbulent plume (Hunt, 2011). The density in the plume is different from the ambient environment and the density difference may cause vertical bulk motion. If the buoyancy force is large enough, a thermal plume rises, and the ambient fluid entrains into the plume by turbulence transportation. These turbulent plumes rising from heat sources determine the thermal stratification and the local ventilation rate of the room.

Turbulent plumes can be classified as pure, forced and lazy plumes based on Richardson number (Hunt, 2011).

$$R_i = \frac{g' b}{U^2}$$  \hspace{1cm} (22)

where, $g'$ is the reduced gravity equal to

$$g' = g (\rho_e - \rho) / \rho_0$$  \hspace{1cm} (23)

where, $\rho_e$ is the density of ambient air, $\rho$ is the air density in the plume, $\rho_0$ is the air density at the source (Fig. 5b).

The simplified theoretical models for thermal plumes are based on the three following assumptions.

1. The density difference is assumed to be small (Batchelor, 1954).

   Hence, Boussinesq assumption is applied:

   $$\frac{\rho_e - \rho}{\rho_0} = \frac{T - T_e}{T_0}$$  \hspace{1cm} (24)

   The meaning of the temperature and fluid density in eq. (24) is given in Fig. 5b.

2. The cone-like outline indicates self-similar behaviour of the velocity and buoyancy profiles, which can be described by Gaussian functions (Yih, 1951; Rouse et al., 1952).

   $$u(x, r) = u(x) e^{-r^2 / b^2}$$  \hspace{1cm} (25)

   $$g'(x, r) = g(x) e^{-r^2 / (\phi b)^2}$$  \hspace{1cm} (26)

   These equations are in the cylindrical coordinate. The self-similarity is the same as the turbulent jet in Section 5.1. The distinction is the characteristic length scale $b$ for velocity profiles, which is the breadth of the plume. The characteristic length scale for the reduced gravity is $b$ times a constant ($\eta = 1.07$) (Hübner, 2004).

3. As mentioned above, the ambient fluid enters the plume by turbulent entrainment mechanism. The rate of fluid entrainment at any height is proportional to a characteristic velocity at that height (Morton et al., 1956; often referred to MTT in literature).

   $$u_e = -\alpha u(x)$$  \hspace{1cm} (27)

   where, $u_e$ is the entrainment velocity. The constant is around 0.093, given by MTT.

   The important parameters for turbulent plumes are volume ($Q$), momentum ($M$) and buoyancy ($B$) flux, which are defined as follow by using assumption 2:

   $$Q = \int_0^\infty u(x, r) 2\pi r dr = \pi u(x) b^2$$  \hspace{1cm} (28)

   $$M = \int_0^\infty u(x, r)^2 2\pi r dr = \frac{1}{2} \pi u(x)^2 b^2$$  \hspace{1cm} (29)

   $$B = \int_0^\infty u(x, r) g'(x, r) 2\pi r dr = \frac{\eta^2}{\eta^2 + 1} \pi u(x) g'(x) b^2$$  \hspace{1cm} (30)

   The general continuity, N–S and energy equations for turbulent plumes originating from a point source in a stratified room environment can be reduced to ODEs based on the above assumptions. These equations are given by MTT and the detailed derivation is given by Hübner (2004):

   $$\frac{dQ}{dx} = -2\pi b u_e = -2\pi b \alpha u(x) = -2^{3/2} \pi^{1/2} \alpha M^{1/2}$$  \hspace{1cm} (31)

   $$\frac{dM}{dx} = \pi \eta^2 b^2 g' = \frac{\eta^2 + 1}{2} \frac{b}{M} \frac{Q}{M}$$  \hspace{1cm} (32)
where, \( G = -\frac{\rho_s}{\rho_0} \frac{B}{h} \). The most important information on thermal plumes is the rising height and the volume flux. The solutions to above equations can provide such information and can be obtained by numerical evaluation of integrals. A ventilated room is very often stratified and hence, the solutions to stratified environment will be discussed based on the type of the heat source in the following sections. As for neutral environment, readers are referred to Awbi (2003).

5.3.1. Point source of heat

Heat sources such as a desk lamp or a laptop might be regarded as point sources. The rising height \( h \) and the volume flux \( Q \) for a point source are given as follow (MTT):

\[
\frac{dB}{dx} = -GQ
\]

\[
h = 0.410a^{1/2}F_0^{3/4}G^{-1/2}h
\]

\[
Q = 2.442a^{1/2}F_0^{3/4}G^{-1/2}Q_1
\]

where, \( F_0 \) is the source buoyancy flux, \( m^4 s^{-3/2} \); \( h \) and \( Q_1 \) are constants, equal to 2.800 and 1.705, respectively. For thermal plumes, the buoyancy flux can be calculated from the heat output following (Batchelor, 1954):

\[
F_0 = \frac{Pg}{c_pT_0}
\]

where, \( P \) is the heat output, W. Note that for heat sources such as lamps, computers and people, \( P \) shall be the convective heat gain. While MTT derived the solutions and equations for turbulent plumes in a more general context, Mundt (1992) tailored the solutions especially for a room with temperature stratifications. Awbi (2003) also summarized these solutions into his book. The equations are essentially the same as Eqs.(34),(35) and will not be repeated here.

Aksenov et al. (1998) numerically calculated the rising height of a lamp \((0.15\ m \times 0.15\ m \times 0.10\ m)\) in a room \((1.5\ m \times 1.5\ m \times 8.0\ m)\).
with a temperature gradient of 0.5 or 1.0 K m$^{-1}$. The convective heat gain is 100 W for the lamp. Aksenov et al. (1998) found out that the rising heights were 3.69 m and 2.80 m for the two temperature gradients, respectively. From equations (34) and (35), the rising heights are 4.00 m and 3.09 m, respectively. The small deviation between this study and Aksenov’s calculation can be reduced by virtual origin correction. The reason is that the lamp has a size and is not a pure point source. It is noted that the deviation between this study and Aksenov et al. (1998) is around the geometrical dimension of the lamp. Hence, the analytical solutions are reliable in this study and Aksenov et al. (1998). The reason is that the lamp has a size and the heights are 4.00 m and 3.09 m, respectively. The small deviation between this study and Aksenov’s calculation can be reduced by virtual origin correction using eq.(34) (Morton, 1959). The accuracy can be referred to the height of the thermal zone. The ejected warm air from the suspended ceiling might also be considered as thermal sources. For example, a large group of students in a crowded lecture room and a patch of floor heated by solar radiation (Fig. 5a). The cold air from diffuse ceiling might also be considered as thermal plumes of area sources. Epstein and Burelbach (2001) observed that a buoyant plume above a finite area source contracted to a neck at a height equal to half of the source radius (Fig. 5c). The behaviour of turbulent plumes from area sources can be described by a mixing layer and a vertical plume from a point source staring from a virtual origin (Kaye and Hunt, 2009). The mixing layer and the vertical plume meet at the neck. Therefore, the rising height can be obtained with a virtual origin correction using eq. (34) (Morton, 1959). The volume flux for thermal plumes of area sources can be estimated as follow (Kaye and Hunt, 2010).

\[ Q = \begin{cases} 
 6.416 \left( \frac{F_0 M_0}{Q_0} \right)^{\frac{1}{2}} x & M_0^2 x < 0.055 \\
 1.887 \left( \frac{M_0^2}{Q_0} x + 0.31 \right)^{\frac{1}{2}} Q_0^0 F_0^0 M_0^2 x > 0.055 \end{cases} \quad \text{(38)}
\]

Other sources such as line, sphere, cylinder and radiator can be found in the book of Awbi (2003) and in the work of Popiolek (1998). The equations established for neutral environment are not given here. Further work shall be done for stratified room environment. Fig. 6 shows the good agreement between measurements and eq. (38) when the scaled height is smaller than 0.055. The transition value of 0.055 indicates the far field of thermal plumes. Therefore, the second equation of Eq. (38) is essentially the same as Eq. (35). The accuracy can be referred to the height of the thermal plumes over a lamp in Section 5.3.1.

5.3.2. Area and other sources of heat

Not all sources heated in a room can be considered as point sources. For example, a large group of students in a crowded lecture room and a patch of floor heated by solar radiation (Fig. 5a). The cold air from diffuse ceiling might also be considered as thermal plumes of area sources. Epstein and Burelbach (2001) observed that a buoyant plume above a finite area source contracted to a neck at a height equal to half of the source radius (Fig. 5c). The behaviour of turbulent plumes from area sources can be described by a mixing layer and a vertical plume from a point source staring from a virtual origin (Kaye and Hunt, 2009). The mixing layer and the vertical plume meet at the neck. Therefore, the rising height can be obtained with a virtual origin correction using eq. (34) (Morton, 1959). The volume flux for thermal plumes of area sources can be estimated as follow (Kaye and Hunt, 2010).

\[ Q = \begin{cases} 
 6.416 \left( \frac{F_0 M_0}{Q_0} \right)^{\frac{1}{2}} x & M_0^2 x < 0.055 \\
 1.887 \left( \frac{M_0^2}{Q_0} x + 0.31 \right)^{\frac{1}{2}} Q_0^0 F_0^0 M_0^2 x > 0.055 \end{cases} \quad \text{(38)}
\]

The steady rising heights ($z_{ss}$) are summarized and classified by Kaye and Hunt (2006), Burridge and Hunt (2012), and Burridge and Hunt (2015).

\[ z_{ss} = \begin{cases} 
 2.46 F_0^{0.3} & F_0 \geq 4.0 \\
 2.80 F_0^{0.3} - 2.10 & 2.0 \leq F_0 \leq 4.0 \\
 0.82 F_0^{0.3} & 1.0 \leq F_0 \leq 2.0 \\
 0.81 F_0^{0.3} & 0.3 \leq F_0 \leq 1.0 \end{cases} \quad \text{(40)}
\]

The dominant physics of turbulent fountains vary with source $F_0$. Therefore, the four types of turbulent fountains described by above equations are classified as forced, intermediate, weak and very weak fountains. An intermediate turbulent fountain is sketched in Fig. 7. The sketch is based on the experimental work of Burridge and Hunt (2012). The accuracy of Eq. (40) is shown in Fig. 8. Very good agreements between experiments and Eq. (40) are found except at transition values. The discrepancy at transition values are smaller than 10%.

Fan et al. (2013) studied the performance of DCV in a cooling system.
scenario with air change of 5.1 h$^{-1}$ in a test chamber (3.6 m × 6.0 m × 3.5 m). If the same airflow rate is assumed for heating, the source $Fr_0$ is only about 0.01. The turbulent fountain in this case is very weak and the rising height cannot be described by eq. (40). In heating situation, the ceiling area used for airflow supply should be reduced or the supply airflow rate should be increased. Under the circumstance of the same source volume flow rate, the first type of DCV by supplying air through connection slots between ceiling tiles provides higher source momentum. It might be more suitable for supplying warm air. Liu and Linden (2006) studied an underfloor distribution system to supply cold air. It is found that the cold air is carried by thermal plumes generated by local heat sources (e.g., people) from lower layer to upper layer. Because of entrainment, the warm air in the upper layer falls back down into the lower layer. Thus, the heat in the upper layer is removed. In the case of DCV for heating, the thermal plumes generated by local heat sources in winter situation are able to penetrate into the turbulent fountains formed by warm air released from the suspended ceiling (Fig. 7c). The cold room air entrains warm thermal plumes and rises to the hot fountain near ceiling. Part of the hot air is cooled by the plume and then recirculates down to the occupied zone. The possibility of such air circulation shall be a future research theme to provide tools to estimate the heating potential of DCV. The theories discussed above considers mainly free turbulent fountains. In a room space with DCV, the fountain is confined by walls. Cooper (1988) and Kaye and Hunt (2007) showed that the penetration depth of the fountain downside the wall depends on the room aspect ratio. When the aspect ratio is close to one, the vertical penetration depth scales on the room height. The interesting fluid dynamics tailored for DCV is missing and worthwhile to be investigated in the future (see Fig. 8. The variation of steady rise height with source Froude number. Circles are measured values from Hunt and Burridge (2015); Solid line shows the values estimated by Eq. (40).
6. Full-scale experiments, CFD and energy modelling

Full-scale experiments are usually used to evaluate the thermal comfort, IAQ, and energy saving potential of DCV systems. Whereas, CFD is often used for parametric study. There are still some challenges in CFD, e.g., the conflict between porous media modelling and radiation modelling as well as the choice between steady and unsteady N–S equations (Mikeska and Fan, 2015). Energy modelling has not been investigated much by the researchers working on DCV.

6.1. Full-scale experiments

The simplified theoretical models have not been applied to design DCV yet. Full-scale experiments are the most common method to study the performance of DCV. Some on-site experiments have been carried out in classrooms (Jacobs and Knoll, 2009; Jacobs et al., 2008; Hviid and Petersen, 2011; Kristensen et al., 2017). Most of the investigations have been conducted in controlled environment, where real-scale rooms are used but the thermal loads and local heat sources can be well controlled (Hviid and Svendsen, 2013; Fan et al., 2013; Petersen et al., 2014; Mikeska and Fan, 2015; Zhang et al., 2016a; Lestinen et al., 2018ab). The performance of DCV can be evaluated by means of thermal comfort, IAQ and energy use, briefly mentioned in Section 1.

As for thermal comfort, the evaluation indexes are air temperature, air temperature gradient, velocity, PMV/PPD, and draught rate (DR) (Zhang et al., 2014). DR is an integrated parameter, which considers the synthetic effect of local air temperature, mean air speed and turbulence intensity. All the variables involved in DR can be easily measured by anemometer and temperature sensors (Yang, 2011; Fan et al., 2013; Chodor and Taradajko, 2013; Zhang et al., 2016a). DCV is confirmed to have a DR of 7% at high supply airflow rate, which is much lower than the threshold set by category B of 20% (CR 1752, 1998). The radiant temperature asymmetry, caused by different internal wall surface temperatures, is also often used to assess thermal comfort. The measurement of the enclosure surface temperature can be done by thermal camera or temperature sensors (Yang, 2011; Hviid and Svendsen, 2013). The radiant asymmetry ranges from 1.5 K to 8.4 K for an office room with DCV (Zhang et al., 2014). The value is much lower than 14 K, which is the threshold to satisfy indoor environment category B (ISO 7730, 2005).

As for IAQ, the evaluation indexes include gas concentration (e.g., CO₂), local ventilation rates and ventilation effectiveness. Gas concentration can be measured by applying Innova Photoacoustic Field Gas monitor coupled with a multipoint sampler (INNOVA air Tech Instruments A/S, Denmark) (Wu et al., 2012a; Fan et al., 2013). The ventilation efficiency can be calculated based on the measured gas concentration. If existing gas such as CO₂ cannot be measured or used for ventilation efficiency calculation, the tracer gas method is a reliable and accurate alternative to determine the local ventilation rate and the ventilation efficiency (Wu et al., 2012b;c; Fan et al., 2013; Hviid and Svendsen, 2013). Two types of tracer gas methods are common in local ventilation evaluation. The first type is called tracer gas decay, where the room should be totally covered initially, and a tracer gas is injected and well mixed in the room. Then the gas concentrations at different locations are monitored over time. The second type is constant release of a tracer gas. Gas concentrations are monitored after a steady state is established.

As for energy savings, the evaluation index is mainly the pressure drop over the dropped ceiling, which is usually measured by pressure transducers. Fan et al. (2013) and Zhang et al. (2016b) placed FC0510 micro-manometers in the plenum and the conditioned space to measure the pressure drop across the diffuse ceiling.

6.2. CFD modelling

Full-scale experiments have disadvantages in flexibility to change parameters. For example, the height of a plenum shall be altered in order to study its influence on the DCV design. This is usually impossible in a full-scale experimental facility. CFD modelling provides a reliable and accurate alternative to determine the local ventilation rate and the ventilation effectiveness. Gas concentrations are monitored after a steady state is established by the researchers working on DCV.
distinguish the heat transfer difference in solid and fluid phase of the porous material. The thermal conductivity of porous ceiling is set as air thermal conductivity. In addition, most CFD models do not include the radiation heat transfer, which is identified to be important (Chodor and Taradajko, 2013; Zhang et al., 2015b, 2016a&b) because the diffuse ceiling temperature is much lower than other wall surfaces. The equation for heat transfer in the porous zone is essentially a combination of eqs. (17) and (18). An effective thermal conductivity of the medium is introduced as the volume average of the gas conductivity and solid conductivity. Radiation has been taken into consideration by Zhang et al. (2015b; 2016a&b). Since the porous media assumption (Wu et al., 2013) is conflict with the use of radiation model, Zhang et al. (2015b, 2016a&b) constructed a second geometrical model, where the porous ceiling is assumed to be as 100% solid. This second model is used for radiation calculation based on surface-to-surface radiation calculation. Therefore, the equations to calculate the rising height of turbulence plumes have been identified to drive the room air circulation. Therefore, the equations to calculate the rising height and the volume flow rate of thermal plumes in stratified environments are summarized according to the type of heat sources, e.g., point or area sources.

6.3. Energy modelling

Energy modelling of ventilated room is based on the thermal dynamic process or heat balance:

\[
c_{p} \rho V \frac{dT_{\text{room}}}{dt} = \dot{Q}_{\text{int}} + \dot{Q}_{\text{solar}} + \dot{Q}_{\text{envelope}} + m_{\text{inf}} c_{p}(T_{\text{out}} - T_{\text{in}}) - \dot{Q}_{\text{HVAC}} = \dot{Q}_{\text{int}} + \dot{Q}_{\text{ext}} - \dot{Q}_{\text{HVAC}} \tag{45}
\]

where, \( \dot{Q}_{\text{int}} \) is the internal heat gain, \( \dot{Q}_{\text{solar}} \) is the solar heat gain, \( \dot{Q}_{\text{envelope}} \) is the heat transmission through building envelopes, \( m_{\text{inf}} \) is the infiltration mass flow rate and \( \dot{Q}_{\text{HVAC}} \) is the heat removed by HVAC. To simplify the equation, solar, envelope and infiltration heat gain are written together as \( \dot{Q}_{\text{ext}} \).

In a building with DCV systems, the energy modelling should consider that the ceiling panel divides the building space into a plenum and a room. Especially when the porous materials (Type 3 in Fig. 1) are used for ceiling panels, the heat balance for the plenum and the room should be separately calculated. If the porous ceiling has thermal mass, the heat balance for the porous ceiling should also been taken into consideration. The heat balance equations for these three zones might be written as:

\[
c_{p} \rho V_{\text{plenum}} \frac{dT_{\text{plenum.in}}}{dt} = \dot{Q}_{\text{plenum.ext}} - \dot{Q}_{\text{plenum.HVAC}} + \dot{Q}_{\text{vent.in.ceil}} - \dot{Q}_{\text{pleum.ceiling}} \tag{46}
\]

\[
c_{p} \rho V_{\text{room}} \frac{dT_{\text{room.in}}}{dt} = \dot{Q}_{\text{room.int}} + \dot{Q}_{\text{room.ext}} - \dot{Q}_{\text{vent.out.ceil}} - \dot{Q}_{\text{room.ceiling}} \tag{47}
\]

\[
\frac{dQ_{\text{ceiling.store}}}{dt} = \dot{Q}_{\text{vent.in.ceil}} - \dot{Q}_{\text{vent.out.ceil}} - \dot{Q}_{\text{pleum.ceiling}} - \dot{Q}_{\text{room.ceiling}} \tag{48}
\]

Where, \( Q_{\text{ceiling.store}} \) indicates the energy stored in the ceiling material. Eq.(46)–(48) can be combined as:

\[
c_{p} \rho V_{\text{plenum}} \frac{dT_{\text{plenum.in}}}{dt} + c_{p} \rho V_{\text{room}} \frac{dT_{\text{room.in}}}{dt} = \dot{Q}_{\text{ext}} + \dot{Q}_{\text{room.int}} - \dot{Q}_{\text{HVAC}} - \frac{dQ_{\text{ceiling.store}}}{dt} \tag{49}
\]

If the ceiling panels are type (1), (2) or (3) neglecting thermal capacitance and the heat capacitance of air can be neglected, Eq. (49) can be further reduced to:

\[
\dot{Q}_{\text{ext}} + \dot{Q}_{\text{room.int}} - \dot{Q}_{\text{HVAC}} = 0 \tag{50}
\]

The energy consumption of a building with DCV systems can be calculated by above heat balance equations. A major energy savings of DCV systems are due to the low pressure of the ceiling panel, which results in low fan energy consumptions. The fan energy consumption can be deduced from \( \dot{Q}_{\text{HVAC}} \). Energy modelling of a building with only DCV is not yet seen in literature according to authors’ knowledge and database search. Yu et al. (2015a) performed energy calculation of a test chamber with DCV, TABS and natural ventilation. The integrated system can reduce annual primary energy use from 63.5 to 5.5 kWh m\(^{-2}\) when the internal heat load is 65 W m\(^{-2}\).

7. Conclusions and suggestions

Diffuse ceiling ventilation (DCV) has received considerable attention in recent years due to its ability to handle high cooling loads without compromising thermal comfort. The system has energy saving potential due to the very low pressure drop through the diffuse ceiling. This review cites the papers on DCV published from 2008 to 2018 to highlight the research outcomes and to identify the research gaps. It also reviews the fundamental physics governing flow and heat transport mechanisms in a building designed with DCV. The fundamental theories include:

1 impingement jet over porous materials;
2 heat transfer in porous materials;
3 thermal plumes generated from heat sources in the room. Turbulent plumes have been identified to drive the room air circulation. Therefore, the equations to calculate the rising height and the volume flow rate of thermal plumes in stratified environments are summarized according to the type of heat sources, e.g., point or area sources.
4 turbulence fountains, which might explain the heating function of DCV. The heating efficiency is related to the steady rising height of turbulence fountains.

The review also discusses the popular research methods to study DCV systems: full-scale experiments and CFD modelling. Full-scale experiments are often used to evaluate the performance of DCV systems based on thermal comfort, indoor air quality and energy efficiency. On the other hand, CFD modelling is used for parametric analysis to improve the design of DCV systems. Finally, future research on DCV is suggested.

After DCV was introduced to residential buildings, researchers initially attempted to prove that DCV is a draught free system. Along this line, a large portion of the research focused on evaluating DCV systems via thermal comfort and the design chart proposed by Nielsen and Jakubowska (2009). Nielsen et al. (2010) investigated the driving force of air circulation in a room with DCV. This study promoted the research on the influence of heat load distributions on the performance of DCV. Since 2015, researchers have begun to combine DCV with other systems including radiant ceiling, natural ventilation and TABS. A few possible directions for future research
are discussed as follows.

The first future research direction is to further explore the potential to combine DCV with other HVAC systems. The ability of DCV to handle high cooling loads without causing cold draught might provide the opportunity to combine natural ventilation (Chen et al., 2017; Tong et al., 2017) with diffuse ceiling systems (Yu et al., 2015a&b, 2016). In naturally ventilated buildings, the ventilation rates are difficult to estimate (Wu et al., 2012b) and control due to fluctuations in outdoor climate conditions. The room could be over-ventilated and cold draught occurs to cause thermal discomfort during high external wind speeds. If wind flow is first introduced to the plenum and then diffused into room space through the porous ceiling, the cold draught can be eliminated. The fluctuation in natural ventilation rate might not result in thermal discomfort using this type of combined system. Natural ventilation is also very often used to cool buildings at night and store the cooling energy in the thermal mass (Lynch and Hunt, 2011). For office buildings, windows might be prohibited to open during night for safety reasons. A combination of natural ventilation and DCV can easily overcome such concerns. Yu et al. (2015a) proposed a novel system solution to combine natural ventilation with TABS into a DCV system. The novel system showed higher energy savings with the increase of internal heat loads. Besides combination with natural ventilation, Zhang et al. (2016b) investigated the possibility to combine radiant ceiling systems with DCV. These researches show that a combination of DCV with other HVAC schemes might provide innovative solutions for better thermal comfort and lower energy use.

The second research direction is to investigate the heating potential of DCV theoretically and experimentally. The DCV is now applied to three types of buildings: livestock buildings (Jacobsen, 2008), office buildings and classrooms (Table 1). However, DCV is designed for any space with high cooling loads without comprising on thermal comfort and IAQ requirement. Such spaces could be restaurants or wheelhouses on marine and offshore vessels. On ships, the AC system is usually run in both heating and cooling mode. In order to extend the application of DCV, the heating potential of DCV has to be understood at a more fundamental level. However, all the work listed in Table 1 paid little attention to the heating potential of DCV. This might be because Nordic countries do not use AC for heating purposes. Instead, a hot water system is used in winter conditions in Nordic region. In other climate regions or on marine ships, DCV might be designed for both cooling and heating. The simplified theoretical models reviewed in Section 5.4 cannot be applied in a general context due to the very low source Froude number of DCV. Therefore, research is needed to explore the heating potential of DCV. This might be done first by the establishment of simplified theoretical models for turbulent funnels from a very large source area. This kind of research can be done by injecting a dyed salt-solution into a tank filled with water (Campbell and Turner, 1989). To understand the role of thermal plumes generated by local heat sources (e.g. people) on the performance of DCV during heating, the interaction between a turbulent plume and a very weak floor should be investigated because theoretical models of this interaction will be crucial to understand DCV. Liu and Linden (2006) developed a theoretical model for the interaction between several forced funnels and a pure plume based on a two-layer stratification assumption. The theoretical model is aimed to predict the interface height of the two layers formed by an underfloor air distribution system. The work may be a good reference for investigating DCV systems used for heating.

The third research direction is to recast the solutions to eq. (31)-(33) for a DCV system in order to develop quick design tools. In Section 5, the simplified theoretical models are reviewed for each segregated space. These models are not in widespread use by building engineers and designers because theories on thermal plumes and funnels are not easy to comprehend. Therefore, the conventional methods are full-scale experiments and CFD modeling. However, full-scale experiments are generally impossible during the design phase. CFD consumes a lot of computational time yet still has uncertainties. Designers need quick tools to determine the dimensions of the plenum and room, the number of nozzles in plenum, air speeds and local ventilation rates in occupied zones. Therefore, it is necessary to integrate the simplified theoretical models and tailor them for DCV systems. An attempt to integrate simplified theoretical models has been done for transient buoyancy driven ventilation (Sandbach and Lane-Serff, 2011a & b). No such work has been done for DCV systems with the involvement of turbulent plumes and funnels.

The fourth research direction is to understand the behaviour of an impingement jet on a porous ceiling with porosity between 0.41 and 0.65. Simplified equations are needed to decide centreline velocity of such a jet. Although the procedure for using impingement jets in a plenum is recommended in Section 5, the criteria for parameters Y1, U1 and U2 are not known for such a confined space. The range for these parameters should be given to meet the thermal comfort requirement in the ventilated room by using full-scale experiments.

The last research direction is to decide the volume flux and rising height of thermal plumes from heat sources of particular shapes, for example, a manikin, a cube or a very large area in a stratified environment. It will be very useful to derive semi-analytical solutions like eqs. (34) and (35) for these particular heat sources in ventilated rooms.

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