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Design charts for sizing *CC*/*DV* system aided with personalized evaporative cooler to the desired thermal comfort

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ABSTRACT

This work extends chilled ceiling displacement ventilation (*CC*/*DV*) design charts to a wider operation range of *DV* supply temperatures and flow rates when aided with personalized evaporative cooler (*PEC*) at typical office of peak load of 70 W/m^2. The overall thermal comfort (*OTC*) of the occupant can be read off from the chart at different operating conditions of *DV*/*CC*-*PEC* system. The charts simplify the sizing procedure of the *CC*/*DV*-*PEC* system by specifying the operational design parameters which include temperatures of *DV* supply and *CC* surface, *PEC* airflow rate, the ratio of total load over *DV* airflow rate named displaced air parameter (*P*), and the *CC* cooling load. In producing the charts, extensive simulations were performed using a multilayer space thermal model to determine the human microenvironment conditions and local comfort models to determine occupant thermal comfort.

The use of the charts is demonstrated through case studies. It is found that the addition of *PEC* at flow rate of 10 l/s provided acceptable thermal comfort up to *DV* supply temperature of 26 °C. In such conditions, the *CC*/*DV* heat removal is about 83% of the space baseline load in design conditions which leads to energy savings while maintaining comfort.

1. Introduction

Most conventional mechanical cooling systems provide uniform thermal environment in the building. Combined chilled ceiling (*CC*) and displacement ventilation (*DV*) system is known to be an effective system [1,2]. Energy saving of 21 percent has been reported by Ghali et al. [3] compared to a conventional cooling system for the same air quality and thermal comfort.

The challenge in design of this integrated system is sizing of the two subsystems (*CC* & *DV*) to deliver the required load removal while providing a good level of indoor air quality and thermal comfort. This complex design process was initially addressed by Behne [2] and Tan et al. [1], who successfully implemented design diagrams for cooling loads below 100 W/m^2, indicating the applicability boundaries of *CC*/*DV* operative combination. For a room cooled with a combined *CC*/*DV* system, stratification height (the elevation at which the density gradients disappear in the rising air and its plume spreads horizontally) shows the level of the clean air zone and if the level is above 1.1 m, good indoor air quality is achieved. Nonetheless, this parameter was not clearly included in the works of Behne [2] and Tan et al. [1].

Ghaddar et al. [4] and Keblawi et al. [5] introduced new operational design charts that permit the user to select supply conditions, chilled ceiling temperature, the ratio of *CC* sensible load to the total sensible load of the combined system and also read off the value of the stratification height. The *CC*/*DV* design charts were developed for six load ranges from 40 to 100 W/m^2 at increments of 10 W/m^2. They considered range of values of supply air and chilled ceiling temperatures between 15 °C and 22 °C. The supply air temperature range is constrained to avoid the risk of cold drafts in the occupied region. The supply mass flow rate was constrained by the fresh air minimum requirement for acceptable *IAQ* with the condition that air velocity is less than 0.15 m/s to reduce draft for thermal comfort [11]. The design parameters, appearing in each of Keblawi et al. charts, are the supply temperature T_s, the chilled ceiling temperature T_c, load ratio *R* which is the fraction of the chilled ceiling ratio to the total sensible load, the displaced air parameter ($P = Q_s/\dot{m}_s$) which is the ratio of the total sensible load ratio to the supply air mass flow rate, the stratification height *H* and the vertical temperature gradient d*T*/d*Z*. The Keblawi et al. design chart for the cooling sensible load range 60–70 W/m^2 is shown in Fig. 1 since it will be used in this work as Ref. [5]. The shaded grey top region in the chart depicted in Fig. 1 represented conditions where the vertical temperature gradient d*T*/d*Z* is above comfort constraint of 2.5 K/m [5].

These charts (see sample chart shown in Fig. 1) were a step forward for determining the operating conditions of *CC*/*DV* systems

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Nomenclature

A	area
CFD	computational fluid dynamics
DV	displacement ventilation
$F_{j\to c}$	view factor from surface j to the chilled ceiling surface
h	enthalpy (kJ/kg)
h_c	convective heat transfer coefficient between ceiling and adjacent air layer (W/m^2 K)
h_{ci}	internal wall convection coefficient (W/m^2 K)
H	stratification height (m)
$\dot{H}$	enthalpy rate of air (kJ/(kg s))
$\dot{H}'_P$	enthalpy rate of the mixture of PEC and plume flow (kJ/(kg s))
IAQ	indoor air quality
J	number of wall surfaces in the space
$\dot{m}$	air flow rate (kg/s)
n	number of heat sources
OTC	overall thermal comfort
P	ratio of total sensible space load to total space supply air mass flow rate ($P = Q/\dot{m}_s$) (kJ/kg)
PEC	personalized evaporative cooler
PV	personalized ventilation
q_{PEC}	volumetric flow rate of the personalized evaporative cooler (l/s))
Q	cooling load (W)
R	ratio of sensible load removal by the chilled ceiling to total space sensible load
RH	relative humidity
T	temperature (°C or K)
$T_{C\text{-}PEC}$	corrected temperature of the evaporative cooler jet at the interface of the plume (°C)
w	humidity ratio (kg H_2O/kg dry air)
Z	vertical coordinate or height from the floor (m)

Greek

ε	emittance of the surface
σ	Stefan–Boltzmann constant (W/K4 m^2)

Subscripts

a	air
c	chilled ceiling
ac	air layer adjacent to ceiling and exhaust
e	exit
i	inlet
p	plume associated with heat source
s	supply air
w	wall

for sizing purposes. However, they do not quantify the level of thermal comfort and can only be used when *DV*'s supply temperature and *CC*'s temperature are both below 21 °C. Ghaddar et al. [4] reported the combination of *DV* supply temperatures below 21 °C and *CC* temperatures below 20 °C that result in comfortable condition. Moreover, the objective of these charts was to use them in sizing the *CC*/*DV* chiller capable of removing the load generated in the space.

Recent studies have proved that addition of a personalized evaporative cooler (*PEC*) to 100% fresh air ventilation system increases the level of thermal comfort, perceived air quality, health, and self-estimated productivity [6–8]. In a study about human response to local exposure, Zhang and Zhao [9] reported that thermal comfort can improve by face cooling; accordingly, the upper boundary of

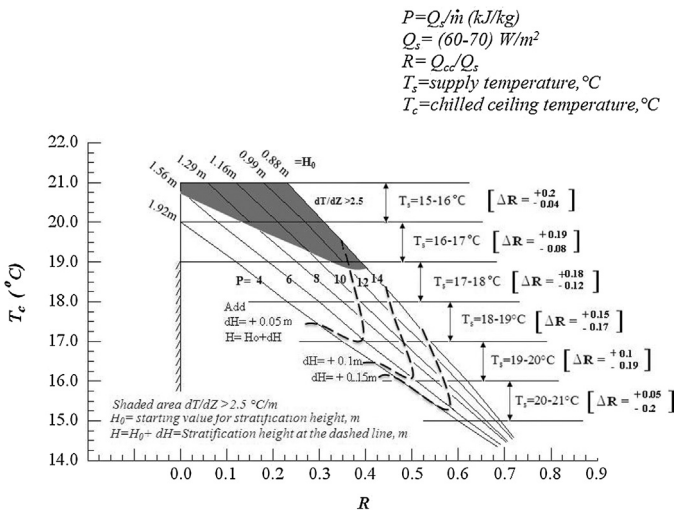

Fig. 1. Design chart of *CC*/*DV* system developed by Keblawi et al. [5] for sensible load range of 60–70 W/m^2.

acceptable room air temperature can shift from the conventional design value of 24 °C up to 30.5 °C. Using this idea, Chakroun et al. [10] studied that combination of *CC*/*DV-PEC* aiming the *PEC* jet toward the face. This combined system could shift the acceptable threshold of *DV* supply air temperature by 3 to 4 °C above 21 °C. Subsequently, findings of Makhoul et al. [11] and Chakroun et al. [10] show that energy savings of up to 20% can be achieved over systems that do not use personalized coolers while maintaining same comfort level.

The combination of *CC*/*DV-PEC* is shown to offer comfortable microenvironment and would result in less energy consumption since supply air flow rate can be set at higher temperature [10]. The integrated system has also been reported as a method to reduce the need for oversized air conditioning devices to cover transient load and reduce discomfort of people move from extreme outdoor climate to conditioned indoor environment [12]. This can also be helpful when actual load exceed building design load due to many factors that have to do with construction, system inefficiencies, and people use of space, particularly in office buildings [13,14]. Nevertheless, *CC*/*DV-PEC* requires a complex design procedure and if not properly performed, occupants may then experience local and overall thermal discomfort. In addition, excessively high *PEC* airflow rates and low *PEC* airflow temperatures compared to the surrounding environment can cause draught and thermal discomfort, respectively. Achieving acceptable design of *CC*/*DV-PEC* system is much easier if early design guidance is offered; consequently, in this work we are proposing to generate extended operational design charts for the *CC*/*DV-PEC* system for heavy load range of office building where such system would be most applicable for stationary occupants [14,15].

In general, integration of *PEC* with 100% fresh air convective cooling systems and more specifically with *CC*/*DV* system has been recently studied [14–23]. However, the following issues have not been tackled:

1. Providing design charts to simplify the complexity of downsizing process of *CC*/*DV* aided with *PEC* for typical cooling load of a conventional office (70 W/m^2)
2. The impact of *PEC* flow rate on comfort, when *PEC* is combined with *CC*/*DV* and is operated at room air temperatures above 26 °C and *DV* supply temperatures above 21 °C

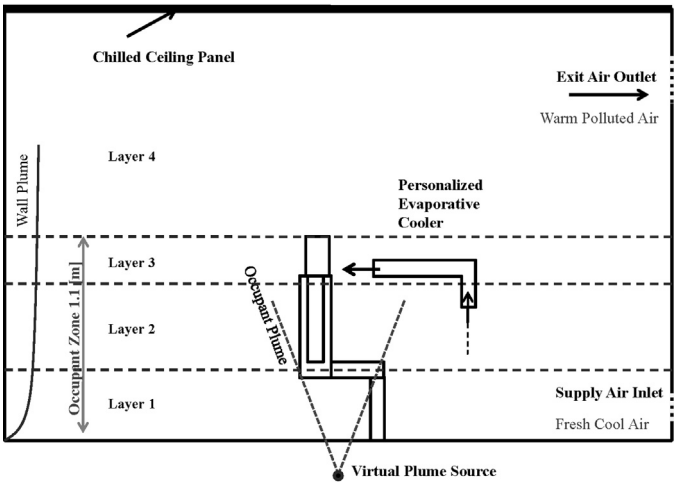

Fig. 2. Schematic of a seated human model in a room equipped with *CC*/*DV-PEC* system [10].

3. Relating the level of thermal comfort to the relevant designed operative conditions of *CC*/*DV-PEC* and the associated energy consumption.

In this work, we aim: (1) to develop a new design chart that can be used for operation of the *CC*/*DV* system at supply temperatures above 21 °C which is a range that has not been covered in previous charts [5]; and (2) to quantify the level of overall thermal comfort (*OTC*) with and without a *PEC* for a downsized *CC*/*DV* system while still providing comfort for the same baseline load. The impact of *PEC* on comfort level will be illustrated against cooling energy consumption and *CC*/*DV* operating variables for an office with heavy baseline load of 70 W/m^2 [14,15,18]. Moreover, in order to illustrate the use of design charts and associated energy saving advantages, a guideline and sample case studies are provided for use in two climates (moderate humid climate and hot dry climate).

2. Methodology

This section discusses the methodology used in developing the *CC*/*DV-PEC* design chart. Fig. 2 shows a schematic diagram of a room conditioned by a chilled ceiling displacement ventilation system where a person is seated in the space while a *PEC* jet is directed toward the person upper body segment. The foremost necessity of design chart's development is a robust model with relatively low computational-cost in order to generate the prerequisite numerous data points. Therefore, the well-tested comprehensive model for *CC*/*DV* conditioned-space developed by Ayoub et al. [16], which was used to generate previous charts [4,5], was selected for this work as well. In addition, the integrated *CC*/*DV-PEC* conditioned space-model of Chakroun et al. [10], which is a modified version of Ayoub et al. [16], was used to estimate the modified vertical temperature gradient in the space when *PEC* is operated. The thermal conditions adjacent to the various human segments obtained from the *CC*/*DV-PEC* model were then incorporated into a Bioheat and segmental thermal comfort models to predict associated human overall thermal comfort and sensation. For that reason, prior to explaining the process of data generation and development of design charts, the physics of the problem and the simulation models are explained.

2.1. The CC/DV-PEC conditioned space model and comfort assessment model

2.1.1. Base model of CC/DV Chart

The multi-layer space thermal model of Ayoub et al. [16] is used in this work for *CC*/*DV* system sizing when the PEC is off. The model predicts conditions of the supply *DV* air and chilled ceiling temperature that are able to remove the load and satisfy comfort and air quality constrains. It is the same model used in development of previously validated and published *CC*/*DV* design charts [4,5]. This model estimates the plume and the flow rates of walls and other heat sources, the vertical temperature distribution of walls and air, and the split of cooling load removal by each of the *CC* and *DV* subsystems based on their operative parameters. The split of load removal is basically the sensible load ratio *R* (see design chart of Fig. 1 [5]) given by

$$R = \frac{Q_{cc}}{Q_s} = \frac{Q_{cc}}{Q_{cc} + Q_{DV,s}} \tag{1}$$

where Q_{cc} and $Q_{DV,s}$ are the sensible loads removed by the chilled ceiling and *DV* systems, respectively. The sizing of the *CC* and *DV* subsystems is complex and involves multiple operational variables: the supply conditions (flow rate, temperature and humidity) of the *DV* system and the chilled ceiling temperature [5]. The *DV* sensible load removal is calculated from the flow rate and the difference between supply and return temperatures as follows:

$$Q_{DV,s} = \dot{m}_s c_p (T_{ac} - T_s) \tag{2}$$

where T_{ac} is the temperature of the exit air equal to ceiling adjacent air layer temperature. The chilled ceiling load removal Q_{cc} is basically convective and radiative whereas it can be calculated as the summation of the ceiling convective load and radiative loads exchanged with each room wall surface layer as follows [17]:

$$Q_{cc} = h_c (T_{ac} - T_c) + \sum_{j=1}^{J} \varepsilon \sigma F_{jc} (T_j^4 - T_c^4) \tag{3}$$

where h_c is the convective heat transfer coefficient between the ceiling and the adjacent air layer at temperature T_{ac}, *J* is the number of wall surfaces in the space with distinct view factors and temperatures, ε is the surface emittance, and σ is the Stefan–Boltzmann constant. The chiller capacity of the combined *CC* and *DV* subsystems can be derived from suitable values selected for the sensible load ratio *R* (chilled ceiling load ratio to total sensible load removed by *CC*/*DV*) and flow parameter *P* (total sensible load ratio over *DV* supply flow rate) [22].

The wall-plume-multilayer model assumes that the air movement is unidirectional and vertical. Thus, the space is divided into *N* horizontal layers each layer is characterized by three lumped air temperatures: the air temperature inside the wall plume, the air temperature inside the hot source thermal plume, and the room air temperature (see Fig. 1). The flow rates resulting from wall plumes and plumes of heat sources are denoted by $\dot{m}_{w,k}$ and $\dot{m}_{p,k}$, respectively. The net circulated mass $\dot{m}_{w,k}$ at each boundary (interface surface between two adjacent air layers *k* and *k*+1) is calculated from each layer mass balance. The net circulated flow rate $\dot{m}_{cir}$ at each height *Z* representing the layer *k* height is calculated as

$$\dot{m}_{cir} = \dot{m}_s - \sum_{i=1}^{n} \dot{m}_{p,i,k} - \sum_{j=1}^{J} \dot{m}_{w,j,k} \tag{4}$$

where *n* is the number of heat sources and *J* is the number of walls.

For given space geometry, wall material, internal heat sources, external solar and environment conditions for walls, and the estimation of plume flow rates emanating from the heat sources Mundt

[23] and the non-uniform temperature walls, the energy balance in each layer can be represented as:

$$q_k + \sum_i \dot{m}_i h_i - \sum_e \dot{m}_e h_e = 0 \quad (5)$$

where $q_k = \underbrace{\Phi_k}_{\text{heat source}} + \underbrace{\sum h_{ci} A (T_w - T_a)_k}_{\text{convective heat transfer}}$ and h_{ci} is the internal wall convection coefficient. The $\dot{m}_i$ and h_i terms represent the flow rates and enthalpies of the air flows entering the layer k (including the wall plumes, heat source plumes and the circulated mass). Similarly, $\dot{m}_e$ and h_e represent the flow rates and enthalpies exiting the air layer. The parameter q represents the external and internal heat sources acting on the control volume. The term Φ represents the internal heat sources such as the human body segments present at a certain layer and generating a heat flux. The total room load $Q = \sum q_k$ can be calculated by adding load of all heat sources and wall load from all layers.

The complete plume-multilayer wall model formulation can be found in the work of Ayoub et al. [17] and Chakroun et al. [10].

2.1.2. Combined model for the CC/DV-PEC chart

Chakroun et al. [10] combined the aforementioned model of Ayoub et al. [16] with their own developed correlation of *PEC* outlet conditions in addition to outlet air stream diffusion correlation of Wang Shan [18] and segmental bio-heat and thermal comfort model of Ghali et al. [19] in order to assess the impact of *CC*/*DV-PEC* on human thermal comfort. The input parameters to their model were the outdoor ambient conditions for external walls and internal walls temperatures, the *CC* temperature, the *DV* system supply flow rate temperature, and the *PEC* nozzle exit flow temperature and flow rate. The model predicted the plume and room air temperature distributions in addition to wall temperatures and wall plumes. The integrated model *CC*/*DV-PEC* of Chakroun et al. [10] combined the space model for the plume prediction and steady state experimental *PEC* correlation for predicting the temperature drop of the air that results from humidification. The model of Chakroun et al. [10] has low computational costs and at the same time its predictions of vertical temperature distribution and comfort showed good agreements with experimental results; therefore, it is used in this work.

As was shown in Fig. 2, the room was represented by N horizontal air layers inside the plume and outside the plume for the room air within each layer. The personalized evaporative cooler jet is included in the layer containing the trunk and head of the human body and is considered to affect the heat source plume temperature and flow rate [11]. Since the *PEC* jet flow rate is small compared to the *DV* supply flow rate, it is assumed that changes in vertical temperature gradient are negligible compared to the vertical gradient without the *PEC* outside plume. However, the plume vertical flow rate and temperature were affected and the *PEC* jet entrainment was hence considered in the mass and energy balances of the plume layer where the jet exists. The *PEC* air flow velocity was assumed to be at low turbulence level entrained in the thermal plume rising profile. The cool air from the PEC module will mix with the existing plume reducing its temperature. The new plume flow rates and temperatures are calculated by adding the flow rates and performing an energy balance on the plumes control volume, respectively, using the methodology of Makhoul et al. [11] where the energy balance equation for the air layer including the *PEC* module is given by

$$\dot{H}'_p = \dot{H}_p + \dot{H}_{PEC} \quad (6)$$

where $\dot{H}_{PEC} = \dot{m}_{PEC} C_p T_{C-PEC}$, C_p is the specific heat of air, $\dot{m}_{PEC}$ is the evaporative cooler mass flow rate, $T_{C\text{-}PEC}$ is the corrected temperature of the evaporative cooler jet at the interface of the plume, and $\dot{H}'_p$ the enthalpy rate of the mixture. The corrected *PEC* temperature when reaching the human plume after leaving the *PEC* nozzle T_{PEC} is found using Wang [18] correction chart for the jet average velocity and temperature drop as a function of distance x from the nozzle outlet. Using Makhoul et al. model, the new average plume temperature associated with the layer including the *PEC* module can then be calculated by updating plume energy and mass balances [11]. The solution of the mass and energy balances provided information on the vertical temperature gradient of the room air and air flow rates passing from one layer to the next. The plume upward flow and temperature were adjusted to account for the entrainment of the *PEC* jet [11]. The human body is considered to be in the seated position and it is represented by a vertical cylinder of diameter $D = 0.47$ m and height $H = 1.1$ m so that the total cylinder area would be equal to the average human body surface area of 1.8 m^2. The *PEC* flow and exit temperature conditions needed for the model are dependent on the design of the *PEC* used and the fan power. In this work we have utilized a water cooling-pad type of *PEC* for which information was available on the relation between inlet air conditions (temperature and humidity) and exit air conditions as a function of flow rate [10] which is given by

$$T_{PEC} = -0.024 \quad q_{PEC} + 1.28 \quad T_{in} + 0.127 \quad RH_{in} - 12.563 \quad (7)$$

where T_{PEC} is the outlet temperature of the *PEC* jet in [°C], q_{PEC} is the volumetric flow rate in [l/s], T_{in} is the inlet temperature in [°C] and RH_{in} is the relative humidity at the inlet of the *PEC*. At low flow rate, the drop in air temperature is about 2 °C to 3 °C depending on inlet temperature and humidity ratio of withdrawn air. The *PEC* is assumed to draw air from the lower fresh air layer.

2.1.3. Comfort assessment model

The environmental conditions adjacent to the various clothed human segments obtained from the *CC*/*DV-PEC* model are incorporated into a transient bioheat model [20]. The bioheat model predicted the segmental skin temperature and heat loss from ventilated clothed body segments. The segmental skin temperatures and their rate of change were used to evaluate local and global thermal comfort and thermal sensation using the models of Zhang [9]. The sensation scale is from −4 (very Cold) to +4 (very hot) and the comfort scale ranges from very uncomfortable (−4) to very comfortable (+4) [9]. The bio-heat model [20] considers 17 segments for the human body. In our study with the aim of simplification and in order to avoid having too many vertical air layers (one for each segment) these 17 segments are clustered to 3 groups. Head is studied alone as it has the highest impact on the level of *OTC* in presence of a *PEC* aiming toward the face. Chest, back, abdominal, buttocks, arms, and hands are grouped as the upper-body segment and thighs, calves, and feet are named as the lower-body segment. In this work, the thickness of air layers in the *CC*/*DV-PEC* space thermal model was adjusted in order to accommodate having each one of the three body-segment groups in one of the three lowest layers. Lumped air temperature in each layer was calculated, and consequently used to calculate the air temperature leaving the *PEC* and the air temperature reaching the face in order to calculate the skin temperatures and level of *OTC* for different operative conditions via the bio-heat model [20]. The sequence of calculations performed in this model is presented as a flowchart in Fig. 3.

2.2. Development of the design charts

2.2.1. Design parameters and base load selection

The ultimate goal of any cooling system is to provide a good level of thermal comfort for the occupant in the context of their task and their environment condition while consuming as less energy as possible. For this reason and to be consistent with previous charts

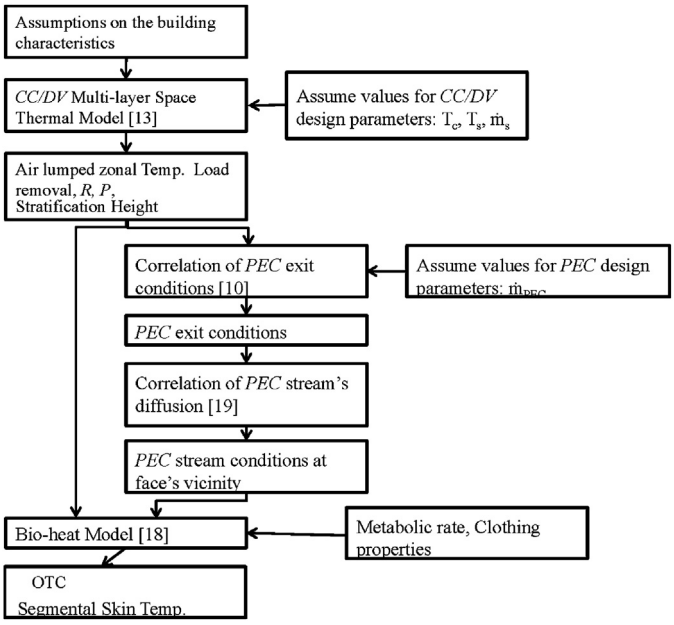

Fig. 3. Flowchart calculations of *CC*/*DV*-*PEC* multi-layer space thermal model.

[4,5], this work associates the thermal comfort level to four different design parameters, including:

- *CC* temperature (T_c in °C) and the *CC* cooling load (Q_{cc} in W/m^2).
- *DV* supply temperature (T_s in °C).
- The ratio of the total sensible load in kW to the *DV* supply mass flow rate ($\dot{m}_s$) in kg/s defined as displaced air parameter *P* in kJ/kg (parameter *P* is independent of room size and condition).
- *PEC* volumetric airflow rate $\dot{m}_{PEC}$ in l/s.

The above parameters are summarized in Table 1 including their feasible operational ranges. To perform simulations, a base load for a *CC*/*DV* conditioned space need to be identified within the load range of the *CC*/*DV* system operation. A peak baseline sensible load of 70 W/m^2 was chosen for this study for an office space with floor area of 5 × 5 m^2 and ceiling height of 3 m and maximum occupancy of 4 persons [15] although existing office buildings actual occupant density is reported around 10 persons/100 m^2 [13]. The baseline load, occupancy load, internal load, and all the other assumptions are based on heavy load density office [15,18]. Since the current study aims at using *PEC* to reduce the load, it dictates that the occupant to be located close to the *PEC* and to be kind of stationary which is the case in office buildings. The sensible baseline load was calculated for design room air temperature of 24 °C. The latent load is associated only with occupants. The selection of a baseline load of 70 W/m^2 is appropriate since it is on the heavy load side of an office space [15] and if the intention is to reduce system size for example by 20%, it means the load removal would reach around 55 W/m^2. For loads that fall below 70 W/m^2 with low occupancy, the saving in system size using *PEC* may not be significant to justify the use of the *PEC*. Moreover, building codes are pushing to limit the energy consumption in office buildings by enforcing an upper load limit of 65 W/m^2 [15]. Existing offices that are characterized by heavy load and are therefore exceeding this upper limit can make use of *PEC* to bring their consumption down [13,14].

Table 1
CC/*DV*-*PEC* design parameters and their operational ranges.

Design parameter	Operational range
Chilled ceiling temperature (T_c in °C)	18 to 24
Chilled ceiling sensible cooling load (Q_{cc} in W/m^2)	0 to 60
DV supply temperature (T_s in °C)	18 to 28
DV cooling load (Q_{DV} in W/m^2)	0 to 40
Displaced air parameter *P* in kJ/kg [the ratio of the total sensible load in kW to the *DV* supply mass flow rate ($\dot{m}_s$) in kg/s]	2 to 14
PEC volumetric airflow rate $\dot{m}_{PEC}$ in l/s	0, 5, 10
Overall thermal comfort, *OTC*	Very uncomfortable (−4) to very comfortable (+4)

Internal loads include heat generated from occupants, lights, and miscellaneous equipment. The lighting and other miscellaneous equipment loads (plug loads) are simplified to produce a fixed load of 36 W/m^2 and the external load through the walls is about 18 W/m^2. It is also assumed that there is no direct solar gain. The wall conductance is set to 1.5 W/K m^2 [23], which conducts the external heat load through the walls. Two walls are external wall and exposed to a fixed outside temperature of 34 °C; while internal walls are partition adjacent to temperature of 25 °C. The office's features were assumed to be the same as the ones used in development of previous *CC*/*DV* charts [4,5] and models [10,17].

It should be noted that since fresh air is directly reaching the human's breathing zone by penetrating through the free convection flow around human body and before mixing with neighboring polluted air, a good level of inhaled air quality can be assumed to be delivered by locating the intake of the *PEC* at the *DV* fresh air supply close to floor level [7,8]. Thus, air quality is not among the determinant design parameters anymore and thermal comfort of the occupant and energy consumption prevails as the design priorities.

Simulations were performed for constant values of $\dot{m}_s$ in the range of 80 to 420 l/s, starting with constant chilled ceiling temperature at T_c = 18 °C and incrementing T_s by 0.5 °C from 20 to 27 °C to find the corresponding total sensible load and chilled ceiling sensible load ratio *R*, stratification height, vertical temperature distribution of room air, and vertical room air temperature gradient. The same process would be followed in increments of 0.5 °C in chilled ceiling temperature T_c up to 24 °C. Afterward, the resulting level of thermal comfort would be evaluated for various *PEC* flow rates.

2.2.2. Feasible CC/DV-PEC design chart constraints and operative ranges

Since *DV* supplies the cool air at low level and stratifies the room air temperature and contaminants as a result of the natural buoyancy forces, stratification height should be above the breathing zone of a seated occupant (1.1 m). However, in this work due to presence of *PEC*, occupant is already protected from airborne transmission; therefore, the acceptable minimum threshold of stratification height is lowered from 1.1 to 0.8 m and can go up to values as high as 1.7 m. This means that if stratification height is less than 0.8, fresh air will not reach the occupant breathing zone. While, if the flow rate is such that the stratification height is above the height of a standing person (1.7 m), then we have a case that is not energy efficient and the *DV* system advantage of creating two zones would be compromised. For these reasons all data points obtained in the simulation were examined, and data points in which stratification heights are below 0.8 and above 1.7 m are eliminated as acceptable operational conditions. This restriction eliminates all the operational conditions with high supply flow rates ($\dot{m}_s > 430$ l/s) represented by data points with *P* values below 3.0.

Comfort is measured on scale from very uncomfortable (−4) to very comfortable (+4) [9]. A minimum *OTC* value is set at −0.5 to assure that the occupant would feel neutral. The *DV* supply flow rate has strong influence on stratification height; thus, by restricting

the stratification height, the range of supply flow rate would be determined. Another limitation on the maximum $\dot{m}_s$ is dictated by *ASHRAE* standard [18] to avoid thermal draft. *ASHRAE* recommends air movement velocity of 0.15 m/s in the occupied zone and limits it to a maximum value of 0.25 m/s. The minimum recommendation of *ASHRAE* is set to remove the generated latent load in the room due to presence of people and operative *PEC*s.

The upper threshold of relative humidity of supply air is set to avoid condensation on *CC* while removing the generated latent load from the room. Latent load is then controlled by maximum supply humidity ratio ($w_{s,\max}$) of the *DV* system. The following correlation is used to calculate the maximum humidity ratio of supply air:

$$w_{s,\max} = w_{a,\max}\Big|_{T_{dp,\min}} - \frac{N_p(w_p + w_{PEC})}{\dot{m}_s} \quad (8)$$

The maximum humidity ratio of the supply air, $w_{s,\max}$, depends on number of occupants (N_p), supply flow rate $\dot{m}_s$, humidity generated by people (w_p), water evaporation in *PEC* (w_{PEC}), and the maximum allowable humidity ratio of the air adjacent to *CC* to avoid condensation, $w_{a,\max}$. The maximum humidity ratio in the room upper layer is assumed to be at the dew point temperature 1.5 °C less than T_c, representing a constraint for avoiding condensation on the chilled ceiling [5]. The room humidity is determined by the internal moisture generation and the amount of moisture added by the *PEC*. It is needed to determine the amount energy spent by the dehumidification system. Since the exit *PEC* jet temperature is computable by the correlation developed by Chakroun et al. [10], the amount of water evaporation and the consequent w_{PEC} is easily calculated.

The considered range for supply air temperature as mentioned earlier was 20 °C to 27 °C results in room temperature range between 26 to 32 °C [9]. Limiting minimum value of *OTC* to −0.5 would define the maximum supply temperature at 26.5 °C and room air temperature at 29.5 °C. The previous chart of Keblawi et al. [5] showed that for supply temperatures below 20.0 °C, the occupant would be comfortable without *PEC* (see Fig. 1) and the addition of *PEC* for these low temperatures is not needed. Generally, when downsizing is not performed, the 100% fresh air *DV*/*CC* system is perfectly capable of providing a good level of thermal comfort. When the room temperature goes below 23.0 °C, the addition of *PEC* would diminish the thermal comfort. Moreover, the minimum threshold of T_s is set with the aim of energy saving on the chiller that is cooling the outside air to the supply temperature.

For the *PEC* airflow rate, as it has to penetrate through human thermal plume, a minimum air velocity of 0.3 m/s has been recommended at the face vicinity by Melikov [6] and this value can go as high as 1.5 m/s especially at room air temperatures above 26 °C. The relevant level of thermal comfort was calculated for three different *PEC* volumetric-flow rates of 0, 5, and 10 l/s. These flow rates result in three *PEC* outlet air stream speed values of 0, 2, and 4 m/s, respectively, while the terminal velocity reaching the face after entrainment is less 0.5 m/s [11].

The chilled ceiling temperature is in the range of 18 °C and 24 °C, which is the most prevalent range for the radiant *CC* operating in a room with high temperature and humidity [5]. As the T_c increases, the contribution of *CC* to the overall cooling load decreases which means a broader range for *DV* and *PEC* operative parameters; until, the maximum T_c equals 24 °C is achieved. The chilled ceiling load removal Q_{cc} varies from 11 to 43 W/m^2. Tan et al. [1] and Behne [2] suggested that *CC* portion in total sensible load removal should be below 67%. Parameter *R* is the ratio of chilled ceiling load to total cooling load and it is calculated from the charts by dividing the read Q_{cc} over the baseline load of 70 W/m^2. The parameter *R* varies from 0.22 to 0.54, which is well below prescribed maximum values in the literature.

A *CC*/*DV* system is down sized when aided with *PEC* and can operate at different design conditions that would result in different comfort levels. Out of 1800 different combinations of operative parameters that comply with stratification height thresholds of 1.1 m, only about 40 of them are capable of providing comfort in the absence of *PEC* at the operating supply condition of temperature higher than 21 °C. For all these operating conditions, combination of design parameters are set in a way that the cooling systems remove at least 97% of the baseline load.

When the *PEC* is aimed toward the face providing cooler local exposure, it results in better segmental comfort for the face and accordingly a better *OTC* sensation. In order to determine the most efficient operational settings in the presence of *PEC* system, careful interdependent study of the operative parameters were performed. Addition of *PEC* improves *OTC* and expands significantly comfortable conditions.

All the operative points that can satisfy the stratification height threshold with downsized *CC*/*DV* cooling system for baseline load of 70 W/m^2 are presented in the new design chart shown in Fig. 3. The introduced design charts show the *DV* supply air temperature as a function of the *CC*/*DV* load removal from the baseline at different constant *P*-line values from 4 to 14. The dotted lines show the limiting threshold for comfort region at *PEC* flow rates of 0, 5, and 10 l/s. In addition, the dotted thick lines represent constant Q_{CC} constant load removal with the associated range of T_C values. In addition to the main design chart that highlights the thermal comfort regions based on different *PEC* flow rates, three other charts are produced for a more precise and more comprehensive study of the comfortable regions for each *PEC* flow rate. The individual charts generated for selected *PEC* flow rates of 0, 5, and 10 l/s will enable the evaluation of the level of the overall thermal comfort (*OTC*) at all operating conditions. Fig. 5 shows the design chart for the case when no *PEC* is used while imposing the lines of constant *OTC* on the chart and showing the operative small region where comfort is attained. Similarly, Figs. 6 and 7 show the *OTC* level on design charts for *PEC* flow rate of 5 and 10 l/s, respectively. It is clear that the thermal comfort region at high *DV* supply flow rate expands as the *PEC* flow rate is increased. The next section explains how the charts can be used.

2.3. Use of CC/DV-PEC design charts

Multiple points at each load removal results in different comfort levels depending on its operative conditions including the *PEC* flow rate. Results show that amount of space load removal of *CC*/*DV* has the highest impact on level of *OTC* while *DV* supply temperature is the most determinant factor on the amount of load removal from the room for a limited range of chilled ceiling temperature. Therefore, the two parameters of *DV* supply temperature and *CC*/*DV* load are represented on vertical and horizontal axes of all developed design charts (Figs. 5 to 8), respectively. The chilled ceiling load removal, Q_{cc}, and the flow rate parameter *P* are plotted in the charts as lines with constant slope. The Q_{cc}-lines intersect with lower values of *P*-lines at high T_s, which mean that higher supply mass flow rates are required to compensate for high supply temperatures to maintain the required comfort (see Figs. 4–7). The maximum stratification height that dictated a maximum supply flow rate, does not allow *P* to go below 4.0 kJ/kg, this is shown as the dashed line with constant inclination named as “Threshold of Maximum Stratification Height”.

The other boundary of comfortable points for each *PEC* flow rate, illustrated in the charts, is the minimum set value of *OTC* (−0.5). This shows that *OTC* restriction eliminated all the points that were resulting in stratification heights below the minimum stratification height threshold. The −0.5 *OTC* threshold is shown as dashed red lines for three *PEC* airflow rates of 0, 5 and

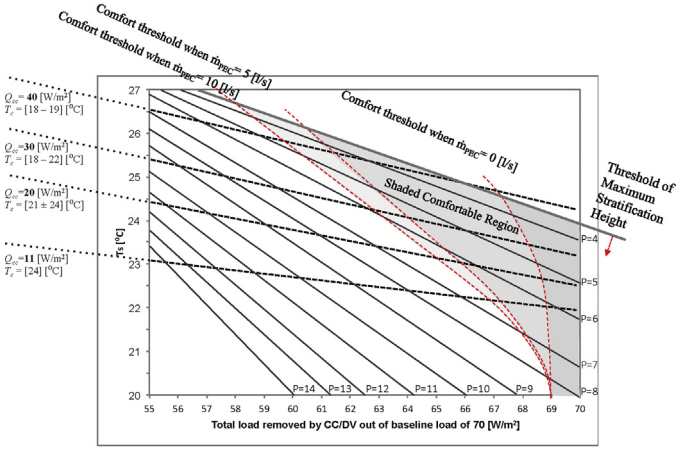

Fig. 4. Design chart of *CC*/*DV*-*PEC* system for the baseline load of 70 W/m^2.

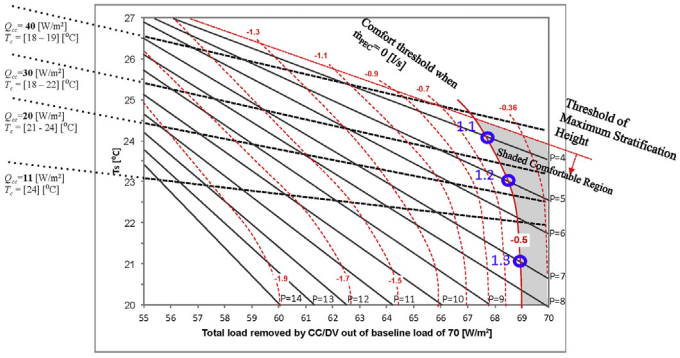

Fig. 5. Design chart of *CC*/*DV*-*PEC* system for PEC flow rate of 0 l/s.

10 l/s. All points on the right side of each linear line meet the required minimum stratification height and the required minimum *OTC* of −0.5 with a combination of operative *CC*/*DV*-*PEC* conditions.

The range of comfort (*OTC*) and the range of possible downsizing of *CC*/*DV* as a percentage of baseline load of 70 W/m^2, for different *PEC* flow rates, are as follows:

- *OTC*: −0.5 to −0.35 and 97%, when $\dot{m}_{PEC}$ is 0 l/s;
- *OTC*: −0.5 to +0.7 and 87%, when $\dot{m}_{PEC}$ is 5 l/s;
- *OTC*: −0.5 to +0.9 and 83%, when $\dot{m}_{PEC}$ is 10 l/s.

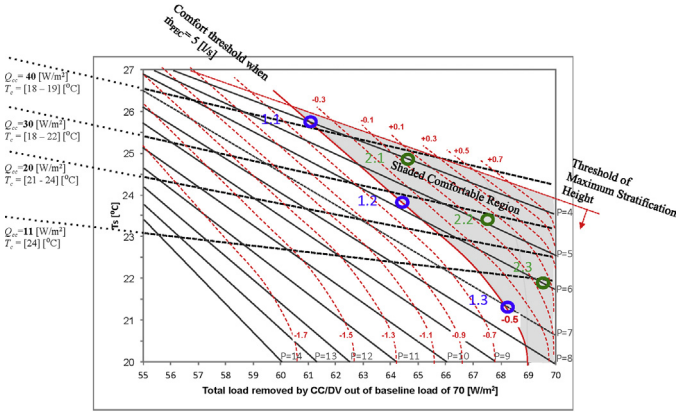

Fig. 6. Design chart of *CC*/*DV*-*PEC* system for PEC flow rate of 5 l/s.

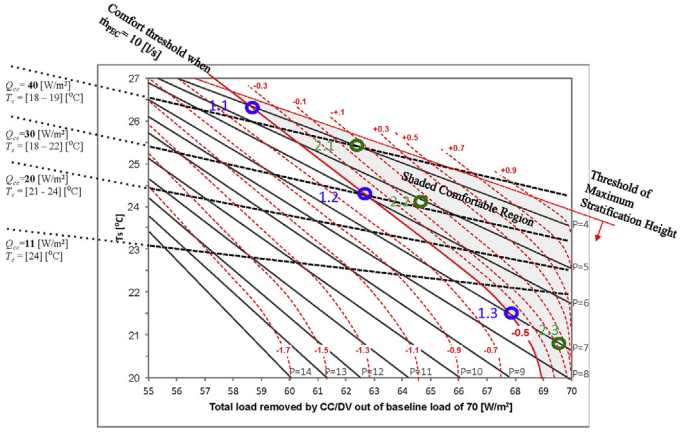

Fig. 7. Design chart of *CC*/*DV*-*PEC* system for PEC flow rate of 10 l/s.

The charts enable selecting operating conditions at higher *DV* supply which result in reducing capacity of *CC*/*DV* system when it is aided with PEC, Figs. 5–7 become then important as they show the thermal comfort level of each operative point and the amount of load removal at each *PEC* flow rate. This will enable user to size the combined *CC*/*DV*-*PEC* system. The *CC*/*DV*-*PEC* chart development depended on published experimentally validated robust models to generate the operating data points and load parameters by the authors [3–5,10,11,17,20,22].

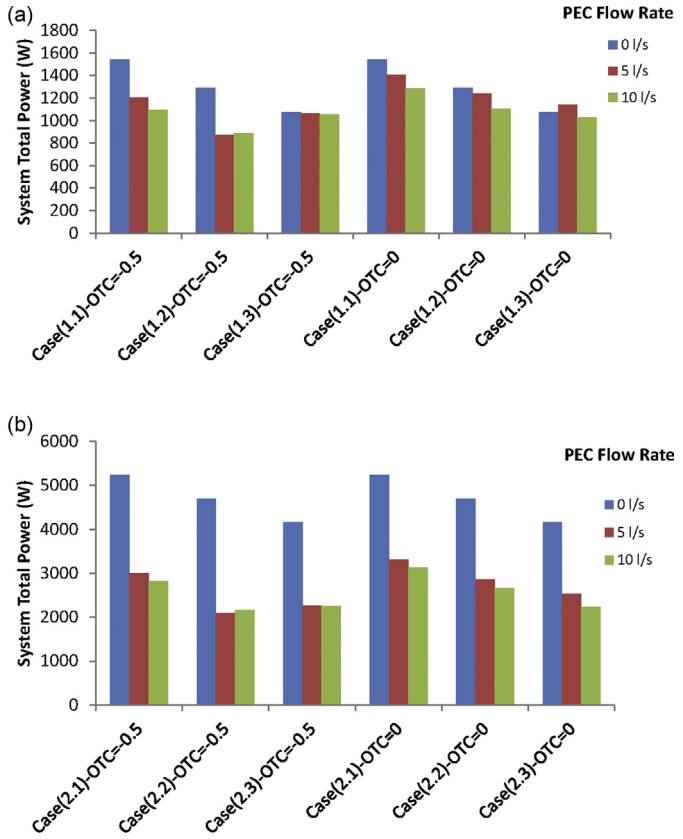

Fig. 8. The *CC*/*DV* system power for the different selected cases at *PEC* flow rates of 0, 5, and 10 l/s and *OTC* levels of −0.5 and zero for (a) Beirut climate, and (b) Kuwait climate.

3. Results and discussion

Upon examining the effect of the various *PEC* flow rates the for baseline load of 70 W/m^2 and for various cooling capacities, we notice that that the cooling load of 58 W/m^2, is the lowest load removal point for the integrated *CC*/*DV*-*PEC* system at *PEC* flow rate of 10 l/s that can provide acceptable *OTC* at −0.5 for an occupant with moderate office activity. This cooling capacity is equivalent to 83% of the space baseline load at design conditions, which results in thermal load reduction of 17.1% from the baseline peak cooling load. At this load, the applicable chiller of *DV* should deliver supply temperature value of 26.5 °C. The 17.1% is actually a reduction in sizing the space cooling load system.

A minimum supply flow rate is dictated on *DV* in order to meet stratification height constraint to uphold the required level of fresh air in the occupied region. Under these conditions, as the supply flow rate attains its lowest value, it should be assisted by low supply temperatures and this is how the minimum of T_s and T_c are achieved for each load removal. The increase of *PEC* flow rate improves the level of thermal comfort; and consequently increases the number of feasible operative points which means that the comfort can only increase with increase in *PEC* flow rate. This is because a portion of the base load (70 W/m^2) is used in air conditioning the space resulting in room air temperature higher than the comfort design temperature (24 °C); therefore, to increase the *OTC* level *PEC* flow rate should be increased and maximum of *PEC* flow rate is determined based on acceptable air velocities at the vicinity of human's face

A close look at the two design charts for *PEC* flow rates of 5 and 10 l/s (Figs. 5 and 6), clarifies that addition of *PEC* with these two conventional flow rates can bring the occupant *OTC* level to be close to the neutral level for load removals around 60 W/m^2 (85% of baseline load). If the load removal is increased to 65 W/m^2 (95% of baseline load), the occupant can experience a slightly pleasant sensation when using the *PEC*. The overall thermal comfort level *OTC* improves much faster in higher *DV* flow rates where it has wider range between constant *OTC* lines at the top of the chart (low values of *P*-line) than bottom of the chart.

Load removals above 98% of baseline load provide the design room air temperature of 24 °C. At this temperature the *OTC* level is already neutral and it can improve by the addition of *PEC* causing the thermal comfort threshold to be vertical at the bottom of the chart. Addition of *PEC* in this load range allowed operation at higher supply temperatures while aided by lower T_c values and higher $\dot{m}_s$ resulting in higher *CC* load removals and lower *P* values. The charts demonstrate that, in presence of *PEC*, *DV*'s operational conditions have high impact on *OTC* at constant *CC* temperature. In addition, high *DV* supply flow rate is needed ($P \leq 8$ kJ/kg) to attain thermal comfort when *DV* supply temperature is above 20 °C. Clearly the operational parameters *P*, T_c, and T_s are influential factors in determining the *OTC* level and sizing the two subsystems. The *DV*'s supply temperature is a core factor in sizing the chillers while *P* is the core factor for sizing supply and exhaust fans and consequently for determining the energy consumption of the integrated system.

To conclude, if the *CC*/*DV* system does not remove the total space load, then it is not capable of providing comfort. The addition of *PEC* with volumetric flow rates up to 10 l/s proved to be a reliable option for the occupant to achieve comfort. In high room air temperatures (26 °C and above), the higher the *PEC* flow rate, the lower the temperature of the air stream reaching the face would be. In general, the higher the temperature difference between the *PEC* stream and the surrounding temperature, the higher the impact of *PEC* on the local and overall comfort would be.

Table 2
Load data of the case studies.

	Kuwait City	Beirut
Room floor area	4 × 4 m^2	4 × 4 m^2
Room height	3 m	3 m
Walls and floor overall heat transfer coefficient	1.5 W/m^2 K	1.5 W/m^2 K
Peak load conditions		
External walls (front & left) are exposed to temperature and relative humidity conditions	48 °C and 22%	34 °C and 65%
Internal walls (back & right) and floor are exposed to temperature condition	28 °C	28 °C
Light bulbs (internal load)	1 × 40 W	3 × 40 W
Personal computer (internal load)	1 × 150 W	3 × 150 W
Occupants (internal load)	2 × 58.2 × 1.8 W	3 × 58.2 × 1.8 W

4. Use of design chart in system sizing for two case studies

A design procedure is presented to illustrate the use of energy-comfort charts when seeking a certain level of *OTC* for an office with 70 W/m^2 peak internal load equipped with a *CC*/*DV*-*PEC* system. The charts depict the resulting *OTC* and provide the ability to estimate the energy consumed to cool the space at different operating conditions of *CC*/*DV*-*PEC*. Hence, determining an operating point with the least requirement for load removal and best comfort is rather straightforward. Nevertheless, when several points result in the same level of *OTC* (which is usually the case), a more comprehensive study should be performed in order to choose the most economical operating condition. Designer should not only consider the space load removal value read from the charts; but also, consider the associated load of chiller, fans, and reheat when applicable.

Two practical examples are presented to clarify the procedure for determining the possible combination of operative parameters and to estimate the associated energy consumption. In the first and the second case an *OTC* level of −0.5 and 0.0 is respectively sought. These cases cover two different climates in a typical summer day, one for cities with high level of humidity such as Beirut climate, and one for cities with very low level of humidity during summer such as Kuwait city. It should be noted that for the same baseline load, the process of using the charts is not climate dependent. However, the amount of energy consumed to bring the outside air condition to that of supply, makes all the difference and determines the total amount of energy saving.

Table 2 details the conditions for the case studies that are selected to be similar to the cases studied by Keblawi et al. [5]. The office space is equipped with a *CC*/*DV* system and one *PEC* for each occupant. The walls and the floor have the same construction which would result in an overall heat transfer coefficient of 1.5 W/m^2 K [23]. It is also presumed that there are no windows or doors and the solar gain is zero. The level of thermal comfort in the space would be calculated for the case of a seated person with metabolic rate of 58.2 W/m^2 of which 70% is sensible heat generation and 30% latent heat loss [15]. The heat sources inside the room include a certain number of occupants as well as light bulbs and personal computers located on the desks that each one would dissipate 40 W and 150 W, respectively (see Table 2).

Here are the proposed steps to use the design charts seeking a certain level of *OTC*:

Step 1: Calculate the baseline load generated in the office space with any certified load calculation software package. The design temperature is usually considered 23 °C without *PEC*. Generally, the addition of *PEC* facilitates an increase of 3 to 5 °C to this value [10]. An overall load of 68.77 W/m^2 and 69.88 W/m^2 was calculated for Beirut case and Kuwait City case, respectively, which are in the

Table 3
Sizing results of different case studies at *OTC* level of −0.5 and zero.

Case 1: sought *OTC* = −0.5												
Case no.	$Q_{internal}/A_{Room}$ [W/m^2]	T_s [°C]	P [kJ/kg]	$w_{s,max}$ [kg/kg]	Beirut				Kuwait city			
					$Q_{internal}$ [W]	$Q_{Chiller}$ [W]	Q_{Fan} [W]	Q_{Total} [W]	$Q_{internal}$ [W]	$Q_{Chiller}$ [W]	Q_{Fan} [W]	Q_{Total} [W]
Reference case: *PEC* airflow rate = 0												
1.1	68	24.5	4	0.0111	1088	1091.8	452.8	1544.6	1088	3089.4	452.8	5242.2
1.2	68.5	23	5	0.0109	1096	1051.9	237	1288.9	1096	2678.6	237	4691.4
1.3	69	21	7	0.0106	1104	985.1	88.3	1073.3	1104	2192	88.3	4162.4
***PEC* airflow rate = 5 [l/s]**												
1.1	61	26	4	0.011	976	876.4	326.9	1203.2	976	2674	326.9	3000.8
1.2	64.5	24	6.5	0.0106	1032	782.5	90	872.6	1032	1997.5	90	2087.6
1.3	68.5	21.5	7	0.0106	1096	976.7	86.4	1063.1	1096	2176.1	86.4	2262.4
***PEC* airflow rate = 10 [l/s]**												
1.1	58.5	26.5	4	0.011	936	805.2	288.3	1093.5	936	2534.4	288.3	2822.7
1.2	63	24.5	6	0.0107	1008	778.1	106.7	884.8	1008	2055.9	106.7	2162.6
1.3	68	21.5	7	0.0106	1088	968.4	84.5	1052.9	1088	2160.2	84.5	2244.7
Case 2: sought *OTC* = 0.0												
Case no.	$Q_{internal}/A_{Room}$ [W/m^2]	T_s [°C]	P [kJ/kg]	$w_{s,max}$ [kg/kg]	Beirut				Kuwait city			
					$Q_{internal}$ [W]	$Q_{Chiller}$ [W]	Q_{Fan} [W]	Q_{Total} [W]	$Q_{internal}$ [W]	$Q_{Chiller}$ [W]	Q_{Fan} [W]	Q_{Total} [W]
***PEC* airflow rate = 5 [l/s]**												
2.1	65	25	4	0.011	1040	1012.4	395.5	1407.9	1040	2919.8	395.5	3315.3
2.2	68	23.5	5	0.0109	1088	1009.7	231.8	1241.5	1088	2626.2	231.8	2858.0
2.3	69.5	22	6	0.0108	1112	998.2	143.2	1141.4	1112	2395	143.2	2538.2
***PEC* airflow rate = 10 [l/s]**												
2.1	62.5	25.5	4	0.011	1000	935.7	351.6	1287.3	1000	2773.6	351.6	3125.2
2.2	65	24.5	5	0.0109	1040	902.4	202.5	1104.8	1040	2453.9	202.5	2656.4
2.3	68.8	21	7	0.0106	1100.8	940.5	87.5	1028	1100	2145.1	87.5	2232.6

operational range of the *CC*/*DV-PEC* design chart. It is assumed that *DV*'s system is supplying 100% fresh air to the room.

Step 2: Refer to

1. Fig. 3 if not seeking a certain level of *OTC* (only general comfort matters) From this chart, according to flow rate of the used *PEC*, one can choose an operative condition from the relevant comfort region.
2. Fig. 4 if no *PEC* is used and the sought *OTC* is in the range of [−0.5 to −0.36].
3. Fig. 5 if the PEC flow rate is 5 l/s and the sought *OTC* in the range of [−0.5 to +0.7].
4. Fig. 6 if the PEC flow rate is 10 l/s and the sought *OTC* is in the range of [−0.5 to +0.9].

Two cases are demonstrated directly on Figs. 4–6. In case 1, an *OTC* level of −0.5 and in case 2, and *OTC* level of 0.0 is sought. Here, only the first case is discussed in details and demonstrated. The second case follows the same procedure. As shown in Fig. 4, any point on the line of −0.5 *OTC* can be selected for all the *PEC* flow rates and various operative conditions. Three different points are picked from each chart (Figs. 4–6) and are shown as blue circles for selected cases 1.1, 1.2 and 1.3 and as green circles for selected cases of 2.1, 2.2, and 2.3.

Step 3: Calculate the total energy consumption by following the routine below and adding them up:

5. Read the amount of load removed by *CC*/*DV* out of the baseline load ($Q_{internal}/A_{room}$) from the charts and multiply by the office floor area to measure the internal load ($Q_{internal}$).
6. Calculate the load on chillers. For dry climates such a Kuwait City, chillers would only perform sensible cooling. In humid climates such as Beirut, humidity of the air should be removed in a very economical method. Here it is assumed that firstly a solar desiccant dehumidifier wheel [21] is used (negligible energy consumption) and then sensible cooling is performed.
7. Calculate the load on Fans [5].

Table 3 shows the results on energy consumption for the case studies assuming a value of 3.5 for the chiller coefficient of performance and system duct pressure drop of 0.6 kPa to calculate fan power consumption. For case study 1 presented here, the most economical choice for each case depends on supply conditions and climate. In general, the energy used to remove the internal space load significantly decreases by addition of *PEC* and the higher the supply mass flow rate, the higher the energy consumed by the fans would be**.** The percent reduction in system size is significant in dry hot climate of Kuwait compared to similar system in humid warm climate. Fig. 7 shows the *CC*/*DV* system power for the different selected cases at *PEC* flow rates of 0, 5, and 10 l/s and *OTC* levels of −0.5 and zero for (a) Beirut climate, and (b) Kuwait climate. It is clear that in the moderate climate of Beirut, the % reduction in size can reach 32% when *PEC* is used with high supply temperature compared to the reference case. In Kuwait dry hot climate, the % reduction in system size when *PEC* is used at flow rates of 5 and 10 l/s can reach 40% to 54% at high supply flow rates.

5. Conclusions and recommendations

Operational design charts for combined *CC*/*DV-PEC* system have been developed for an office with baseline load of 70 W/m^{2}in order to be used for prediction of thermal comfort based on the design parameters. Since the *PEC* is capable of lowering the air temperature by few degrees, a higher *DV* supply temperature can be tolerated to provide the desired comfort level which leads to energy savings; thus, the *OTC* level is the most important parameter when considering energy consumption and thermal sensation of the occupant.

Results show that by increase in *PEC* flow rate, level of thermal comfort increases. For cooling load removals below 58 W/m^2, comfort is unachievable even with the highest PEC flow rate (10 l/s). In high load removals the 100% fresh air *CC*/*DV* system is perfectly capable of providing a very good level of thermal comfort and as the room temperature goes below 22.5 °C, comfort would start decreasing. As in these cases we are dealing with high supply mass flow rates, meaning low *P* values, and low temperatures, addition of *PEC* would not be in favor; specially, for temperatures below 22 °C it would cause discomfort.

Studying the data points for low load removals shows that *DV* supply flow rate is the most determinant operative condition which determines the minimum values for *CC* temperature and supply temperature. At these low load removals, the room air temperature is very high and results in very low levels of *OTC*.

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