

Displacement Ventilation and Passive Cooling Strategies

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ABSTRACT: *The publication of the ASHRAE “System Performance Evaluation and Design Guidelines for Displacement Ventilation” [1] has contributed to the wider acceptance of displacement ventilation (DV) as a ventilation strategy, by offering clear guidelines from an established organisation. A significant advantage of DV is that it lowers the supply air quantities required for cooling compared to conventional mixing ventilation (MV) at the same supply air temperature. This opens up opportunities for the use of passive (non refrigeration-cycle based) cooling sources, which typically are limited in supply air temperature and based on 100% fresh air supply when compared to conventional refrigeration-cycle based sources.*

This paper quantifies the impacts of using DV in comparison to MV on the peak capacity, size and humidity levels associated with the following passive cooling sources: evaporative cooling, two stage evaporative cooling, thermal stores and air to ground heat exchangers. A generic office building in Johannesburg, South Africa, is used as a model.

The paper illustrates the extent to which the use of DV expands the ability of passive cooling strategies to serve spaces previously considered as having too high a heat load (when calculated using MV system guidelines). The paper however also recognises that passive cooling strategies are unlikely to be widely implemented until design guidelines exist from organizations similar to ASHRAE.

Keywords: *passive design strategies, displacement ventilation*

INTRODUCTION

Based on the authors' experiences as consultants in the built environment, conventional Heating Ventilation and Air Conditioning (HVAC) engineers keep to using clear guidelines and calculations methods provided by established organisations. It is unlikely that unconventional strategies and systems will be incorporated in a project in the absence of such guidelines or methods, unless it is a showcase project with sufficient resources to adequately investigate alternatives.

Displacement ventilation (DV) can be seen as an example of this. While popular in Scandinavia [2], the American Society for Heating, Refrigeration and Air-conditioning Engineers (ASHRAE) only recently released “System Performance Evaluation and Design Guidelines for Displacement Ventilation” [1], which provides generic DV sizing and design guidelines. This publication differs from existing supplier-specific DV design guidelines in that it provides a simple ten-step guide to design the DV system [1]. While the authors have not found literature that documents the worldwide growth in the number of installed DV systems, since the ASHRAE publication was released they have found HVAC engineers in general to be more receptive to the use of DV for conventional projects.

The aim of this paper is to illustrate the impacts of using DV in comparison to mixing ventilation (MV) on the peak capacity, size and humidity levels associated with

various passive cooling strategies, informed by the ASHRAE DV guidelines. The intention is NOT to provide clear design guidelines for these passive strategies, but merely to demonstrate the implications of these strategies through quantitative analysis.

Through this analysis it is demonstrated that a number of passive cooling strategies, considered impractical for MV applications, can potentially be of benefit in building designs utilising DV. The comparison also highlights the risk of over-design when conventional MV guidelines are used to design cooling sources for DV applications.

DISPLACEMENT VS. MIXED VENTILATION

DV introduces air at low level and low velocity, and at high supply air temperature (T_s), typically around 18°C, when compared to conventional air conditioning systems which use MV. The air that is slightly cooler than the intended room temperature runs along the floor until it reaches a heat load (Fig. 1). The heat load induces a plume of warmer air that rises due to lower density. This induces stratification in room temperature with the occupied area of the room within comfort conditions and the space near the ceiling at higher temperature conditions. The air near the ceiling is continually exhausted to prevent a build up of warm air into the occupied zone.

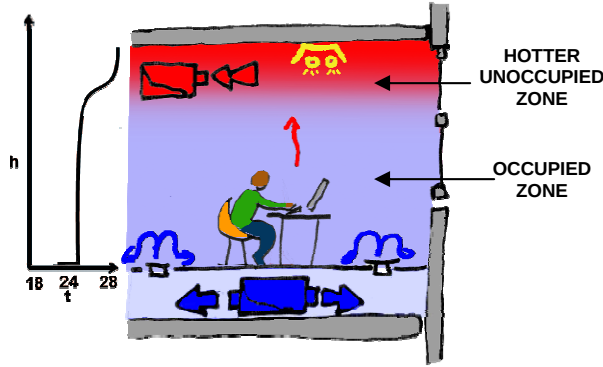


Figure 1: Displacement ventilation with indicative temperature gradient relative to height

When using MV the cool air - typically around 12 to 14°C - is introduced at higher velocities and at a high level, inducing room air to mix with it as shown in Fig. 2. There should be a relatively consistent temperature in the room.

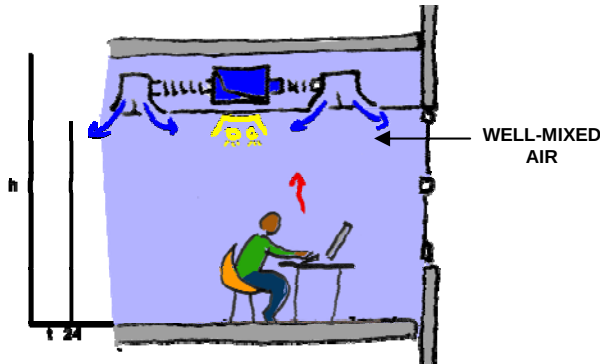


Figure 2: Mixing ventilation with indicative temperature difference relative to height shown as constant

Supply air flow rate calculations

A fundamental difference in the calculation of the supply air flow rate between DV and MV becomes apparent in the ASHRAE DV guidelines. Assuming no latent cooling, with MV the rate is calculated according to the entire room load with

$$V = \frac{Q_t}{\rho C_p (T_h - T_s)} \quad \dots (1)$$

where V is the supply air rate (m^3/s), Q_t is total room load (kW), ρ is the air density ($\text{kg}_{\text{dryair}}/\text{m}^3$), C_p is the specific heat capacity of air ($\text{kJ}/\text{kg}_{\text{dryair}}$), T_h is room design temperature ($^{\circ}\text{C}$) and T_s is supply air temperature ($^{\circ}\text{C}$).

However with DV the flow rate is based on the portion of the total room load that is present in the occupied zone using the following breakdown [1]:

$$Q_{\text{dis}} = a_{\text{oe}} Q_{\text{oe}} + a_l Q_l + a_{\text{ex}} Q_{\text{ex}} \quad \dots (2)$$

with Q_{dis} as the load between the head and the feet of a sedentary occupant being served by the DV system, Q_{oe}

as the loads from the occupants, low level lights and equipment (kW), Q_l as the load from the overhead lights (kW) and Q_{ex} as the envelope loads with a_{oe} , a_l , a_{ex} as the fractions of the respective loads that occur in the occupied zone (0.295, 0.132 and 0.185 in [1]). The resulting supply air flow rate is calculated as

$$V_h = \frac{Q_{\text{dis}}}{\Delta T_{\text{hf}} \rho C_p} \quad \dots (3)$$

with ΔT_{hf} as the temperature difference between head and feet and set at 2°C .

Exhaust air temperature

When using DV the exhaust air temperature (T_e) is in general higher than when using MV. In the case of MV the T_e should be close to the design room temperature, while the ASHRAE DV guidelines supply the following equations to calculate T_e for DV systems:

$$T_e = T_s + \frac{Q_t}{\rho C_p V} \quad \dots (4)$$

where

$$T_s = T_h - \Delta T_{\text{hf}} - \frac{\theta_f Q_t}{\rho C_p V} \quad \dots (5)$$

from

$$\theta_f = \frac{1}{\frac{V \rho C_p}{A} \left(\frac{1}{\alpha_r} + \frac{1}{\alpha_{\text{cf}}} \right) + 1} \quad \dots (6)$$

where T_f is the floor temperature ($^{\circ}\text{C}$), T_h is the temperature at the head or room design temperature ($^{\circ}\text{C}$), θ_f is the dimensionless temperature, α_r is the radiative heat transfer coefficient of the floor ($\text{W}/^{\circ}\text{C m}^2$) and α_{cf} the convective heat transfer coefficient of the floor ($\text{W}/^{\circ}\text{C m}^2$), both put forward as 5 by ASHRAE [1].

BACKGROUND TO MODEL

For the purpose of this paper a generic 100m² open plan office space in Johannesburg, South Africa, is used as a model onto which different ventilation and passive cooling strategies can be applied. The peak cooling loads within this office space are shown in Table 1.

Table 1: Peak cooling loads of the 100m² space (7W/m² for people, 13W/m² for equipment and 10 W/m² for lighting)

Occupants and equipment	Q_{oe}	2 kW
Overhead lighting	Q_l	1 kW
External loads	Q_{ex}	4 kW
Total loads	Q_t (for MV)	7 kW
Loads with occupied zone	Q_{dis} (for DV)	1.462kW

Three ventilation systems will now be considered: a DV system, as well as two MV systems, one that supplies air at the same high temperature as the DV system (MV_{ht})

and the other at a much lower supply temperature (MV_{lt}). Using Eq. 1 to 3, the required supply air flow rate into the room for each of these systems can now be calculated, as shown in Table 2.

Table 2: Resulting air flow from MV (low and high T_s) and DV systems with % relative to low (and high temperature mixing ventilation (MV_{lt} and MV_{ht}))

Mode	MV_{lt}	MV_{ht}	DV
T_s (°C)	14	22	22
Th/Te (°C)	26	26	26/31
V (m ³ /s)	0.59	1.61	0.74
% of MV_{lt}	100%	273%	125%
% of MV_{ht}	37%	100%	46%

From Table 2 it is clear that the air flow rate required for DV is less than half that required for MV_{ht} , even though both ventilation systems have the same supply temperature. The airflow rates for DV and MV_{lt} are similar, but MV_{lt} typically require a much larger cooling source than DV in order to obtain the low T_s .

The comparison in Table 2 highlights the efficiencies introduced by using DV instead of MV. The rest of this paper will explore how these efficiencies impact on a selection of passive cooling strategies. A number of passive design strategies are potentially applicable in the Johannesburg climate based on Fig. 3, which indicates a wide diurnal temperature range and low wet bulb temperature for much of the year. This city was therefore chosen as the context for the simulations presented in this paper. Typical office hours were assumed.

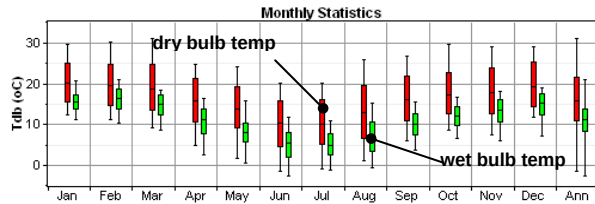


Figure 3: Dry and wet bulb temperatures for Johannesburg. [3,4]

IMPACT OF DISPLACEMENT VENTILATION ON PASSIVE COOLING STRATEGIES

The applicability of passive cooling strategies is a function of climate, comfort expectations related to the activity in the space, internal loading of the space and equipment/design implications – often relating back to cost. What this paper puts forward is that the method of introducing the cooling into the space also impacts on the suitability. The impact is quite specific to each passive cooling strategies, and will now be explored for the following strategies: evaporative cooling – single and two stage, thermal storage based on night time air temperatures and air to ground heat exchangers.

Evaporative cooling

Evaporative cooling uses the change of phase of the water contained in an air stream/volume to cool down the air. Energy from the air and non-evaporating water is used to supply the latent energy required. This results in a temperature drop in the air and residual water, and an increase in absolute and relative humidity of the air.

The minimum temperature achievable through evaporative cooling is expressed by the wet bulb temperature (T_{wb}) of the air. In reality the air can only be cooled a part of the way between the dry bulb (T_{db}) and wet bulb temperature and this is known as the evaporative cooling efficiency (η_{evap}), assumed as 80%.

$$T_s = T_{db} - \eta_{evap} (T_{db} - T_{wb}) \quad \dots(7)$$

Figure 4 provides the probability of T_s throughout the year based on eq. 7 and on annual climatic data [4].

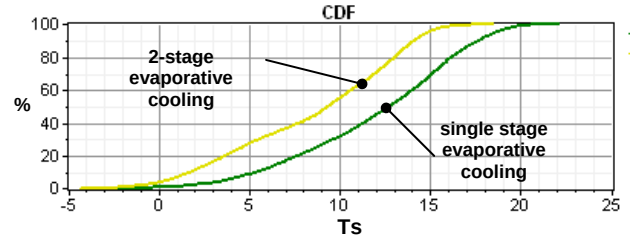


Figure 4: Cumulative distributed frequency of T_s for 2-stage evaporative cooling (1st line) and evaporative cooling (2nd line)[4,3]

Although evaporative cooling might be climatically suitable, the comfort expectations and room loading represent the actual limitations to this strategy. As shown in Fig. 5, evaporative cooling results in an increase in the absolute and relative humidity of the space, which impacts on comfort.

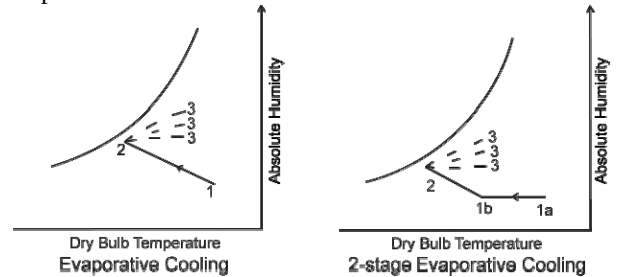


Figure 5: Psychrometric charts showing (left) the evaporative cooling process with 1 as the outdoor, 2 the supply and 3 the room air conditions with increasing absolute humidity during the day, and (right) the 2-stage evaporative cooling process indicating the sensible (1a to 1b) as well as adiabatic component (1b to 2)

The increase in humidity in the space over time is relative to the amount of vapour being added to the space. The amount of vapour entering a space is directly related to the supply air flow. Therefore the lower the air flow, the lower the absolute amount of vapour introduced into the space, slowing the build up of vapour of the

space during the day. Fig. 8 indicates a significantly lower inflow of humidity for DV compared to MV.

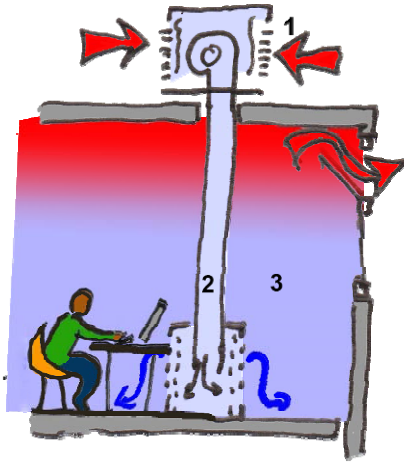


Figure 6: Fan driven evaporative cooling

This means that the comfort and cooling capacity (and therefore also cost) requirements of a specific project might be met through the use of evaporative cooling utilising DV, where it might have failed utilising MV. In other words, using evaporative cooling with DV rather than with MV means that a higher room load can be met (more cooling for the same humidity increase), or alternatively a lower increase in humidity can be realised for the same room load.

Two-stage evaporative cooling

Two-stage evaporative cooling adds a sensible pre-cooling component to the cooling process, which lowers the wet bulb temperature before applying direct evaporative cooling, as shown in Figs. 5 and 7:

$$T_{1bdb} = T_{1adb} - \eta_{sens} (T_{1adb} - T_{1awb}) \quad \dots(8)$$

from which

$$T_S = T_{1bdb} - \eta_{evap} (T_{1bdb} - T_{1bwb}) \quad \dots(9)$$

where η_{sens} is taken to be 90% and η_{evap} 80% [5]

This sensible pre-cooling component is typically provided by water that is cooled through evaporative cooling, using residual water from the evaporative part. Other strategies can also be used for sensible pre-cooling, e.g. passing air through an air-to-ground heat exchanger before applying an evaporative cooling component.

The result impacts in two ways; air is delivered at a lower temperature to the space and/or there is an increase in duration in the year that evaporative cooling can be used (Fig. 4) based on a maximum T_s of 22°C.

This influences the size of the room load that can be served by this strategy and the comfort conditions of the occupants as less humidity is being added to the space as indicated in Fig. 9.

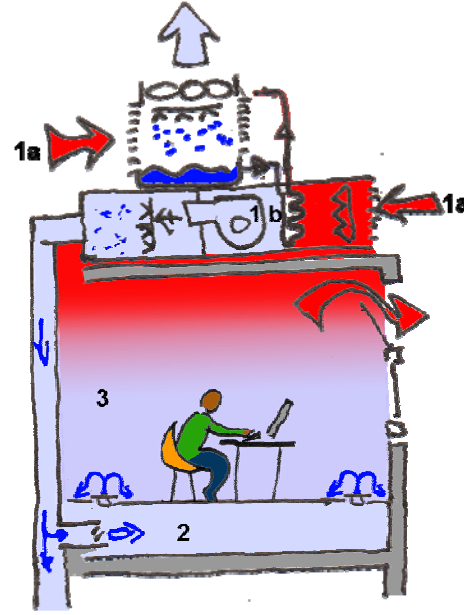


Figure 7: Diagram indicating 2-stage evaporative cooling equipment

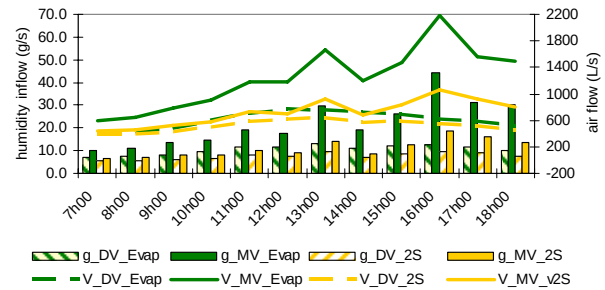


Figure 8: Air flow rates (V) and humidity inflow (g) over a day for single (Evap) and 2-stage (2S) evaporative cooling for a space with a peak total cooling load of 70W/m² indicated the difference between DV and MV.

Diurnal range and thermal store

Night time ventilation and exposed thermal mass is a relatively well known passive cooling strategy. It uses low night time air temperatures to flush the structure of the building of heat built up during the previous day. The same principle can be used in the introduction of a remote thermal store in the form of a packed bed, as shown in Fig 9. This bed is cooled down at night by flushing with outside air through mechanical ventilation. During the day outside air is then introduced through this bed, which cools the air before introduction into the space. The packed bed can be constructed by incorporating a variety of thermal storage materials, including ceramic balls and tiles, rocks and possibly phase change materials.

The climatic limitation of the packed bed thermal store strategy is the number of days that the night temperature drops to a usable temperature. Based on a T_s of 22°C,

and a minimum temperature difference of 3°C between the store material and the supply air, the duration that the strategy is applicable is indicated in Fig. 10. It is clear that the ambient temperature between 24h00 and 05h00 is lower than 19°C for 98.5% of the year.

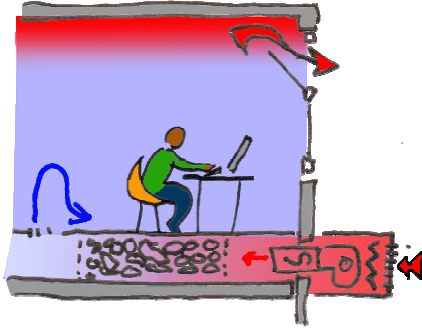


Figure 9: Packed bed thermal store

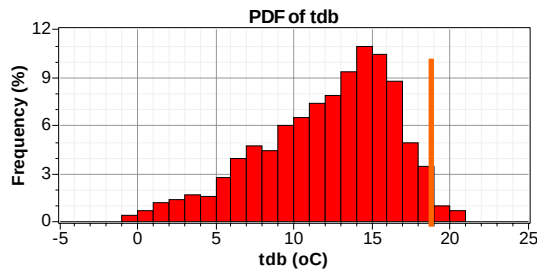


Figure 10: DVVIEW indicates duration that the night time cooling strategy is applicable in Johannesburg.

The feasibility of the strategy is further influenced by the cost and size of packed bed, which is a function of the storage capacity of the thermal material, the efficiency of the heat transfer between the storage medium and the air, and the air flow rate through the bed. Hollmuller et al. [6] notes that ideally a packed bed should cause a 180° phase shift in the daily temperature profile, as shown in Fig 11. For such a 180° phase shift, they suggest a packed bed volume of roughly 1m^3 for every $100\text{m}^3/\text{h}$ air flow rate. They further note that the amplitude change (transmission) of the phase-shifted output is related to the thermal storage material that is used: for a cost-effective material like gravel the transmission of the input amplitude is around 50%. Using the volume-airflow relationship, the size of a 180° phase shift packed bed for DV can be roughly estimated at 27m^3 , compared to a bed of 58m^3 for MV_{ht} .

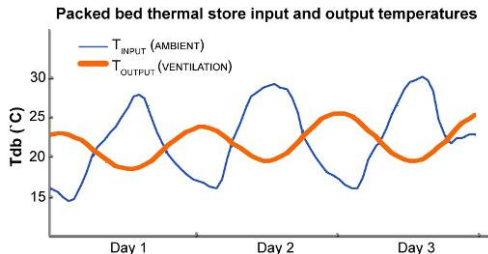


Figure 11: The input and output temperatures of a packed bed thermal store over time, illustrating the effect of a 180 degree shift in phase. Based on [6].

Using DV rather than MV therefore significantly reduces the required size of the packed bed thermal store, impacting on both material costs and more importantly the space that the store takes up. This last impact is, in the authors' experience, the common factor that makes the strategy unfeasible, due to the high cost of making the space available. An additional benefit derived from the lower air flow rate is the reduction in fan power and therefore energy consumption.

Air to ground heat exchangers

Air to ground heat exchangers consist of two distinctly different strategies. The first one, seasonal ground source cooling (Fig. 12), utilises the difference between the ambient temperature and the ground temperature at depths typically greater than 3m. Fig. 14 indicates how the amplitude of seasonal variation in the soil temperature decreases, and shifts in phase, as the depth increases. The second strategy, daily thermal storage (Fig. 13), uses the lower night time air temperature in a way similar to the thermal storage strategy described in the previous section, with the storage medium being the ground immediately around the piping. Compared to seasonal ground source cooling, the piping for daily thermal storage can be located nearer to the surface, and the piping runs are typically closer together.

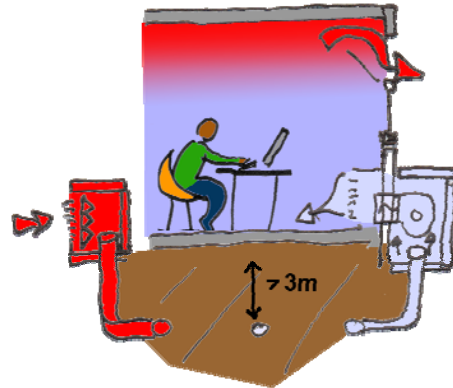


Figure 12: Seasonal ground source cooling

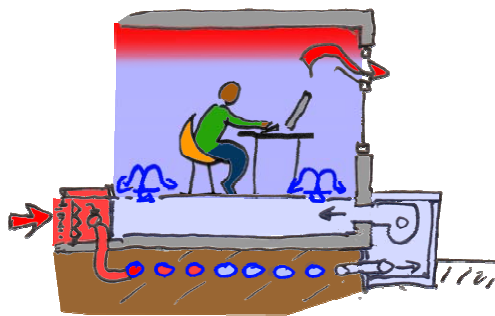


Figure 13: Daily ground source cooling.

The climatic applicability of the daily ground source cooling strategy is similar to that described in the previous section on thermal stores. The climatic applicability of the seasonal ground source cooling is

demonstrated in Figs. 14 and 15. The percentage of office hours that this strategy is applicable in Johannesburg depends on a variety of factors, including the depth, diameter and length of the exchanger, and the air flow rate. Fig. 15 illustrates how the percentage applicability of this strategy can be up to 100% at a T_s of 22°C, depending on the length of the exchanger, in this case shown for an air velocity of 5m/s and a depth of 3m.

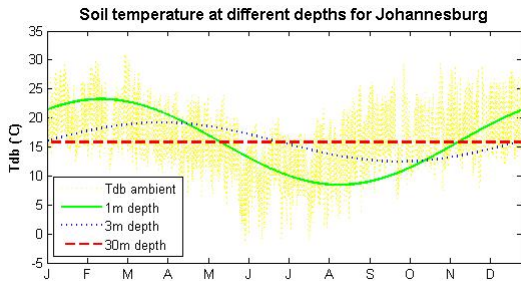


Figure 14: Annual air temperature and expected ground temperature at 3m, based on [7]

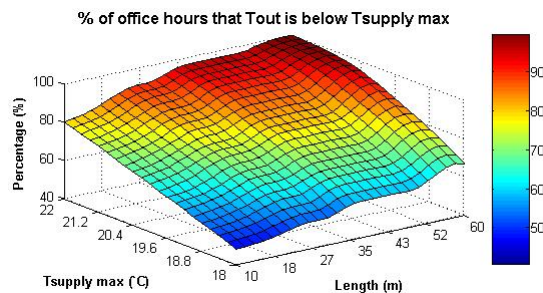


Figure 15: Percentage of office hours (6h to 18h) that T_{out} of the exchanger is below maximum T_s of the load, for air velocity of 5 m/s, pipe diameter of 0.38m, and depth of 3m. Based on [8]

Using the methodology described by Mihalakakou et al. [8] to estimate the length of exchanger required for the simulated Johannesburg office, with the same assumptions as in Fig. 15, results in the following: DV requires roughly 80m of pipes, compared to MV_{ht} requiring roughly 170m.

The above estimation illustrates that, as DV lowers the volume air flow rate required, it significantly reduces the pipe lengths required and therefore increases the technical and financial feasibility of the strategy. An additional benefit is a relative reduction in fan power and therefore energy consumption.

CONCLUSIONS

DV significantly lowers the required supply air flow to serve a room heat load, when compared to MV at the same T_s . This reduction impacts the feasibility of using passive cooling strategies in projects in a various ways:

Evaporative cooling – significantly less humidity is being introduced into the space due to the lower air flow.

2-stage evaporative cooling – intrinsically 2-stage evaporative cooling introduces less humidity than conventional evaporative cooling due to the sensible cooling component. This advantage is further amplified by DV in the lower supply air flow required compared to MV making the strategy applicable to more projects due to comfort levels.

Thermal storage – the storage volume is reduced, with the main impact being the reduction in the cost of providing the space for the store.

Air to ground heat exchangers – the pipe length is reduced dramatically.

The publication of the ASHRAE Guidelines has assisted in determining with confidence the flow rate required for DV: significantly lower than previously estimated using MV calculations. However, the sizing and calculations of the various passive cooling strategies (with exception of evaporative and 2 stage evaporative cooling) still depends on non-standardised methods that would typically not be considered by conventional HVAC engineers. Establishing guidelines for passive cooling, which are endorsed by organisations that HVAC engineers refer to in confidence e.g. ASHRAE, would assist greatly with gaining further penetration of these strategies beyond environmentally extreme projects.

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