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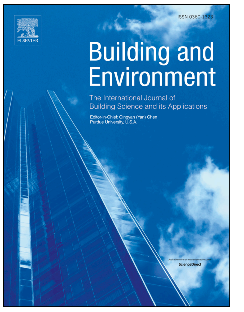

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Sustainable active cooling strategies in hot and humid climates – A review and a practical application in Somalia

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Abstract

Providing spaces with thermal comfort is a critical need, particularly in hot-humid regions, where more than a third of the world's population are settled. In terms of space cooling technologies, the global market is largely dominated by window air conditioners and ductless mini-splits. However, numerous other options are available and require more exploration to clearly identify their advantages against conventional systems. In this regard, there are plenty of studies about active cooling systems, but still little is known about the best choices for their application in high-density buildings in extreme humid climates, where latent loads are dominant. In such a framework, this paper briefly examines and identifies possible solutions for space cooling in non-residential buildings in hot-humid climates. In this sense, an all-air system with heat recovery and a radiant ceiling coupled with air handling were identified as the most recommended options for such environments, based on their working principles. Furthermore, the study includes a detailed assessment of the application of these solutions on a case-study building in Mogadishu, Somalia, from the point of view of the cooling demand to the energy consumption of the selected cooling systems. The findings of this work can be extrapolated to be then applied in other developing cities, and outline the path future research should follow to improve the systems.

Keywords: Cooling; Air conditioning; Indoor air quality; Thermal comfort; Hot-humid climates

Nomenclature

A_z	Zone floor area (m^2)
c_p	Specific heat of air (J/kg-°C)
E_{fan}	Energy consumption for the AHU fans (W)
E_{pump}	Energy consumption for the pumps (W)
h_{fg}	Enthalpy of vaporization of water (J/kg)
$\dot{m}_{people}$	Vapor mass per person (kg/s per person)
N_{people}	Number of people in the space
P_z	Maximum occupancy of the zone during typical usage
$q_{ceiling}$	Cooling capacity of the radiant ceiling panel (W/m^2)
Q_{cc}	Total cooling coil load (W)
Q_{cl}	Latent cooling coil load (W)
Q_{cs}	Sensible cooling coil load (W)
Q_l	Latent loads of the space (W)
Q_s	Sensible loads of the space (W)
R_a	Outdoor airflow rate required per unit area (m^3/s per m^2)
R_p	Outdoor airflow rate required per person (m^3/s per person)
T_{ae}	Temperature of air entering the cooling coil (°C)
T_{cc}	Temperature of air leaving the cooling coil (°C)
$T_{ceiling}$	Temperature of the radiant ceiling panel (°C)
T_{in}	Temperature of the indoor air (°C)
$\dot{V}_{outdoor}$	Outdoor airflow rate (m^3/s)
$\dot{V}_{supply}$	Supply airflow rate (m^3/s)
$\dot{V}_{water}$	Water volumetric flow rate (m^3/s)
w_{ae}	Specific humidity of air entering the cooling coil (kg/kg)
w_{cc}	Specific humidity of air leaving the cooling coil (kg/kg)

Greek letters

Δp	Pressure drop (Pa)
ε_s	Sensible effectiveness of the HRW
η_{fan}	Fan efficiency
η_{pump}	Pump efficiency
ρ	Air density (kg/m^3)

Acronyms

AC	Air conditioner
ACB	Active chilled beams
ACC	Air-cooled chiller
AHU	Air-handling unit
CDD	Cooling degree days
COP	Coefficient of performance
DAHU	Desiccant-based air-handling unit
DOAS	Dedicated outdoor air system
DPT	Dew point temperature
DX	Direct expansion
EER	Energy efficiency ratio
FCU	Fan-coil unit
GHG	Greenhouse gas
HRW	Heat recovery wheel
IAQ	Indoor air quality
PMV	Predicted mean vote
SHF	Sensible heat factor
SSLC	Separate sensible and latent cooling
VAV	Variable air volume
VCRS	Vapor-compression refrigeration system
VRF	Variable refrigerant flow

1. Introduction

The incessant expansion of the building and construction sector is a concern worldwide, the construction and operation of buildings accounting for 36% of the global final energy consumption and almost 40% of the total greenhouse gas (GHG) emissions [1]. As the world's population grows rapidly – and is expected to increase by 26% within 2050, from the current 7.7 billion people [2] - more energy is required to meet people's needs. According to the United Nations, Sub-Saharan Africa is the region that will concentrate the biggest population growth in the coming decades, followed by northern Africa and Western Asia [2], being these regions also where most of the least developing countries are located [3].

More in detail, developing countries are increasing their demand for cooling due to climate change, population growth, higher access to energy, and other socioeconomic factors [4]. High temperatures and humidity directly affect cooling energy consumption, which is the main issue in hot-humid regions, considering that these are inhabited by more than 33% of the global population [5]. Since 1990, energy demand for space cooling has experienced a very fast growing, becoming three times higher today [6]. Window air conditioners and ductless mini-splits, characterized by their high electricity requirements, are the common technologies used to provide space cooling. Their also present an environmental challenge, strongly related to the consumption of resources and their entire impact during their life cycle [4]. In terms of energy consumption, it was estimated that, between 1990 and 2016, about 20% of the total electricity used in buildings around the world was due to air conditioning systems and fans [7].

However, there is still a large gap in the access to space cooling to achieve the thermal comfort requirements, especially in developing countries. South Asia region is the one with the highest energy cooling gap, followed by Africa; while in Asia this gap is directly related to the lack of air conditioners, in Africa it is also due to the lack of access to electricity [8,9]. Projections indicate that, in the case of Africa, energy consumption for space cooling will increase by more than 13% per year until 2040 [10], not only due to population growth, but also because of the climate change [11]. Thus, the increase in energy consumption in the coming years requires more efforts aimed at optimizing energy use, particularly for space cooling.

In this context, the present article proposes a brief literature review about possible active cooling strategies suitable for non-residential buildings in hot-humid climates, and a detailed assessment of a case-study building in Mogadishu, Somalia. This study aims to identify optimal active cooling solutions in terms of energy consumption, thermal comfort, indoor air quality (IAQ), and investment costs, which are paramount in hot-humid climates, since these present high sensible and latent loads due to their extreme weather.

2. A literature review on active cooling strategies

In general terms, active cooling solutions use energy to remove the sensible and/or latent heat from the building, generally through air conditioners (AC). According to global market surveys [7,12], window ACs, ductless mini-splits, and packaged units (hereinafter identified as "conventional units") are the most widespread technologies, mainly due to their low cost and simple installation. Nevertheless, in recent years, the focus has shifted to high-efficiency solutions, such as desiccant or radiant cooling [13–15] and AC systems powered with renewable energy, to reduce their environmental impact.

Based on the functions of the plant, an active cooling system can be organized into four main subsystems: i) generation, ii) distribution, iii) ventilation/air-handling, and iv) emission or in-room terminal units. Similarly, considering the fluid used to transfer thermal energy, active cooling plants can be classified as all-air systems, all-water systems and direct expansion (DX) systems, and can also combine two or more of the previously mentioned groups, being air-water system the most commonly used. Today, there are also air-water systems that handle sensible loads (through water terminals) and latent loads (through ventilation subsystem) separately and operate at different conditions, and these plants are known as "Separate sensible and latent cooling systems" (SSLC). Figure 1 shows a simplified scheme of the most common configurations of an active cooling system. It should be pointed out that some restrictions may exist for the connections between the components presented in this scheme, as for example, a water-LiBr absorption chiller, which cannot be coupled with an air-cooled condenser.

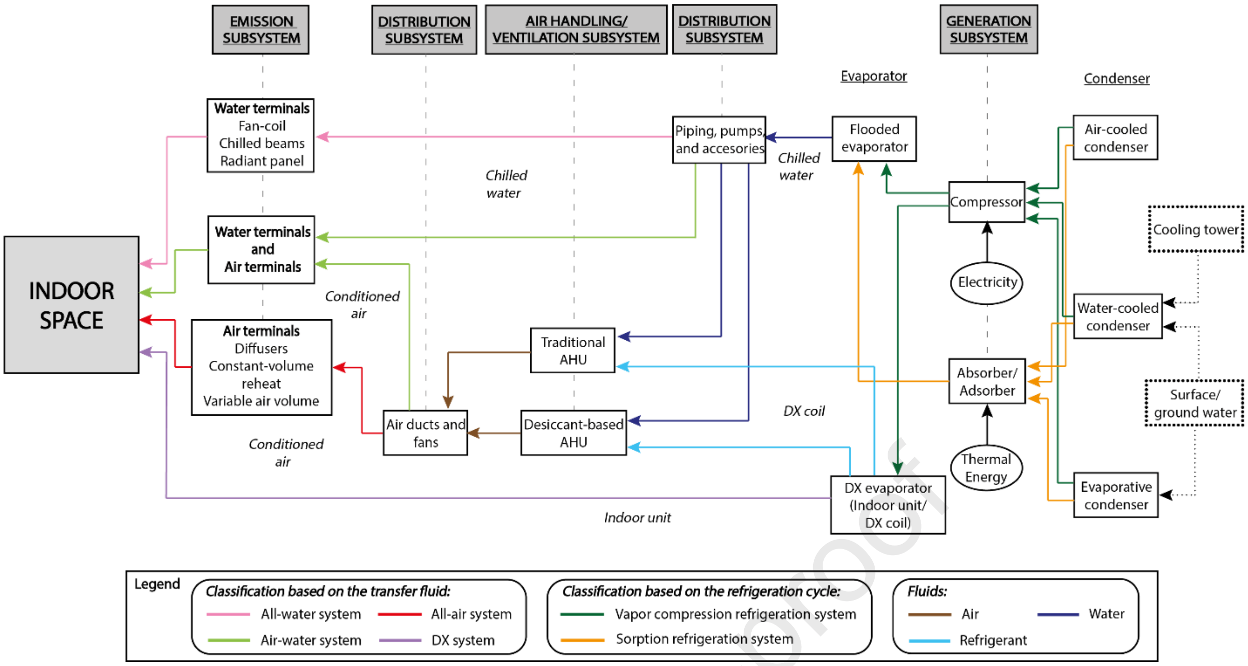


Figure 1. Scheme of the most common configurations of an active cooling plant (some restrictions may exist for the connections between components).

In the following subsections, generation, air-handling, emission subsystems, as well as other components (i.e., ceiling fans), and SSLC technologies are reviewed, to identify the best solutions for their application in non-residential buildings in hot-humid climates.

2.1. Generation subsystems: vapor compression chillers and sorption chillers

There are two main solutions to subtract thermal energy: vapor-compression and absorption/adsorption systems. Vapor-compression refrigeration systems (VCRS) can be further categorized into direct expansion (DX) systems and chillers. As introduced, DX systems are the most commonly used technologies in small to medium-sized buildings, due to their simple design, compactness, and low installation cost [16]. Among this group, variable refrigerant flow (VRF) systems have gained much attention in recent years as they can compensate their high initial costs by offering lower operational costs than conventional units [17]. However, VRF units and other DX systems lack a ventilation function and accurate humidity controls, but also present environmental risks regarding the high amounts of global warming potential refrigerants used for their operation [17–20].

In contrast, vapor-compression chillers are classified as air-cooled and water-cooled chillers. Water-cooled chillers exhibit higher energy efficiency ratios (EER), since they reject heat from the refrigerant by using water from a cooling tower or other sources (i.e. groundwater, river, lake), working at lower condensing temperatures than in air-cooled chillers [21–23]. In addition, the heat capacity of water – which is higher than the air's – allows for more compact plants. However, their use implies a more complex work to obtain the water and must consider the water regulations of the site, but this can be solved by incorporating a cooling tower. Air-cooled chillers are preferred in regions where water is scarce or expensive, but they are strongly dependent on ambient temperature, which reduces their performance in very hot climates. To overcome this issue, improvements on these systems are based on evaporative technologies [24].

From the chiller performance perspective, the EER of an evaporative chiller is about 1.3-1.5 times higher than a traditional air-cooled chiller [25,26]; however, water-cooled chillers outperform the evaporative ones by providing 14% more refrigeration capacity and achieving 10% higher EER [22].

In comparison with VCRS, sorption chillers are thermally driven technologies and can use low-grade thermal energy to operate. In this regard, solar energy is usually utilized, particularly in hot climates, where the cooling necessity coincides with high solar radiation values [27–29], but in case of insufficient solar resources (or waste heat), they require a backup heater. Generally, solar sorption systems present COP below 1 [28,30]; in this respect, VCRSs still outperform sorption chillers and are preferred in cooling mode [31]. Besides, sorption cooling plants are generally larger and require higher investment costs than VCRS. Thus, improvements in solar system technologies are yet to be achieved to increase their efficiency while reducing their costs [32].

When comparing solar cooling systems, PV-driven VCRS are more energy-efficient than solar sorption systems, while the latter solution is more suitable from an environmental perspective [33,34]; also, today both alternatives are economically comparable [35]. In this regard, Eicker et al. [36] compared a PV-powered VCRS with a solar thermal cooling system in different climates; particularly, in the city with the highest latent loads, they found primary energy savings above 9% regarding the case when no solar energy is used. However, it was observed that the choice between solar technologies depends on their costs and application context (e.g., grid situation), resulting in lower overall costs of PV-driven VCRS system when only cooling was required [36].

2.2. Air-handling subsystems: Traditional air-handling units (AHU) and Desiccant-based air-handling units (DAHU)

AHUs can cool, heat, dehumidify, humidify, and circulate the outdoor air to ensure IAQ and thermal comfort. In order to handle both sensible and latent cooling loads, the air is cooled below the dew point temperature, which implies a high energy consumption of the chiller.

Some strategies have been examined to reduce the cooling load in traditional AHUs by improving the thermodynamic processes using energy recovery units [37,38] and evaporative coolers [39]. For instance, Yari et al. [40] used air-to-air heat exchangers in an AHU with 100% fresh air and an AHU with return air and observed that the use of such devices was more effective in the case of 100% fresh air, where the cooling load was reduced by 32%. Besides, they evaluated the performance of the air-to-air heat exchanger working at different humidity levels of the inlet air; their results showed higher cooling load reductions of 27% in humid conditions, highlighting the benefits of using these systems in humid environments. Similar results were observed by Min et al. [41] when comparing the performance of an AHU with an indirect evaporative cooler and an AHU with a heat recovery wheel (HRW). They reported that both systems achieved annual energy savings up to 26% compared to a traditional AHU but, the one with a HRW was more suitable for applications in cities with high latent loads.

Alternatively, desiccant cooling technologies emerged as a solution for handling latent loads independently, thereby reducing the energy consumption for the AHU chiller. Desiccant wheels can be regenerated with any heat source energy, even low-grade such as solar energy, reducing environmental impacts [42] and decreasing the dependence on primary resources [43]. Some improvements in these systems are based on the introduction of alternative desiccant materials to silica gel, such as polymer desiccants [44,45] or

Titanium Dioxide [46], which can improve the dehumidification capacity of the system working at low regeneration temperatures (around 50°C).

Nevertheless, it has been observed that desiccant systems do not perform well in extreme climates, especially in very high-humidity conditions [47,48], and therefore, higher regeneration temperatures are required [49]. In fact, when temperatures below 60°C are used to regenerate the system, its performance is similar to an energy recovery unit without regeneration heat [50]. In this regard, Fong and Lee [51] re-explored the best use of the desiccant wheel as a component of a solar cooling system when performing in a hot-humid climate. Their simulations revealed that using the desiccant wheel only for energy recovery, rather than humidity control, can further decrease primary energy consumption by 11.6% compared to a typical solar desiccant cooling system with regeneration. Thus, further efforts are required to optimize the performance of desiccant systems when operating under extreme climates.

2.3. Emission subsystems: fan-coils, active chilled beams, and radiant panels

Apart from all-air systems for cooling distribution, there are water-based in-room terminals or emission subsystems such as fan-coil units (FCU), active chilled beams (ACB), and radiant panels [52]. Commonly, these technologies require less space, due to piping systems, and less energy for pumps than air ducts and fans in conventional air units [24].

Generally, FCUs are used only for temperature control and in applications where ventilation requirements are minimal, and a precise management of the latent load is not required [52]. However, FCUs can be coupled with an air system to provide ventilation and outdoor air treatment.

On the other hand, ACBs present many advantages, such as easy maintenance, noiseless operation, and low energy consumption for fans. ACBs can manage the sensible loads of the space by convection and treat limited latent loads by using the induced room air that comes from an AHU. Consequently, ACBs are not suitable for hot-humid climates or high-density buildings characterized by high latent loads. Indeed, to improve their energy efficiency, they should be coupled with air systems to provide adequate ventilation levels, handle the latent loads, and reduce condensation risk. ACBs can also be designed to operate at high water temperatures to reduce energy consumption [53], providing energy savings in the range of 10-20% compared to variable air volume (VAV) systems [54].

Radiant panels for cooling are also being studied by researchers worldwide. These systems handle only sensible loads, primarily through radiation, and therefore, they should also be complemented with ventilation systems, to deal with the latent load and control the indoor humidity [55] to avoid condensation problems [56–59]. However, they work with higher water temperatures (above 16°C) than FCUs and air-handlers, ensuring higher chiller efficiency [60]. In this sense, radiant cooling systems have exhibited energy savings of about 30% compared to FCUs [61] and reductions of about 20% compared to all-air systems [13,62].

In terms of thermal comfort, all these terminals can produce comfortable conditions in hot-humid zones, being ACB better than FCU [63], while radiant panels can produce more uniform conditions than ACB and air terminals [57,64,65]. Nevertheless, it is always recommended to introduce air movement through ceiling fans or other systems, to reduce the sensation of stagnant air and air stratification, mainly in the case of radiant floor panels [66].

2.4. Separate sensible and latent cooling (SSLC) systems

Today, it is common to use two or more cooling systems to handle sensible and latent loads independently. In the existing literature, this approach is called "separate sensible and latent cooling" (SSLC), which consists mainly of three types [67]: 1) air dehumidification/ventilation system used to treat the air and a supplementary radiant panel to handle the space sensible load [68], 2) two parallel vapor-compression cycles, one for sensible loads (no radiant panel) and one for latent loads or DX based system with two evaporators [69], and 3) air systems coupled with desiccant units, which were briefly reviewed in Subsection 2.2. Studies on SSLC have shown their potential for energy savings, between 15 and 47% compared to conventional systems [70]. This subsection includes a brief review of the first type of SSLC, since these systems have been more studied in recent years. Among these, different combinations of air dehumidification/ventilation systems have been coupled with radiant panels to solve ventilation problems in these systems and treat latent loads while providing significant energy savings. In this sense, reductions of 20-40% in cooling energy consumption compared to all-air systems have been observed in regions with hot-humid summers when combining radiant panels with AHU [68,71], desiccant-based AHU [68,72–74], and other outdoor air systems [62,75]. However, in terms of thermal comfort, they may still present limitations, especially in very humid climates, because to handle the high cooling loads at certain hours of the day, the system may overcool the space during the remaining hours. For instance, Saber et al. [76] examined the potential of a decentralized dedicated outdoor air system (DOAS) coupled with a radiant panel to provide thermal comfort in tropical Singapore and reported that thermal comfort was achieved only during specific hours of the day. Therefore, an automatic control was required to modulate the capacity of the ventilation and radiant systems under different indoor and outdoor conditions. The latter is one of the challenges of SSLC systems, associated with the different operating principles of the outdoor air and radiant units. To overcome this issue, Hu et al. [77] explored a strategy based on the intermittent supply of fresh air by prioritizing the use of the radiant system in unoccupied periods. In this sense, the intermittent operation avoided extra energy consumption in the range of 68-157% compared to the schemes with continuous fresh air supply, allowing for better comfort conditions during the occupied periods, considering that radiant systems have long response times [77]. Overall, SSLC presents an inherent complexity and can imply higher initial costs, leading to the need for further studies to better assess the applicability of these technologies in extremely humid and developing regions.

2.5. Ceiling fans

Ceiling fans as stand-alone devices or as a support for ACs have received further attention since they are perceived as a more sustainable and simpler alternative for cooling. Their effectiveness in providing thermal comfort at different airflows and speeds has been evaluated in hot-humid climates [78–83], allowing for higher indoor conditions to achieve energy savings in these contexts. For instance, Mihara et al. [78] found reductions of about 26% in annual energy consumption when increasing the space temperature from 24 to 27°C, while integrating a DOAS with parallel VAV systems and ceiling fans to provide indoor thermal comfort in the tropical climate of Singapore. Likewise, it has been observed that it is possible to reach acceptable comfort conditions even at 30°C/80%RH when providing air movement through ceiling fans at 1.2 m/s without causing dry-eye discomfort [79]. Higher temperatures/RH also require higher air speeds, that can cause inadequate conditions for users. In this sense, Huang et al. [84] reported that, at 34°C, the human thermal sensation was always below 0.5 even at a high air speed of 2 m/s, but also that people preferred to avoid the feeling of wind blowing and noise.

Recently, the blowing direction of ceiling fans has also been tested to determine the best configuration for spatial uniformity of thermal comfort [83,85,86]. In this sense, Raftery et al. [87] reported that fans blowing upwards (in reverse) produced a more homogeneous air speed distribution, regardless of their location in the room, than when blowing downwards. Although the air speeds achieved in the former (average: 1.17 m/s) were lower than in the latter (average: 1.56 m/s), the authors concluded that they were sufficient to ensure comfort conditions.

2.6. Identification of the most suitable active cooling systems for their use in hot-humid climates

Selecting the most effective active cooling system considering both climate and context is a challenging task, that entails a comprehensive analysis of the available technologies. In practice, this decision often depends on the investment costs, space restrictions, and other criteria that do not always pursue energy savings or GHG emission reductions.

According to the described scenario, designers in hot-humid environments lack clear, comprehensive guidelines that support the selection of suitable cooling plants beyond standard configurations, in order to identify the best option according to the site and building conditions. In addition to technical solutions, little is known about the optimal indoor conditions to ensure thermal comfort and IAQ, while providing energy savings regarding typical temperature set-points and relative humidity.

To support the decision-making process in non-residential buildings in hot-humid climates, Table 1 presents a comparative analysis of the different active cooling technologies reviewed in this section. In the following sections, indoor conditions are also analyzed, to provide an integrated assessment on the active cooling design in these contexts.

From the point of view of the generation subsystems, water-cooled chillers exhibit more technical advantages, but their use is restricted to the water regulations and on-site availability; therefore, air-cooled chillers can always be an option in these cases. In addition, among air-handling systems, traditional AHUs coupled with energy/heat recovery units present more benefits for their use in buildings that require 100% fresh air in hot-humid climates compared to DAHUs, especially in terms of plant simplicity and costs. In contrast, radiant cooling panels outperform other water terminals, due to their high energy savings and thermal comfort levels, but must work in parallel with ventilation subsystems. In this sense, the reviewed SSLC approaches are also good options, since they integrate the benefits of stand-alone systems; however, this increases the complexity and costs of the solution. Overall, regardless of the cooling plant, ceiling fans support the adoption of high indoor temperatures for energy savings, while maintaining adequate thermal comfort levels in these contexts.

Table 1. Pros and cons of different active cooling systems.

Active cooling system	Pros	Cons	Cost*	Climate	References
Generation subsystems					
Air-cooled chiller (ACC)	• Advantageous when groundwater or surface water is scarce or has inadequate temperature. • When coupled with evaporative technologies, the EER is 1.3-1.5 times higher than a traditional ACC.	• The air-cooled condenser handles condensing temperatures ~15-20°C above the ambient air, which reduces the EER of the system compared to WCC. • High energy consumption and possible noise problems due to the fan in the condenser.	Medium	All (Except extreme hot climates in traditional ACC and extreme humid climates in evaporative ACC)	[21–26]
Water-cooled chiller (WCC)	• Higher EER due to lower condensing temperatures compared to ACC. • Lower acoustic and aesthetic impact compared to ACC.	• Need of more complex works (e.g., groundwater wells, water heat exchangers, etc.) • High maintenance costs compared to ACC. • Must consider the water regulations of the site.	Medium - High	All (Better performance in dry to moderate humid climate)	[21–24]
DX system	• Simple design and less space for installation compared to chillers. • Low operation and maintenance costs.	• Not suitable for large-size buildings. • Lack of accurate temperature and humidity controls. • Require a significant amount of refrigerants to operate, with a possible high global warming potential.	Low-Medium	All	[16–20]
Solar absorption/adsorption chiller (gas-driven systems are excluded due to the limited availability of such resource in the application context)	• Can use low-grade energy sources (solar energy/waste heat); therefore, potential for reducing the fossil fuel consumption and GHG emissions. • Suitable for off-grid applications. • Require low temperatures for hot water (generally below 100°C), which can be achieved using solar collectors.	• Elevated capital cost and long payback period. • Backup heaters may be required to support the solar collector plant due to its weather dependence. • They require large spaces (~50% more than VCRS.) • Not applicable for buildings operating at night hours. • Their efficiency is dependent on future improvements in solar technologies and currently VCRS outperform them.	Medium - High	Hot and sunny	[27–32,36]

Air-handling subsystems					
Traditional AHU	• Can handle both sensible and latent loads without additional components. • Can provide 100% outdoor air for ventilation. • AHUs coupled with HRW, or evaporative coolers can provide energy savings above 20%.	• Individual room-level control is complex without water terminals. • The air is cooled close to the dew point temperature, which implies high chiller energy consumption. • Require more space for ductwork than water systems using piping installation.	Low - Medium	All	[37–41]
Desiccant-based AHU	• Do not require temperatures close to the dew point for the dehumidification process. • Regeneration process can be accomplished by using low-grade energy sources, which can decrease the dependence on primary resources. • Higher EER of the chiller than traditional AHU.	• More complex plant because it requires thermal systems for the regeneration process. • Regeneration temperatures should above 60°C for better performance. • Their efficiency is dependent on improvements in desiccant materials and solar technologies.	High	All (Except extreme hot-humid climates)	[42–51]
Emission subsystems					
Fain-coil unit (FCU)	• Can be installed in different locations (wall, floor, and ceiling). • Low installation costs.	• Can be used only for recirculating air. Therefore, a parallel ventilation system should be installed. • Supply water is usually set at lower temperatures compared to other water terminals, which reduces the EER of the chiller. • Their design includes fans, which may be noisy and increase the energy consumption compared to other water terminals.	Low	All (Except extreme humid climates)	[24,52]
Active chilled beam (ACB)	• Simple maintenance and controls. • Can operate at higher water temperature and achieve better thermal comfort levels than FCU.	• Cannot handle latent loads and require coupling with and additional ventilation system. • Not suitable for spaces with high latent load requirements. • Can present condensation problems and risk of water leaks.	Low	All (Except extreme humid climates)	[24,53,63,65]

	• Can provide energy savings of about 10-20% compared to conventional air systems.				
Radiant panels	• Supply water temperature above 16°C, which increases the EER of the chiller. • Ceiling panels provide better thermal comfort and uniform conditions than ACB and air systems. • Can present energy savings of about 20-30% compared to conventional air systems and FCU.	• Cannot handle latent loads. For large fresh air needs and latent loads, they require a parallel air system. • Their cooling capacity is limited (usually below 100 W/m^2), and it is not suitable for spaces with high cooling load requirements. • Risk of condensation. • Risk of stagnant air and stratification when using floor panels.	Medium	All (More robust control for application in humid climates)	[13,24,55–62,66]
Separate sensible and latent cooling systems					
Air dehumidification/ventilation system + radiant panel	• Independent and dynamic temperature and humidity controls. • Can provide energy savings between 20-40% compared to conventional systems. • Allow for setting high indoor temperatures.	• More complex system because involves two different technologies. • Cannot work at fixed operating conditions; requires an automatic and more robust control.	High	All (More robust control for application in extreme humid climates)	[62,68,71–77]
Fans					
Ceiling fans	• Offset the comfort zone by increasing the air velocity of the space, providing opportunities for higher temperature set-points (above 26°C). • Create more uniform thermal comfort conditions when using blowing upward (reverse) distribution.	• Cannot work as stand-alone systems in extreme hot-humid climates. • The system does not allow for temperature and humidity control.	Low	All	[78–87]

*Costs vary according to the region and its context.

3. Case-study description

According to the literature review, the application of the most suitable technologies for hot and humid climates was assessed and compared on a real case-study in Somalia.

3.1. Location and climatic conditions

Mogadishu has a hot-humid climate, defined by the Köppen-Geiger climate classification [88,89] as semi-arid (BSh) and tropical savannah (Aw) around its coastline [89]. Figure 2 shows the main climate conditions in Mogadishu. The monthly average temperatures vary between 26.5 and 30.4°C. The annual average relative humidity is about 64% and monthly values range from 57.5 to 72.1%. The monthly global horizontal radiation varies from 5.4 to 7.4 kWh/m^2, exhibiting its maximum value in March. Based on its climate, Mogadishu has an annual number of 3,711 Cooling Degree Days (CDD), calculated considering a base temperature of 18°C.

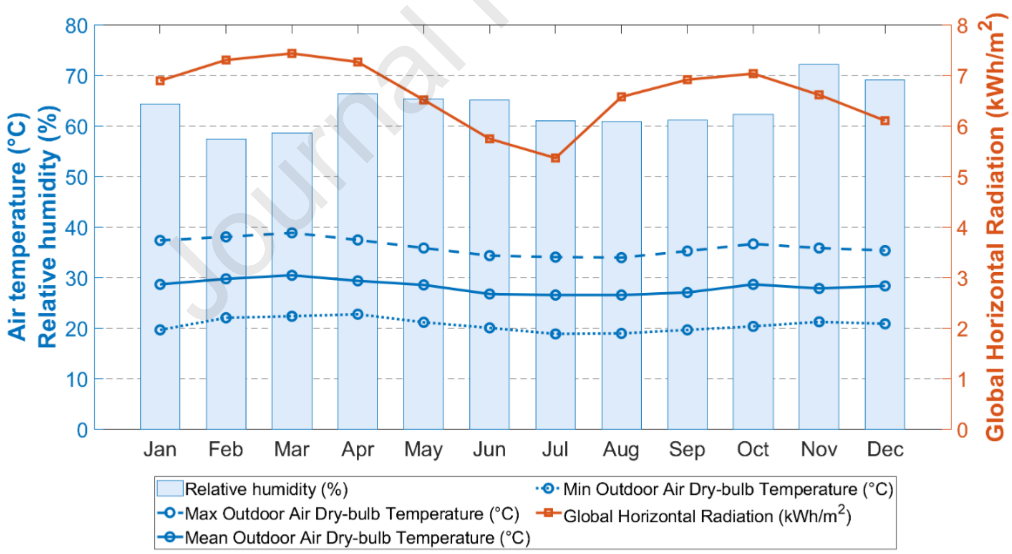
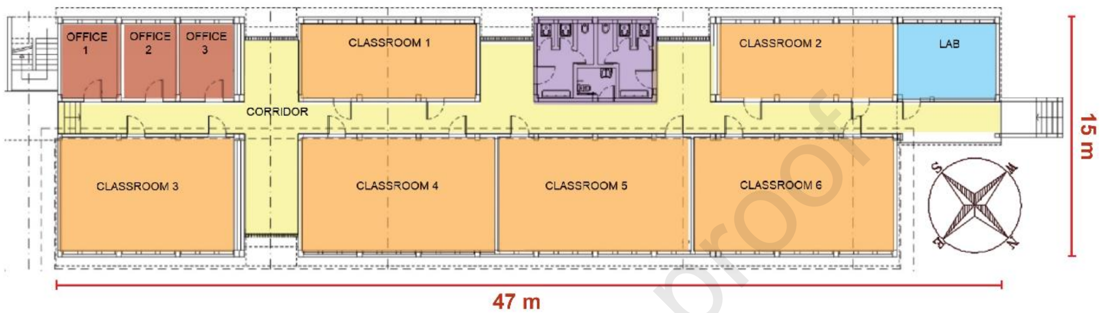


Figure 2. Climate conditions in Mogadishu, Somalia (EPW format, [90]).

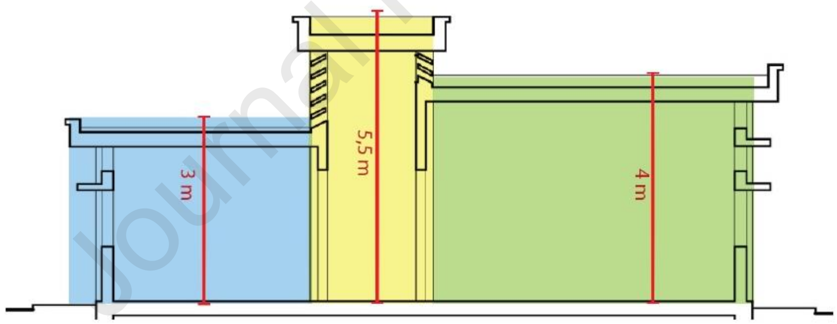
3.2. Building description

The case-study is a representative mixed classroom building located in Mogadishu and is part of the Campus of the National University of Somalia. It has a usable area of about 600 m^2 and a volume of 1800 m^3. It is a single-story building 47 m long and 15 m wide, longitudinally oriented according to the south-east/north-west axis (see Figure 3a).

The described layout is characterized by a more complex organization of the volumes in elevation: the corridor reaches an internal height of about 5 m from the ground line, while the rooms located north and south 4 m and 3 m high, respectively (see Figure 3b).



(a)



(b)

Figure 3. Layout of the reference building: (a) Plan view and (b) Cross sectional view.

Table 2 lists all the reference features of the building's envelope and its internal loads. Lighting, plug loads, and occupancy schedules were set considering the building operation schedule from 9:00 am to 8:00 pm during the weekdays as a traditional university classroom.

Table 2. Building materials and internal loads.

Item	Description	Values
Envelope		
Walls	Medium-weight concrete blocks, finished with cement-lime plaster	U = 0.8 W/m^2-K
Roof/Floor	EPS insulation layer added to the reinforced concrete slab	U = 1 W/m^2-K
Windows	Double glazing system	SHGC = 0.6 U = 2.7 W/m^2-K
Internal loads		
Classroom Other spaces	Lighting and plug loads	6 W/m^2 2 W/m^2
People	Number of people (N_{people}) Vapor mass per person ($\dot{m}_{people}$)	400 120 g/h

The energy and power demand as well as the free-floating behavior (which means that the simulation is run without the active cooling system) of the building have been simulated using EnergyPlus software [91]. Given the size and the destination of the spaces inside the building, the virtual model has been subdivided in 19 thermal zones. An image of the virtual model is provided in Figure 4.

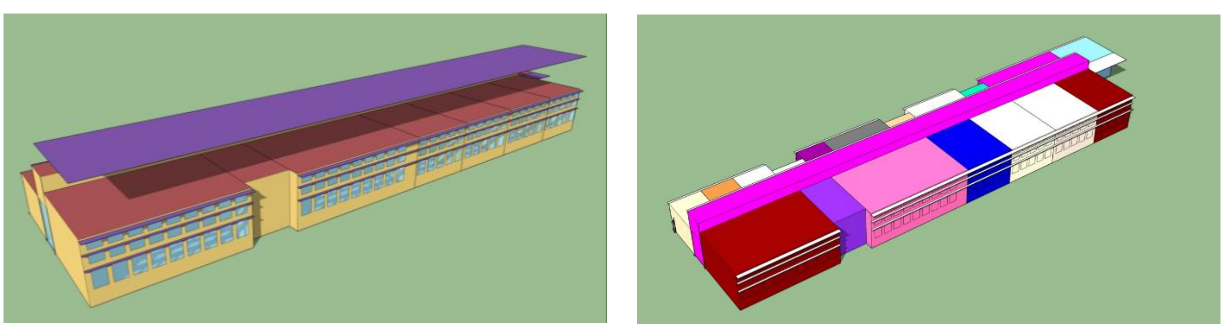


Figure 4. Isometric view of the virtual model in EnergyPlus (left) and thermal zones distribution (right).

4. Assessment of cooling energy demand

In the present section, a detailed assessment of the cooling energy demand of the case-study building is reported. First, a thermal comfort analysis is presented, to determine the best option in terms of indoor temperature/humidity, to ensure comfortable conditions while minimizing the energy demand.

Subsequently, the cooling loads are calculated excluding ventilation and then a detailed analysis is presented, to determine the best outdoor flow rate.

4.1. Thermal comfort analysis

The typical set-point temperature used in summer conditions worldwide has been observed to be 26°C [92–94], but considerable energy demand is required to maintain the space at this temperature [95,96]. Thus, there are application cases where higher temperatures are used to obtain relevant energy savings. For instance, increasing the indoor temperature from 26 to 27°C allowed for potential energy savings up to 34%, as shown in [97]. Nevertheless, the acceptance of high temperatures must be assessed to prevent discomfort. In this regard, there are two main approaches to assess indoor thermal comfort: the predicted mean vote (PMV) [98] and the adaptive models [99].

The PMV model offers a way to forecast the response of people on their perception of thermal comfort based on a scale (-3 cold, -2 cool, -1 slightly cool, 0 neutral, +1 slightly warm, +2 warm, +3 hot) [98], and according to the ASHRAE Standard 55, the acceptable PMV range is between -0.5 and +0.5 [100]. On the other hand, the adaptive model is based on a linear relationship between the indoor design temperatures and outdoor climatic parameters, also considering the "behavioral adaptation" of users to achieve their thermal balance [99], i.e. how people interact and adapt to their environment by changing their clothing levels, switching on/off fans, or opening/closing windows [101].

Despite the fact that the adaptive model has gained more attention for its application in hot-humid climates, it is well-known that the PMV model is more restrictive, since values that move away from neutrality do not necessarily imply discomfort for the users [102,103]. Therefore, in this study, the PMV is used as a conservative approach, assuming that by achieving acceptable conditions with the latter, the adaptive approach will be also met.

To evaluate the thermal comfort of the case-study, the 26°C/50% RH condition is assessed in order to predict the acceptance of users towards the typical space conditions. Using the CBE Thermal Comfort Tool [104,105], the PMV approach was carried out considering the proposed space temperature and humidity, a typical air speed of 0.1 m/s, typical summer clothes (0.5 clo), and the metabolic rate for sedentary activities (1.0 met). As depicted in Figure 5, typical space conditions are in the middle of the range with a PMV between -0.5 and +0.5.

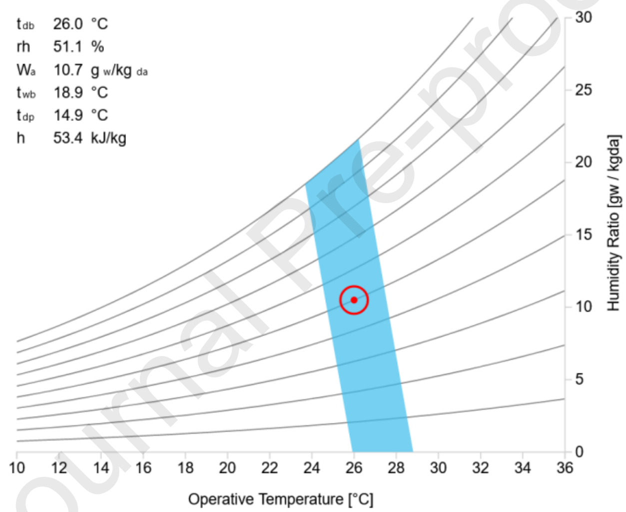


Figure 5. Thermal comfort analysis considering the PMV approach for 26°C/50% RH. The blue area corresponds to the comfort zone, with a PMV between -0.5 to +0.5.

To assess energy saving options, in this study the 28°C/70% RH condition is proposed for hot-humid climates as a good compromise between comfort and the energy saving objectives. To overcome this issue and ensure thermal comfort, adaptive solutions, such as introducing elevated air speeds, could also be adopted to increase the maximum indoor operative temperature under certain conditions [106]. As shown in Figure 6, providing elevated air speeds to the indoor space could improve the cooling effect. Figure 6a offers a different temperature offset for each value of temperature difference between radiant and air temperatures [100]. On the other hand, Figure 6b can be used to determine the temperature offset according to different values of relative humidity [24]. For instance, to obtain an offset of 2 K for 70% RH, an increase in air speed up to 0.6 m/s is required.

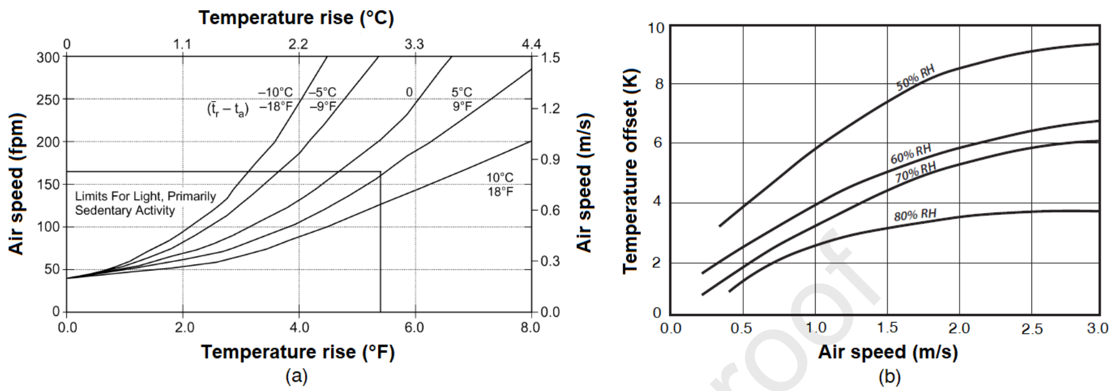


Figure 6. Air speed and temperature offset based on (a) difference between radiant temperature and air temperature [107], and (b) relative humidity [24].

Considering the above principle, the 28°C/70% RH condition was re-analyzed for an elevated air speed of 0.6 m/s. As reported in Figure 7, under these conditions it was possible to reach thermal comfort, because the range shifted approximately 2°C to the right, providing a cooling effect of the same magnitude. In this case, the analyzed conditions were in the middle of the desired range, with a PMV of 0.11 equivalent to a neutral sensation.

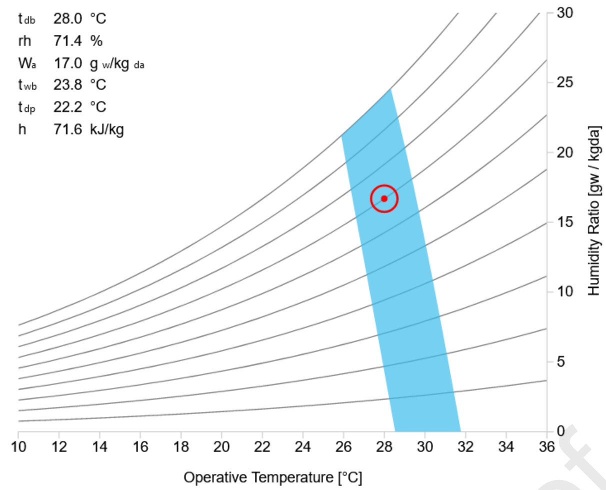


Figure 7. Thermal comfort analysis based on the PMV approach for 28°C/70% RH and elevated air speed at 0.6 m/s. The blue area corresponds to the comfort zone, with a PMV between -0.5 to +0.5.

From this brief analysis, it was confirmed that setting the indoor temperature at 28°C does not ensure thermal comfort if adaptive comfort principles are not considered. Therefore, to achieve thermal comfort in such conditions it is necessary to introduce air movement, with speeds around 0.6 m/s, using ceiling fans [84,108,109], which ensure a constant and homogeneous air distribution.

It should be highlighted that this is an analytical evaluation addressed to predict the users' response to different indoor conditions prior to the active cooling plant design. Thus, some simplifications were considered, based on the climatic conditions in Mogadishu and the building envelope features. More in detail, it was assumed that the mean radiant temperature is equal to the air temperature and, therefore, to the operative temperature, as stated in previous studies [110,111].

4.2. Results of energy simulations on cooling demand

In this section, detailed results of energy simulations applied to the case-study building are reported.

4.2.1. Impact of the indoor temperature and humidity on cooling loads

To assess the impact of the thermostat setpoint on the cooling demand and estimate the energy saving potential in the specific context of Mogadishu, two scenarios were simulated without considering the impact due to outdoor air intake.

- Scenario A: Indoor conditions at 26°C/50% RH and 100% recirculation (no fresh air).
- Scenario B: Indoor conditions at 28°C/70% RH and 100% recirculation (no fresh air).

Space cooling loads, sensible and latent, were calculated separately. Sensible loads were estimated using EnergyPlus [91], considering the features and parameters described in section 3.2, while latent loads (Q_l) were calculated based on the metabolic heat gain from people using Eq. (1 and the parameters from Table 2.

$$Q_l = N_{people} * \dot{m}_{people} * h_{fg} \ (W) \tag{1}$$

where N_{people} is the number of people in the space, $\dot{m}_{people}$,is the vapor mass per person (kg/s/person), and h_{fg},is the enthalpy of vaporization of water (J/kg).

Table 3 shows the results for the cooling loads without considering ventilation loads. As can be observed, cooling loads are higher for Scenario A. This implies that setting the space at higher design conditions reduce the energy consumption for cooling. In fact, Scenario B provided energy savings of 17% in the considered climatic conditions, compared to Scenario A.

Table 3. Results for the annual cooling loads without ventilation.

Scenario	**Outdoor conditions**	**Indoor conditions**	**Cooling energy demand**			**Peak loads**	
			Sensible [kWh]	**Latent [kWh]**	**Total [kWh/m²]**	**Sensible [kW]**	**Latent [kW]**
A	Mogadishu climate (EPW)	26°C – 50% RH	131,787	146,058	463	38	33
B		28°C – 70% RH	83,345	146,058	382	26	33

4.2.2. Impact of outdoor flow rate

A dedicated assessment was carried out on the optimal value of outdoor flow rate. In fact, although ventilation represents an additional cooling load (i.e. outdoor air must be cooled/dehumidified), in the presence of a high internal latent load such as in the proposed case-study, under some conditions outdoor airflow helps to reduce the energy needed for air-handling, since the energy required for such purpose varies with the design indoor conditions. Therefore, if the temperature/humidity difference between outside and inside spaces is negative, ventilation reduces the cooling load.

To examine this effect, the energy demand in presence of a variable outdoor airflow rate was estimated. In this sense, the minimum acceptable ventilation rate for IAQ and the supply airflow rate to meet the space cooling loads were calculated to define the maximum ventilation rate. If the minimum outdoor airflow rate for IAQ is lower than the supply airflow rate, then the difference between both will be considered for recirculation.

In this study, the minimum outdoor flow rate for IAQ was calculated based on the recommendations of the ASHRAE 62.1 [112], using Eq. ((2) and the parameters for lecture classrooms.

For the minimum ventilation rate in the breathing zone (obtained from Table 6-1 in [112]):

$$\dot{V}_{outdoor} = V_{bz} = R_p * P_z + R_a * A_z \ (m^3/s) \tag{2}$$

where R_p is the outdoor airflow rate required per person (m^3/s per person), R_a is the outdoor airflow rate required per unit area (m^3/s per m^2), P_z is the maximum occupancy of the zone during typical usage, and A_z is the zone floor area (m^2).

Similarly, the supply airflow rate to meet the cooling loads of the space was estimated for both indoor design conditions and the supply conditions listed below, following the procedure in [113]. It should be pointed out that, in each case, the supply air temperature was fixed at a state that ensures comfort conditions in order to evaluate the effect of the outdoor air intake on the cooling loads.

- For indoor conditions at 26°C/50% RH, supply conditions at 15°C/ 6.8 g/kg.
- For indoor conditions at 28°C/70% RH, supply conditions at 22°C/ 13.9 g/kg.

The above mentioned supply conditions refer to design conditions (35°C and 60%). Then, it was assumed to work with fixed temperature and variable flow air on the basis of hourly operating conditions. Consequently, cooling loads were calculated using the following equations [113].

For the sensible cooling coil load (Q_{cs}):

$$Q_{cs} = \rho * \dot{V}_{supply} * c_p * (T_{ae} - T_{cc})\ (W) \tag{3}$$

where T_{ae} is the temperature of air entering the cooling coil (°C) and T_{cc} is the temperature of air leaving the cooling coil (°C).

For the latent cooling coil load (Q_{cl}):

$$Q_{cl} = \rho * \dot{V}_{supply} * h_{fg} * (w_{ae} - w_{cc})\ (W) \tag{4}$$

where w_{ae} is the specific humidity of air entering the cooling coil (kg/kg) and w_{cc} is the specific humidity of air leaving the cooling coil (kg/kg).

Finally, the total cooling coil load (Q_{cc}):

$$Q_{cc} = Q_{cs} + Q_{cl}\ (W) \tag{5}$$

Moreover, taking into account the recirculation rates, the equations above must consider the conditions of the adiabatic mixture at the inlet of the cooling coil, which can be calculated based on the expressions in [113].

After performing the calculations for the outdoor airflow rate for the two indoor design conditions, it was found that the minimum rate for IAQ was equivalent to 15.3 m^3/h per person, for a total airflow rate of 6,120 m^3/h. On the other hand, supply airflow rates were calculated considering the above supply conditions and the peak sensible loads reported in Section 4.2.1. In this sense, the supply airflow rate obtained for indoor design conditions at 26°C/50% RH was 10,404 m^3/h, while for 28°C/70% RH it was 12,960 m^3/h. Therefore, the maximum outdoor flow rate was set equal to the corresponding supply airflow rate in both cases.

Figure 8 depicts the impact of the outdoor airflow rate on the cooling load, being the maximum value equal to the supply airflow rate, that is the amount of air needed to maintain internal setpoint temperature. As can be seen in Figure 8a, the energy consumption for cooling is directly proportional to the outdoor airflow rate for ventilation when indoor conditions are set at 26°C/50% RH. Therefore, when the system operates at 100% fresh air, the energy consumption is higher. In contrast, when indoor conditions are fixed at 28°C/70% RH, the variables do not exhibit direct proportion, as shown in Figure 8b. In fact, as the outdoor airflow rate increases, the cooling load decreases until it reaches a minimum value. In this case, this value is equal to 8,539 m^3/h (approx. 4.7 vol/h) and represents the optimal outdoor airflow rate for the lowest energy demand for cooling. After this value, if the outdoor airflow rate continues to increase, the cooling loads also increase, exhibiting a similar behavior as in the previous indoor conditions. This was achieved only in the scenario set at 28°C/70% RH and the latter can be attributed to the fact that, in such specific indoor conditions, the cooling power to treat outdoor air is lower than that, to dispose of the indoor latent load (i.e. the outdoor air absolute humidity is 12.4 g/kg while the supply air absolute humidity is 13.9 g/kg, thus no dehumidification of the air is required and the latent load is reduced). However, if the airflow rate continues to increase, this favorable effect will be offset by the amount of hot-humid air to be treated.

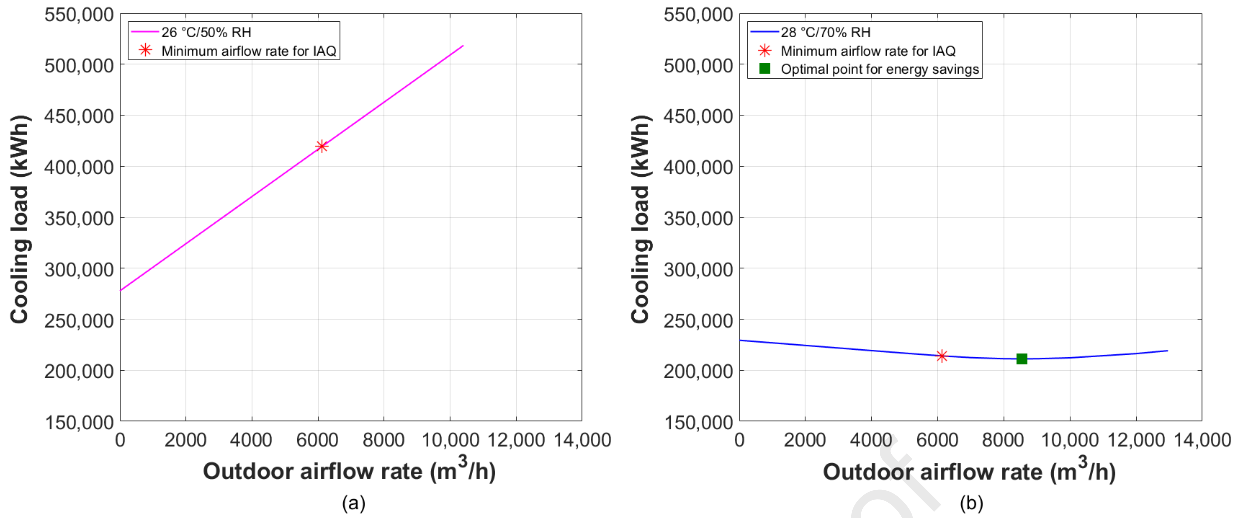


Figure 8. Relationship between the total cooling load and the outdoor airflow rate for: (a) 26°C/50% RH and (b) 28°C/70% RH.

One of the advantages of using systems that provide 100% fresh air is that they reduce the risk of transmission of airborne virus diseases (i.e. COVID-19) [114,115]. Also, it compensates the high CO_2 concentration levels in high-occupancy buildings [116]. However, to avoid high energy consumption requirements, as observed for the 26°C/50% RH condition., a minimum outdoor flow rate that guarantees IAQ must be set in those cases.

Consequently, four scenarios were simulated to evaluate the impact of the ventilation loads on the energy consumption for cooling.

- Scenario A1: Indoor conditions at 26°C/50% RH and the minimum recommended outdoor airflow rate (6,120 m^3/h, equivalent to 59% of fresh air).
- Scenario A2: Indoor conditions at 26°C/50% RH and 100% fresh air (10,404 m^3/h).
- Scenario B1: Indoor conditions at 28°C/70% RH and the optimal outdoor airflow rate for energy savings (8,539 m^3/h, equivalent to 66% of fresh air).
- Scenario B2: Indoor conditions at 28°C/70% RH and 100% fresh air (12,960 m^3/h).

Finally, Table 4 presents the results for the annual cooling loads considering both indoor design conditions and different outdoor airflow rates for ventilation. In both cases, scenarios with 100% outdoor air intake (A2 and B2) were the most energy consuming ones. For instance, Scenario A2 increased the energy consumption by about 24% compared to Scenario A1. In contrast, Scenario B2 exceeded Scenario B1 by only 3.8%. Therefore, Scenario B2 offers the best alternative in terms of IAQ, considering the high occupancy of the building, but its configuration is also simpler, because it does not include recirculation. In general, the difference in the cooling loads due to the impact of ventilation was not very significant in both cases, since the recirculation rate is low and conditions approximate further to the 100% outdoor air scenario. However, the proportional increase in Scenario A2 is more apparent than in the case of Scenario B2, as inspected in more detail in Figure 8.

Table 4. Results for the annual cooling loads considering the impact due to ventilation.

Scenario	Outdoor conditions	Indoor conditions	Outdoor airflow rate [m³/h]	Cooling energy demand: Sensible [kWh]	Latent [kWh]	Total [kWh/m²]
A1	Mogadishu climate (EPW)	26°C – 50% RH	6120	176,158	243,208	699
A2			10,404	207,218	311,213	865
B1		28°C – 70% RH	8539	122,581	88,604	352
B2			12,960	142,897	76,285	365

5. Description of the proposed cooling systems

As explored in the previous section, there are various parameters influencing the cooling energy demand as the indoor temperature/humidity and the outdoor airflow rate. In contrast, the energy consumption of a cooling plant is affected by the working conditions in which its subsystems operate. In this section, we introduce different possible active cooling plants to support the decision-making process in hot-humid climates. To provide a holistic analysis, the influence of the indoor parameters on the selection of the technical solutions is also explored.

All options will be compared considering the energy consumption of their generation and emission subsystems. It will be assumed that all proposed solutions are coupled with an air-cooled chiller, since this is the only viable option to provide chilled water, considering that underground water in Mogadishu has average temperatures above 30°C.

5.1. Baseline: All-air system with standard setpoint (26°C/50% RH)

Figure 9 shows the scheme for the baseline, which corresponds to an AHU in recirculation mode coupled with an air-cooled chiller, operating at a standard setpoint (26°C/50% RH). Here, the outdoor air enters the AHU and is cooled and dehumidified in the cooling coil to reach the supply conditions. All-air systems supply air to the target zone based on the sensible and latent heat gains in the space. Since these conditions vary with time, AHU could operate under two principles [52]: (a) varying the supply airflow rate by maintaining a constant supply temperature, ensuring that it is at least equal to the minimum airflow rate for IAQ, or (b) varying the supply air temperature by maintaining a constant supply airflow rate. In this case, the baseline is considered as a variable volume all-air system that provides the appropriate supply airflow rate to control the space sensible heat gains. Since it operates in recirculation mode, the outdoor airflow intake corresponds to the minimum airflow rate for IAQ (6,120 m^3/h).

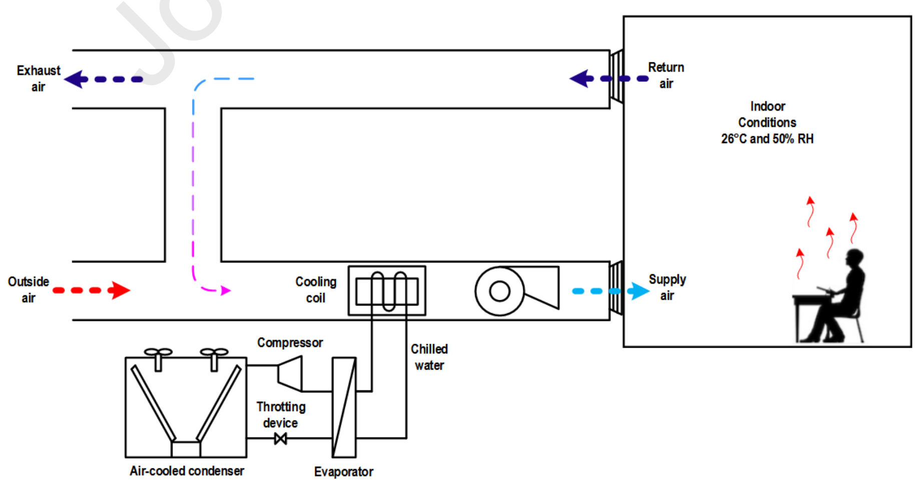


Figure 9. Baseline, typical all-air system with AHU in recirculation mode, working at standard setpoint (26°C/50 %RH).

On the other hand, the air-cooled chiller was modelled using manufacturing data provided by the Aermec company (Eurovent certified) [117]. More in detail, the Magellano software [118] was used to generate the chiller performance curve, according to the proposed design conditions. In this sense, the evaporating temperature is set as 7°C, while the condensing temperature is set as a function of the outdoor air temperature based on the hourly weather data of Mogadishu. Consequently, the hourly energy consumption of the chiller can be estimated via Eq. (6) for the hourly energy demand (Section 4.2) and EER values.

$$W_{in} = \frac{Q_{Cooling}}{EER} \tag{6}$$

where $Q_{Cooling}$ is the energy demand for cooling (kWh), W_{in} is the energy consumed by the compressor to provide work, in (kWh), and EER is the energy efficiency ratio of the chiller.

On the other hand, the energy consumption of the AHU fans in estimated through Eq. (7).

$$E_{fan} = \frac{\dot{V}_{supply} * \Delta p}{\eta_{fan}} \text{ (W)} \tag{7}$$

where Δp is the pressure drop (Pa) and η_{fan} is the fan efficiency (0.63).

5.2. Option 1: Radiant ceiling coupled with an AHU with standard setpoint (26°C/50% RH)

This option consists of a radiant cooling surface combined with an AHU for dehumidifying the outside air and treat the latent load of the space. As depicted in Figure 10, the proposed radiant surface is composed of ceiling panels.

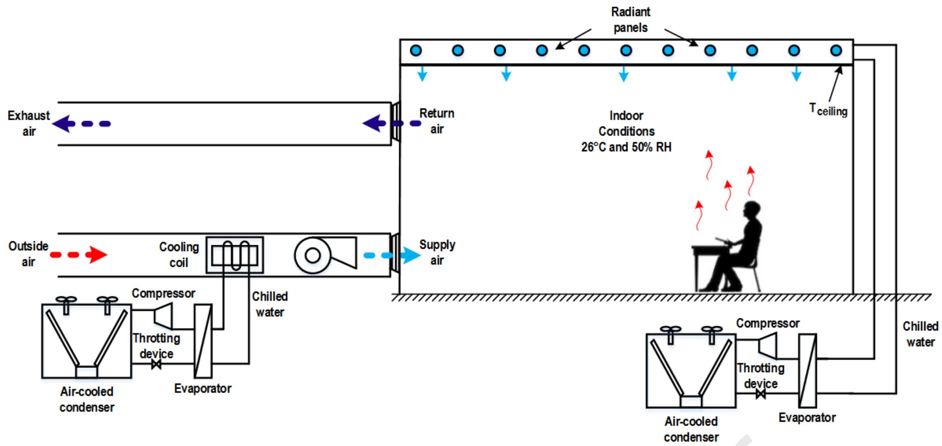


Figure 10. Option 1, radiant ceiling coupled with an AHU in 100% outdoor air mode, working at standard setpoint (26°C/50% RH).

The cooling capacity ($q_{ceiling}$) of the proposed scenario can be calculated following the equations from ISO 11855 [119,120]. For ceiling cooling and floor heating:

$$q_{ceiling} = 8.92 * \left(T_{in} - T_{ceiling}\right)^{1.1} \; (W/m^2) \tag{8}$$

where $T_{ceiling}$ is the temperature of the radiant ceiling panel (°C).

For typical indoor conditions at 26°C/50% RH, radiant panels were designed to account for the total space sensible load, for a maximum cooling power of 88.0 W/m^2. To accomplish this, the surface temperature was fixed at 18°C to avoid condensation on the panel (DPT = 15°C). The total panel area was set to cover no more than 70% of the total ceiling area (420 m^2), which has been observed to be a good practice [121,122]. On the other hand, the AHU was designed to provide fresh air at a rate of 6,120 m^3/h to accomplish minimum IAQ levels.

In this scenario, two generation subsystems are considered. A conventional air-cooled chiller is used to feed supply chilled water at 16°C to the radiant system (radiant chiller). In contrast, the AHU is coupled with an air-cooled chiller working at similar conditions as in the baseline (AHU chiller). In both cases, the EER and energy consumption of the chillers were estimated as indicated in the Baseline. The energy consumption for the AHU fans was calculated using Eq. (7), while the energy consumption for the pumps in the radiant system was estimated through Eq. (9).

$$E_{pump} = \frac{\dot{V}_{water} * \Delta p}{\eta_{pump}} \text{ (W)} \tag{9}$$

where $\dot{V}_{water}$ is water volumetric flow rate (m^3/s) and η_{pump} = pump efficiency (0.87).

5.3. Option 2: All-air system with optimized setpoint (28°C/70% RH) and ceiling fans

As illustrated in Figure 11, Option 2 consists of an AHU, operating at 28°C/70% RH. In this case, there is no recirculation, and the supply airflow rate in design conditions is 12,960 m^3/h for a total outdoor air intake of 100%. The chiller was modelled using the Magellano software [118], and therefore, the hourly EER values were obtained, considering an evaporating temperature of 18°C, while the condensing temperature was set as a function of the hourly outdoor temperature of Mogadishu. Then, the energy consumption of the chiller was estimated via Eq. (6) and the energy consumption for the AHU fans was calculated through Eq. (7) as in previous options.

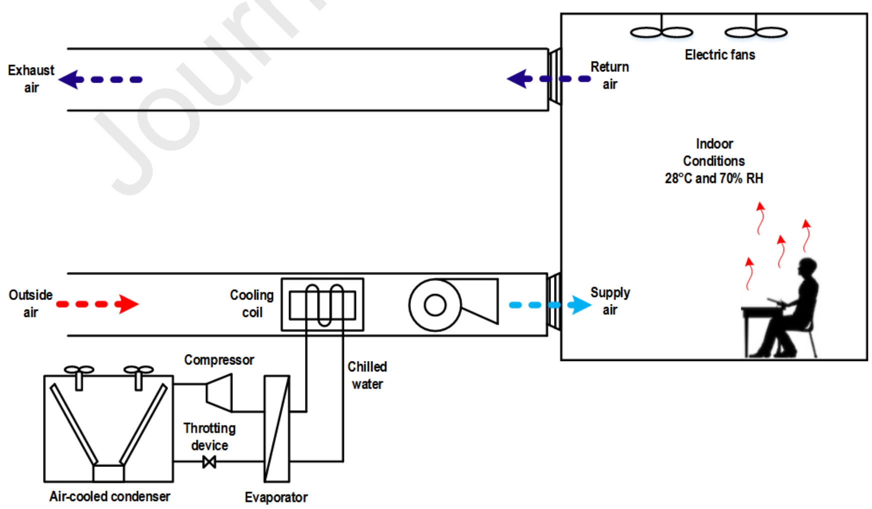


Figure 11. Option 2, all-air system with AHU in 100% outdoor air mode, working at optimized setpoint (28°C/70% RH) and ceiling fans.

Considering that air movement is required to ensure thermal comfort at high indoor conditions, this option includes ceiling fans. In this sense, the layout of the ceiling fans was arranged based on the procedure reported in [123]. As a result, a total of 74 fans, with a diameter of 1.2 m and an average power of 70 W, were estimated to move the air at an elevated speed of 0.6-0.8 m/s at user height. It should be pointed out that the same considerations were assumed for options with indoor conditions at 28°C/70% RH.

5.4. Option 3: Radiant ceiling coupled with an AHU with optimized setpoint (28°C/70% RH) and ceiling fans

Similar to Option1, Option 3 consists of radiant ceiling panels coupled with an AHU, operating at an optimized setpoint of 28°C/70% RH (Figure 12). For the proposed indoor conditions, radiant panels were designed to cover 70% of the total ceiling area to account for about 66% of the total sensible cooling load and a maximum cooling power of 52.4 W/m^2. In this case, the surface temperature was fixed at 23°C, to avoid condensation on the panel (DPT = 22°C). The remaining sensible load and the total latent load were handled by the AHU, which was designed to provide fresh air at a rate of 6120 m^3/h to accomplish minimum IAQ levels.

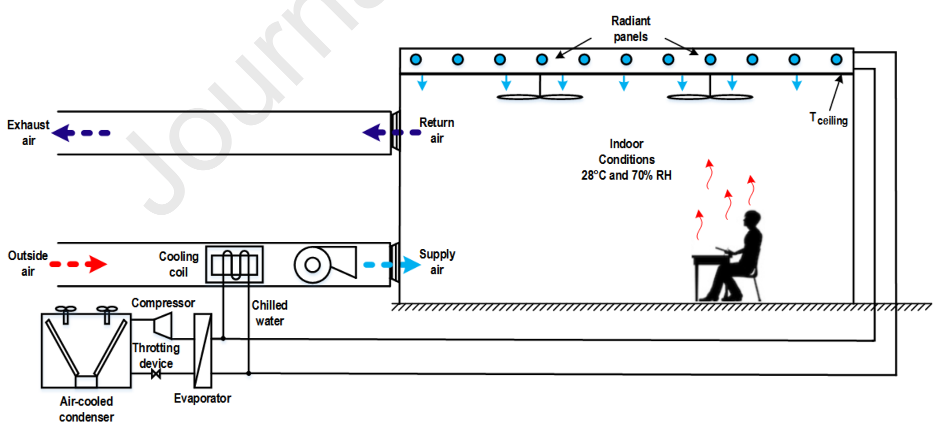


Figure 12. Option 3, radiant ceiling coupled with an AHU in 100% outdoor air mode, working at optimized setpoint (28°C/70% RH) and ceiling fans.

In contrast to Option 1, this configuration requires only one air-cooled chiller to provide supply chilled water at a temperature around 20°C. Therefore, the air-cooled chiller operated as detailed in Option 2. The energy consumption for AHU fans and radiant pumps was calculated via Eq. (7) and Eq. (9).

5.5. *Variants with heat recovery (Options b)*

For every option above a variant is defined, where a heat recovery wheel (HRW) is introduced in each AHU to precondition the outside air and reduce the cooling energy demand, as illustrated in Figure 13. Supply and exhaust air streams flow through the two sides of the HRW. Since in cooling mode the temperature, humidity, and enthalpy of the return air stream are lower than the conditions of the outdoor supply air stream at the inlet of the wheel, sensible heat transfers from the outdoor supply to the exhaust air stream due differences in temperatures [52].

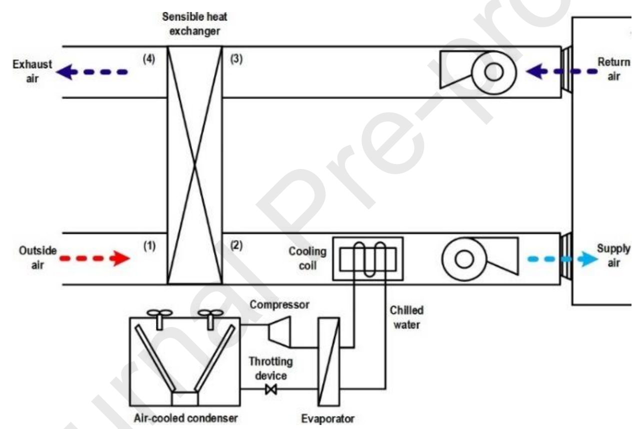


Figure 13. Options b, all-air system with AHU in 100% outdoor air mode and heat recovery wheel (HRW).

The temperature of the fresh air at the outlet of the HRW (cooling coil inlet) can be estimated from the effectiveness of the heat exchanger. For the sensible effectiveness (ε_s) of the HRW and 100% fresh air the following equation holds [52]:

$$\varepsilon_s = \frac{\dot{m}_2 c_{p_2} \times (T_1 - T_2)}{\min(\dot{m}_2 c_{p_2}; \dot{m}_3 c_{p_3}) \times (T_1 - T_3)} \tag{10}$$

where T_1 is the dry-bulb temperature of the outdoor air (°C), T_2 is the temperature at the outlet of the wheel (°C), T_3 is the temperature of the exhaust air inlet (°C), $\dot{m}_2 c_{p_2}$ and $\dot{m}_3 c_{p_3}$ are the capacity rates for each stream (J/s-K), as indicated in Figure 13.

6. Results and discussion

After conducting the annual simulation, it was found that EER values are between 2.4 and 3.7 for 26°C/50% RH and between 3.2 and 5.5 for 28°C/70% RH, which are in agreement with chillers at temperature lifts around 40 K and 30 K, respectively [117,124–127]. Therefore, the annual energy consumption in each case was estimated through Eq. (6), considering the hourly energy demand simulated in Section 4 for each indoor conditions. Figure 14 illustrates the results, showing that all options provided energy savings compared to the Baseline. For instance, Option 1 achieved energy savings of 26%, while Option 2 reduced the annual energy consumption by 51%. On the other hand, Option 3 resulted the less energy-consuming one, with savings of 57%. In all cases, the incorporation of the HRW (options b) reduced the energy consumption of the proposed configurations by 6-11%.

As stated in section 4.2.1, the impact of the temperature and humidity in the cooling demand is evident since, when changing indoor design conditions from 26°C/50% RH to 28°C/70% RH, energy savings of 17% were achieved. Moreover, when the typical all-air system with AHU (Option 2) works at 28°C/70% RH, it reduces the energy consumption of the chiller by 70% with respect to the Baseline, as observed in Figure 14. From the thermal comfort perspective, it was necessary to introduce elevated air speeds using ceiling fans to achieve favorable conditions when working at 28°C/70% RH. In this case, an additional 22,688 kWh were required to operate the ceiling fans annually, and despite this, the proposed configuration needs 51% less electricity compared to the Baseline, which supports the decision of maintaining the space at high indoor conditions. In general, similar savings have been observed and reported in [96,128], which vary according to the climate and building type. Therefore, these results reaffirm that the use of high temperature setpoints combined with elevated air speeds are particularly beneficial in hot-humid regions, where the cooling requirements are high throughout the year.

Similarly, the working principles of each active cooling system also affect the energy consumption for cooling. Figure 14b compares the annual energy consumption of the proposed options and their components at 28°C/70% RH. As can be observed, the incorporation of the HRW in Option 2 (b) offered energy savings of about 11%, with regard to Option 2, by reducing the energy demand for ventilation. Similarly, Option 3, consisting of a radiant ceiling for handling the sensible load of the space and a typical AHU for ventilation loads, reduced the energy consumption of about 11% compared to Option 2. Moreover, coupling Option 3 with a HRW provided savings of about 17% compared to Option 2. Our results for the options with radiant systems differ with previous studies that evaluated the performance of similar systems [68,129,130] since, as reported by the authors, radiant cooling systems working in parallel with mechanical ventilation exhibited higher energy savings of more than 30%. However, there are some differences between these works and the present research that should be highlighted.

- *Low-mid lift:* Setting the indoor air temperature at 28°C allows for higher supply temperatures (e.g. about 22°C instead of 15°C or below) and, therefore, rising the evaporating temperature. Consequently, the chiller can operate at low-mid lift conditions, similar to those for radiant systems, which increase its EER and reduce its energy consumption. In this sense, it is evident that indoor conditions also influence the selection of such options since for the scenario at 26°C/50% RH, radiant options (Options 1 and 1b) are preferable due to their high energy savings of about 30% compared to the baseline at same conditions (Figure 14a). In contrast, little difference is observed when comparing the air system with the radiant one in Figure 14b.
- *Latent loads and fresh air*: in the proposed case-study, latent loads represent about 35% of the total cooling load, while in [68], latent loads share is below 20% for a 18 m^2 office. Even in larger areas, but with low ventilation requirements, it has been observed that the use of radiant systems can be more convenient compared to all-air systems [131]. Our results are also consistent with the study conducted by Wang et al. [132], since they found that by increasing the airflow rate and decreasing the sensible heat fraction (SHF), the exergy of the radiant cooling system with an AHU increases, which implies a higher energy consumption.

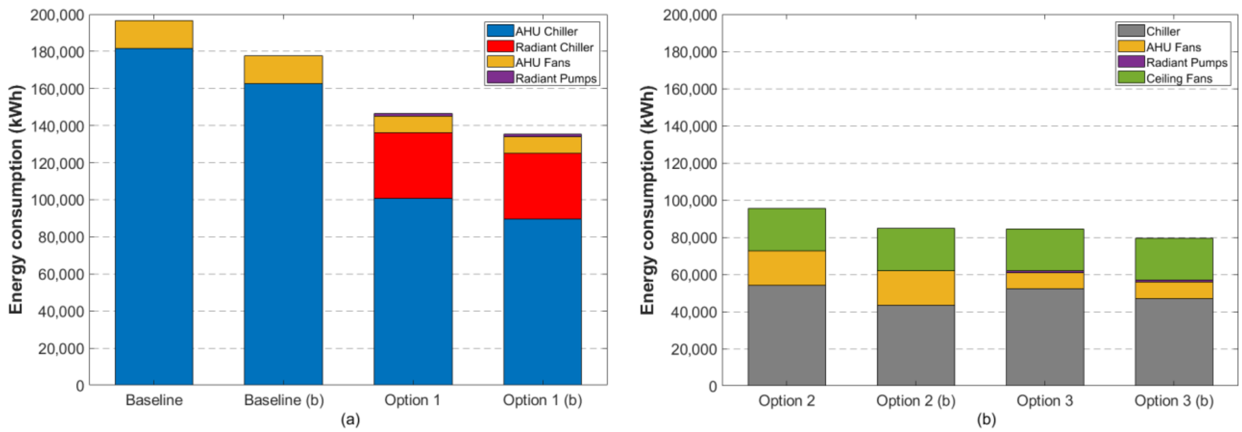


Figure 14. Annual energy consumption of the Baseline, Options 1-3, and variants with heat recovery (Options b), operating with indoor conditions at: (a) 26°C/50% RH and (b) 28°C/70% RH.

On the other hand, if only options 2 and 3 are compared (Figure 14b), Option 2(b) requires almost the same energy as Option 3 to operate, but the outdoor airflow rate of the latter is 53% lower. In general, the analyzed options 2 and 3 show very little difference in terms of energy savings, thus, the decision must evaluate the cost-benefit perspective.

From the point of view of the cost analysis, options 1-3 were compared to the Baseline, considering only the investment and operating costs of the proposed active cooling solutions. Initial costs were fixed based on cases studies, references from manufacturers, and literature values for similar equipment in Europe [133–140]. The sizing parameters and reference costs of each option are summarized in Table 5.

Similarly, the annual operating cost of each option was evaluated, considering that the cost of electricity in Somalia is on average €0.66/kWh (about $0.75/kWh), which is one of the highest worldwide [141]. As can be observed in Figure 15, the Baseline and Option 2 presented the lowest investment costs compared to other alternatives working at the same indoor conditions, given that they have the simplest configuration. On the other hand, Options 3 had the highest investment costs (excluding costs related to advanced condensation controls), showing an increase of about 38-46% with respect to the Baseline. However, if the operating costs are based on the lifespan of this option, this can be attractive in the long term. Therefore, Options 2 and 3 are the most recommended based on the cost analysis. In general, the estimated specific costs are in agreement with those reported in [142] for all air systems (approx. 132–154 €/m^2) and radiant systems (up to approx. 155-195 €/m^2), considering that in this work only ceiling fans accounted for a specific cost of about 31 €/m^2.

Table 5. Sizing parameters and reference costs of components in the Baseline, Options 1-3, and variants (Options b).

<table>
<tr><th>Baseline</th><th>Option 1</th><th>Option 2</th><th>Option 3</th></tr>
<tr><td>Chiller: 200 kW
AHU: 10,404 m^3/h</td><td>AHU Chiller: 110 kW
Radiant Chiller: 40 kW
AHU: 6120 m^3/h
Panel area: 420 m^2</td><td>Chiller: 140 kW
AHU: 12,960 m^3/h</td><td>Chiller: 100 kW
AHU: 6120 m^3/h
Panel area: 420 m^2</td></tr>
<tr><th>Baseline (b)</th><th>Option 1 (b)</th><th>Option 2 (b)</th><th>Option 3 (b)</th></tr>
<tr><td>Chiller: 175 kW
AHU: 10,404 m^3/h
HRW ε: 0.7</td><td>AHU Chiller: 100 kW
Radiant Chiller: 40 kW
AHU: 6120 m^3/h
Panel area: 420 m^2
HRW ε: 0.7</td><td>Chiller: 120 kW
AHU: 12,960 m^3/h
HRW ε: 0.7</td><td>Chiller: 90 kW
AHU: 6120 m^3/h
Panel area: 420 m^2
HRW ε: 0.7</td></tr>
<tr><th colspan="2">Item</th><th colspan="2">Unit cost</th></tr>
<tr><td colspan="2">Air-cooled chiller</td><td colspan="2">230 €/kW</td></tr>
<tr><td colspan="2">AHU</td><td colspan="2">1.2 €/m^3/h</td></tr>
<tr><td colspan="2">HRW</td><td colspan="2">0.4 €/m^3/h</td></tr>
<tr><td colspan="2">Radiant ceiling</td><td colspan="2">100 €/m^2</td></tr>
<tr><td colspan="2">Ductwork</td><td colspan="2">2 €/m^3/h</td></tr>
<tr><td colspan="2">Ductwork (with HR)</td><td colspan="2">3 €/m^3/h</td></tr>
<tr><td colspan="2">Radiant piping accessories
(No controls)</td><td colspan="2">15 €/m^2</td></tr>
<tr><td colspan="2">Ceiling fans</td><td colspan="2">250 €/unit</td></tr>
</table>

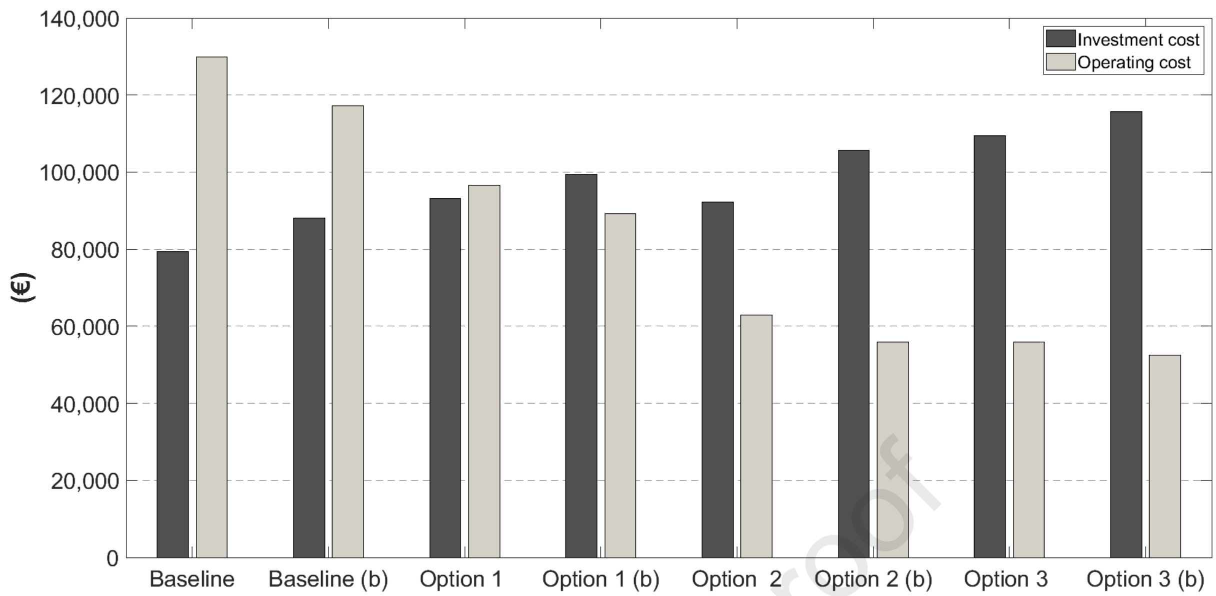


Figure 15. Investment and operating costs (by year) of the proposed options.

Overall, Option 2 shows a low global cost and presents a simplest configuration, with no additional controls required (compared to radiant options) and no exhaust ducts for air recirculation. Considering the context studied, this is a good choice, which provides a complete handling of sensible and latent loads without extra components, and allows for 100% fresh air operation.

7. Conclusions

Window ACs and ductless mini-splits are the most used systems for space cooling worldwide, but they still present several weaknesses related to their low energy efficiency and poor humidity controls. Given that the existing literature is unclear in terms of the most optimal active cooling system to be used in hot-humid climates, this paper analyzes the pros and cons of different strategies for their implementation in these climates. More in detail, this study evaluates the application of two active cooling systems in a building with high latent loads in Somalia: all-air system (with/without heat recovery) and radiant ceiling with AHU. First, the temperature setpoint and humidity of the target space were assessed in terms of thermal comfort and energy savings. Then, the impact of the outdoor airflow rate on the energy demand for cooling was also explored, in order to define the best scenario for IAQ. From both analyses, the following conclusions can be formulated:

- By increasing the indoor conditions from 26°C/50% RH to 28°C/70%, it is possible to achieve thermal comfort conditions if elevated air speeds (i.e. above 0.6 m/s) are introduced through ceiling fans.
- By increasing the indoor conditions from 26°C/50% RH to 28°C/70%, the energy demand for cooling was reduced by 17% and the energy consumption decreased by 51%, considering that the technical solution was based on an all-air system (Baseline vs Option 2).
- The outdoor airflow rate had a significant impact on the energy demand, but presented different behaviors depending on the indoor conditions. In the case of 26°C/50% RH condition, the increase in energy demand is directly proportional to the outdoor airflow rate. In contrast, in the case of 28°C/70% condition, the energy demand decreases as the outdoor airflow rate increases until it reaches a minimum, where it exhibits the highest energy savings. These results are valid in similar climatic conditions in the presence of high latent loads.

The influence of the selected active cooling technologies on the energy consumption was examined in more detail for indoor conditions at 28°C/70% RH. The results are summarized below:

- In general, Option 2 (b) (all-air system with HRW) outperforms Option 3 (radiant panel + AHU) in energy performance if the latter does not include heat recovery technologies.
- Option 2 could be the best choice in terms of the overall costs and due to the simplicity of the system layout and its controls.
- Option 3 (b) contributes to the highest energy savings of about 57%. This solution also presented the lowest costs in long-term, but the advanced controls required for its optimal operation could imply higher costs.

- Both options 2 and 3 are good choices for their application in hot-humid climates, but the selection of one or the other strongly depends on the availability of capital investment, but also on the accessibility of such technologies (i.e. radiant panels) in developing contexts.

Finally, it was also observed that the design indoor conditions can affect the selection of technical solutions. In this sense, considering the proposed active cooling technologies, radiant systems were the preferred ones when setting the indoor temperature/humidity at typical conditions. However, little differences were found for high indoor conditions at 28°C/70% RH.

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Highlights:

- Pros/cons of various active cooling systems in hot-humid climates were highlighted.
- Indoor conditions at 28 °C/70% RH were evaluated for thermal comfort and energy.
- At 28 °C/70%, energy savings of 17% in cooling were achieved.
- AHU+HWR and radiant ceiling + AHU were compared in terms of performance and costs.
- Both options exhibited potential for their use in hot-humid climates.

Declaration of interests

☒ The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

☐ The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: