



Critical analysis of the design of atmospheric water generators based on vapour compression refrigeration cycle: A numerical study

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ABSTRACT

Water scarcity is a problem affecting millions of people in the world, which is amplified by climate change, pollution and population growth. To address this problem, an Atmospheric Water Harvesting system is evaluated, introducing a novel critical analysis using five refrigerants typically used by manufacturers and broadens the current spectrum of ambient conditions. The system is mathematically modelled by integrating semi-empirical models of every component. Each semi-empirical model is individually validated, with errors ranging from 2.5% to 6%. The system is evaluated through a) a parametric analysis and b) an annual analysis using climate conditions of seven cities and five refrigerants. From the parametric analysis it is concluded that the trend of the main indicators is similar for the five refrigerants. The best operating conditions are located at ambient temperatures lower than 25 °C with relative humidities within 40% and 80%. The results are highly influenced by the control system. Cities, such as Beijing and Kabul, having low temperatures during winter, present almost 5000 non-operating hours. In terms of water collection, Sao Paulo presents the highest values, ranging from 3.58 L h⁻¹ for R1234yf to 4.54 L h⁻¹ for R407C. For the SEC, the best performance is also obtained in Sao Paulo, with R407C having the best SEC of 0.34 kWh L⁻¹ and refrigerants R134a and R1234yf having the worst SEC of 0.43 kWh L⁻¹. According to the limitations found for this technology, it is recommended to consider a system with an adaptive control and to combine it with sorption systems.

1. Introduction

Most freshwater used for human consumption comes from rivers, reservoirs, lakes, or groundwater (United Nations, 2025). Droughts caused by climate change have led to a shortage of these sources, affecting at least 4 billion people in the world (UNESCO, 2025). Additionally, many projections claimed that more than half of the total population will be affected by water stress by 2050 (Boretti and Rosa, 2019). This context increases the interest in alternative sources and devices to produce fresh water. An alternative, with a lower environmental impact than seawater desalination (Shahzad et al., 2017), are the Atmospheric Water Generators (AWG), which avoids the production of brine as a by-product (Alvez et al., 2020).

Active refrigeration is one of the most promising methods for the AWG technologies, which condenses the moisture content in the atmospheric air by cooling it below its dew point temperature (Tashtoush and

Alshoubaki, 2023). Among the systems used to generate this effect, Vapour Compression Refrigeration (VCR) systems are of special interest because of their extended operation and relatively higher Water Harvesting Rates (WHRs) compared to other alternatives, such as thermoelectric cooling systems (Tu and Hwang, 2020; Raveesh et al., 2021).

The efficiency of VCR systems strongly depends on the climatic conditions, having their optimum at an ambient temperature of 35 °C and a relative humidity (RH) of 95 % according to previous studies (Patel et al., 2020). At lower temperatures and relative humidities, their energy consumption increases, and their operation time and the WHR decreases (Raveesh et al., 2021; Zolfagharkhani et al., 2018; Nikkhah et al., 2023). Additionally, in cold climates the risk of frosting increases, as the dew point temperature approaches 4 °C, which degrades the system's performance (Raveesh et al., 2023; Cendoya et al., 2023).

Most of the previous scientific studies based on single-stage VCR systems used in Atmospheric Water Generators (AWGs) use refrigerant R134a, while commercial units use refrigerant R410A as working fluid.

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Nomenclature

<i>A</i>	Area, m ² / Parameter (Friedel equation)
$\dot{C}$	Heat capacity rate, W·K ⁻¹
<i>c_p</i>	Specific heat, J·kg ⁻¹ ·K ⁻¹
<i>C</i>	Velocity, m s ⁻¹
<i>D</i>	Diameter, m
<i>E</i>	Energy consumption, kW·h
<i>e</i>	Thickness, m
<i>Err</i>	Error, %
<i>f</i>	Friction factor, -
<i>G</i>	Mass velocity, kg s ⁻¹ m ⁻²
<i>Gz</i>	Graetz number, -
<i>g</i>	Gravitational acceleration, m s ⁻²
<i>h</i>	Specific enthalpy, J·kg ⁻¹
<i>h_{cv}</i>	Convective heat transfer coefficient, W·m ⁻² ·K ⁻¹
<i>j</i>	Colburn factor, -
<i>JP</i>	Correlating parameter, -
<i>K</i>	Constant / Coefficient
<i>k</i>	Thermal conductivity, W·m ⁻¹ ·K ⁻¹
<i>L</i>	Length, m
$\dot{M}$	Mass flow rate, kg·s ⁻¹
<i>M</i>	Mass, kg
<i>M_l</i>	Emissions of CO ₂ per harvested water, kg CO ₂ eq·L ⁻¹
<i>N</i>	Number / Rotational speed, s ⁻¹
<i>n</i>	System operating life, years
<i>NTU</i>	Number of Transfer Units, -
<i>Nu</i>	Nusselt number, -
<i>P</i>	Pressure, Pa / Pitch, m
<i>Pr</i>	Prandtl number, -
$\dot{Q}$	Heat flow, W
<i>r</i>	Ratio, -
<i>R</i>	Thermal resistance, W ⁻¹ ·m·K
<i>Re</i>	Reynolds number, -
<i>RH</i>	Relative Humidity, %
<i>t</i>	Temperature, °C
<i>T</i>	Torque, N·m
<i>U</i>	Overall heat transfer coefficient, W·m ⁻² ·K ⁻¹
$\dot{V}$	Volumetric flow rate, m ³ ·s ⁻¹
<i>v</i>	Specific volume, m ³ ·kg ⁻¹
<i>V</i>	Volume, m ³
$\dot{W}$	Power, W
<i>W</i>	Humidity ratio, kg _w ·kg _{as} ⁻¹
<i>x</i>	Quality, -

Greek symbols

α	Electromechanical loss factor, - / Void fraction, -
β	Indirect emission factor, -
Δ	Difference
δ	Liquid film thickness of annular ring, m
ε	Effectiveness, -
ϕ	Two-phase multiplier, -
η	Efficiency, -
λ	Recycling factor, -
μ	Dynamic viscosity, N s m ⁻²
ν	Kinematic viscosity, m ² s ⁻¹
ρ	Density, kg·m ⁻³
σ	Surface tension, N m ⁻¹ / Free flow area over frontal area, -
ω	Angular speed, rad·s ⁻¹ / Heat capacity ratio, -
ψ	Heat transfer enhancement factor, -

Subscripts

<i>a</i>	Air
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<i>ad</i>	Adapted
<i>amb</i>	Ambient
<i>av</i>	Average
<i>c</i>	Contact
<i>cat</i>	Catalogue
<i>cd</i>	Condenser
<i>charge</i>	Charge
<i>cp</i>	Compressor
<i>crit</i>	Critical
<i>db</i>	Dry bulb
<i>ex</i>	Exhaust
<i>ev</i>	Evaporator
<i>f</i>	Frictional
<i>fa</i>	Fictitious air
<i>fan</i>	Axial fan
<i>fr</i>	Frontal
<i>g</i>	Gas (saturated state)
<i>go</i>	Gas only
<i>h</i>	Homogeneous
<i>i</i>	Inside
<i>if</i>	Interfacial
<i>in</i>	Internal
<i>L</i>	Longitudinal
<i>l</i>	Liquid (Saturated state)
<i>lo</i>	Liquid only
<i>loss</i>	Loss
<i>loss0</i>	Loss without charge
<i>m</i>	Momentum
<i>max</i>	Maximum
<i>min</i>	Minimum
<i>o</i>	Outside
<i>p</i>	Pressure
<i>r</i>	Refrigerant
<i>ra</i>	Rouhani-Axelsson (void fraction)
<i>ref</i>	Reference
<i>s</i>	Isentropic / Swept / Singular
<i>sc</i>	Sub-cooling
<i>su</i>	Supply
<i>su1</i>	Supply after heating-up
<i>T</i>	Transverse
<i>t</i>	Tube
<i>tp</i>	Two-Phase
<i>U</i>	U-bend
<i>v</i>	Volume
<i>w</i>	Water
<i>wall</i>	Wall
<i>wb</i>	Wet bulb
<i>wet</i>	Wet
<i>zi</i>	Zone “i”

Abbreviation

<i>AWG</i>	Atmospheric Water Generators
<i>AWH</i>	Atmospheric Water Harvesting
<i>COP</i>	Coefficient of Performance, -
<i>GWP</i>	Global Warming Potential
<i>HC</i>	Hydrocarbon
<i>HFC</i>	Hydrofluorocarbon
<i>HFO</i>	Hydrofluoroolefins
<i>SEC</i>	Specific Energy Consumption, kW·h·L ⁻¹
<i>TEWI</i>	Total Equivalent Warming Impact, CO ₂ eq·year ⁻¹
<i>VCC</i>	Volumetric Cooling Capacity, MJ·m ⁻³
<i>VCR</i>	Vapour Compression Refrigeration
<i>WHR</i>	Water Harvesting Rate, L·h ⁻¹

Zolfagharkhani et al. used refrigerant R134a in a numerical study developed under the climate conditions of cities located in the Iranian coast. Their system allowed to collect between 22 and 26 L day⁻¹ with Specific Energy Consumptions (SECs) varying between 0.22 kWh L⁻¹ and 0.3 kWh L⁻¹. Cattani et al. (Cattani et al., 2023) also considered refrigerant R134a for a large size AWG, collecting between 1229 L day⁻¹ and 1777 L day⁻¹, with SECs ranging from 0.628 L day⁻¹ to 0.476 kWh L⁻¹ for ambient temperatures and relative humidities varying between 19.6 °C and 21.7 °C and 53.6 % and 74.2 %, respectively. Raveesh et al. (Raveesh et al., 2023) evaluated a VCR system using refrigerant R134a under the climate conditions of 30 Indian cities, whose results were presented as monthly average water generation map for each city. They obtained low harvesting rates and a high number of non-operation hours during winter. Concerning the SEC, for the favourable cities, they obtain average annual SEC of the order of 0.6 kWh L⁻¹, in a range that varies between 0.514 kWh L⁻¹ and 0.950 kWh L⁻¹. According to these authors, SECs higher than 1 kWh L⁻¹ are not economically rentable for this kind of technology. Patel et al. carried out an experimental study of a single VCR system using refrigerant R22 under seven different climate conditions. Further studies, such as Cendoya et al. (Cendoya et al., 2023) combine VCR systems, using refrigerant R290, with desiccant wheels and selective membranes to improve the performance of the VCR system in dry climates. The energy input for these systems is mainly electricity; however, it is also possible to find in the literature VCR systems powered by gas-engine, such as the study developed by Ozkan et al. (Ozkan et al., 2017).

In the case of commercial AWGs, most manufacturers use single-stage VCR systems in their equipment. Some manufacturers, such as Watergen (Watergen, 2025), Veragon (Veragon, 2025), Rainoflife (RainofLife, 2025) and Tsunami (Tsunami Products, 2025), use refrigerants R410A, R410C and R407C as working fluids. Other commercially available systems employ refrigerants R22 (Atlantis Solar and Wind, 2025), R134a (Hendrx Water, 2025; Airowater, 2025; Fujian Wanjuan Technology Co, 2025), R417A (Aeronero, 2025) and R407C (Vayuneer, 2025). In the case performance characteristics, Watergen specifies their AWGs at an ambient temperature of 26.6 °C and a relative humidity of 60 %. On the other hand, Rainoflife use a nominal condition of 28 °C and 80 % of relative humidity, and Tsunami 26.7 °C and 80 % of relative humidity. Manufacturers do not use more sophisticated VCR configuration, such as booster configuration used in systems using refrigerant R744 (Sacacas et al., 2022) or cascade configurations (Faruque et al., 2022; Faruque et al., 2022), because of economic limitations, complexity of the systems and a more important reason related with technical constraints: unrealistic very low pressure ratio on the compressors.

Technological advances have allowed the creation of the fourth-generation refrigerants, reducing their Global Warming Potential (GWP), shifting the refrigeration industry from HFC (hydrofluorocarbon) to HFO (Hydrofluoroolefins) to comply with environmental policies and global conventions (Ciconkov, 2018). Another important decision of the refrigeration industry is the reconsideration of natural refrigerants.

HFC refrigerants have zero Ozone Depletion Potential but still have a relatively high GWP. Therefore, they are set to be replaced in the next decade by HFO refrigerants, because they offer the best trade-off of environmental properties and performance. Natural refrigerants also are a suitable option due to their low GWP; however, they present some restrictions. For example, the flammability of the refrigerant R290 (propane) limits the size of the system to refrigerant charges lower or equal to 500 g (Colbourne and Suen, 2021; Jones et al., 2022; Colbourne et al., 2020). The refrigerant R744 (carbon dioxide) presents excellent GWP but bad performances in warm climates, where it operates at transcritical conditions (Sacacas et al., 2022). Finally, the use of refrigerant R717 (ammonia) is also restricted due to its toxicity (ASHRAE Handbook, 2022). From the state-of-the-art, it is possible to conclude that there are several studies using only one refrigerant under limited

climate conditions and that the AWG manufacturers use different refrigerants but specifying their equipment to only one operating condition. No previous studies have considered the used of HFO refrigerants and few studies have considered natural refrigerants. Thus, this study aims to evaluate simultaneously the effect of climate conditions and different refrigerants on the performance of AWGs. Thus, more specifically, the scope of this paper is to evaluate the performance of a given AWG considering five refrigerants (current reference HFC refrigerants R134a, R407C, R410A and future refrigerants R1234yf, R290) and seven different climate conditions of seven cities (Beijing, Cairo, Harare, Kabul, New Delhi, Sao Paulo, and Santiago (Chile)). In a first step, the system is sized for a nominal condition of a dry bulb temperature of 20 °C and a relative humidity of 60 %, to collect higher than 2.7 L h⁻¹, among other considerations. These conditions are used to size the evaporator and the condenser, and to select the compressor. Then, the system is modelled through semi-empirical models including a compressor, a condenser, an evaporator and an axial fan. Each individual model is validated using catalogue data. After that, the individual models are interconnected to develop the overall AWG model. Then, the system is evaluated through two analyses: 1) a parametric analysis resulting in a performance map for each refrigerant, and 2) an annual simulation using the climatic conditions of the seven cities, resulting in recommendations for the designers concerning the effect of the refrigerant, climate conditions and operating conditions on the AWGs performance and environment impact. The main performance results obtained from these analyses are the Water Harvesting Rate, the Specific Energy Consumption, the Coefficient Of Performance and the equivalent CO₂ emissions. The methodology applied in this study is summarised in Fig. 1, which illustrates the integration process adopted in this research.

2. Models development

2.1. System description

Fig. 2 shows a schematic of the reference system, with the atmospheric air firstly circulating through the evaporator where it is cooled below its dew point temperature to condense the water contained in the airstream. Then, the air circulates through the condenser, where it is heated-up before being released into the environment through the axial fan.

2.2. Refrigerants

The system is designed by considering 5 refrigerants: 3 HFCs, 1 HFO and 1 natural refrigerant. Table 1 presents relevant thermodynamic information about these refrigerants, as their critical pressure and temperature, Global Warming Potentials and security classification. Although refrigerants R134a, R407C and R410A are being phased-out, they are included in this analysis because of their high use in commercial AWGs and in order to give a good baseline for the AWG designers, who should select a future or actual refrigerant having similar thermophysical properties to the one that give the best performance. In the case of the refrigerants R290 and R1234yf, there is no problem of phasing-out, but they are flammable, which also impose some limitations in the size of the system.

2.3. Ambient conditions

The ambient conditions used to evaluate the performance of the AWG consider two different approaches: a) a set of combinations of dry bulb temperatures and relative humidities, and b) climate conditions of seven drought-affected cities.

a) Set of combination of ambient conditions. In this analysis, the system is evaluated using a combination of dry bulb temperatures and relative humidities varying from 0 °C to 40 °C with steps of 1 °C and from 5 % to 100 % with steps of 5 %, respectively, for an atmospheric

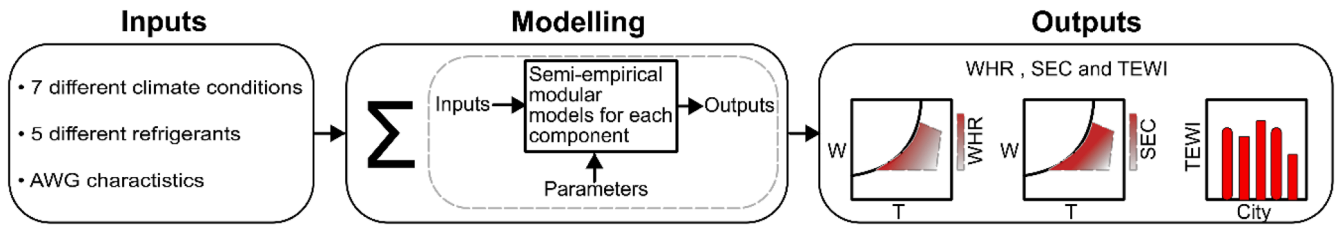


Fig. 1. Methodology used in this research. It considers 7 different climate conditions, 5 different refrigerants and different AWGs characteristics, which are mathematically modelled to determine the *WHR*, the *SEC* and the *TEWI* of the different combinations.

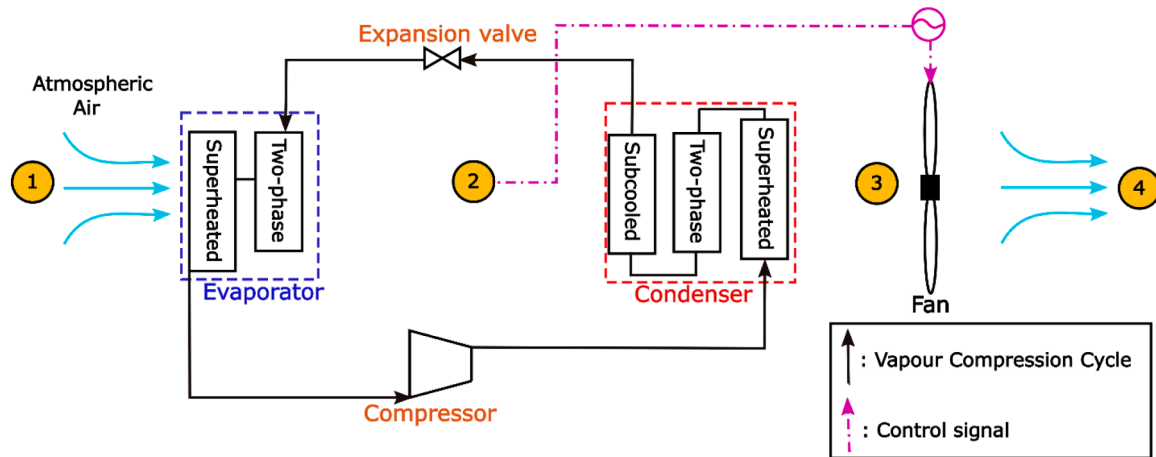


Fig. 2. Configuration of the VCR system. It includes a compressor, a condenser, an expansion valve, an evaporator and an axial fan. The ambient air circulates in series, first through the evaporator where the atmospheric water condenses, then through the condenser to condense the refrigerant of the VCR system and finally through the axial fan, being delivered to the ambient. The control system only control the fan speed to modulate the air mass flow rate circulating through the system to maintain a constant temperature of 4 °C at the evaporator exhaust.

Table 1

Selected refrigerants and their respective thermodynamic, environmental and security characteristics.

Refrigerant	R134a	R290	R407C	R410A	R1234yf
Type	HFC	HC	HFC	HFC	HFO
P_{crit} [bar]	40.6	42.5	46.3	49.0	33.8
t_{crit} [K]	374.2	369.9	359.4	344.5	367.9
GWP 100 years	1430	20	1800	2100	4
Security Classification	A1	A3	A1	A1	A2L

pressure of 1 atm.

The range of temperatures and relative humidities were chosen to have a wide combination of climate conditions. The ambient temperature was limited to values higher than 0 °C to avoid any problem of frosting on the evaporator surface, but in reality it was fixed to a minimum of 4 °C because of the condition imposed in the control system, imposing an air temperature at the evaporator exhaust of 4 °C to avoid frosting on the evaporator surface. On the other hand, high temperatures, up to 40 °C, were chosen to analyse the performance of the VCR system at high ambient temperatures, because some refrigerants under those conditions reach the critical pressure. Concerning the relative humidity, the lower range was chosen in order to evaluate the performance of AWGs working with high sensible loads in the evaporator. On the contrary, the higher range was chosen to evaluate the performance of the AWG under high moist content and high latent loads in the evaporator.

b) Climate conditions of drought-affected cities. Drought-affected cities, such as Beijing (Zhao et al., 2017), Cairo (Mohamed, 2019), Harare (Manzungu and Chioreso, 2012; Musemwa, 2021), Kabul (UNICEF, 2022; Akhtar and Shah, 2019), New Delhi (Dubey and Mishra,

2021; Bhattacharyya and Prasad, 2020), Sao Paulo (Lazaro et al., 2023), and Santiago (Chile) (Gobierno de Chile, 2022) are selected to evaluate the annual performance of the atmospheric water generator. Their climate conditions are obtained from Meteonorm (Meteonorm Version 8, 2025) for a Typical Meteorological Year with a time step of 1 hour. Figs. 3a and 3b present the average monthly dry bulb temperatures (t_{db}) and relative humidities (RH) respectively, whereas Figs. 3c and 3d show the cumulative bin hours for the dry bulb temperature and the relative humidity respectively. Sao Paulo is expected to have the best water harvesting rates due to its higher relative humidities and intermediate temperatures. On the other hand, Harare and Sao Paulo have similar temperature distributions, but Harare has lower relative humidities, and therefore it is expected to have lower water harvesting rates. By the same logic, Kabul is the worst city, due to its lower temperatures and relative humidities. Similarly, Beijing has temperatures lower than 5 °C for almost half of the total hours of a year, which means that an atmospheric water generator would not be able to operate during several hours in a year due to the risk of frosting on the evaporator. Finally, New Delhi is one of the driest and hottest cities during summer, which also could affect to the atmospheric water generators, because it causes higher condensing pressures (even above the refrigerant critical pressure) and higher compressor input powers.

Table 2 presents the Köppen classification of each city and a brief description of the weather conditions, to complement the information obtained from the weather data.

2.4. System design

The system could be designed in different ways, for example, by targeting the same amount of *WHR*, the same *SEC*, the same compressor pressure ratio, a given compressor input power or the same air mass flow

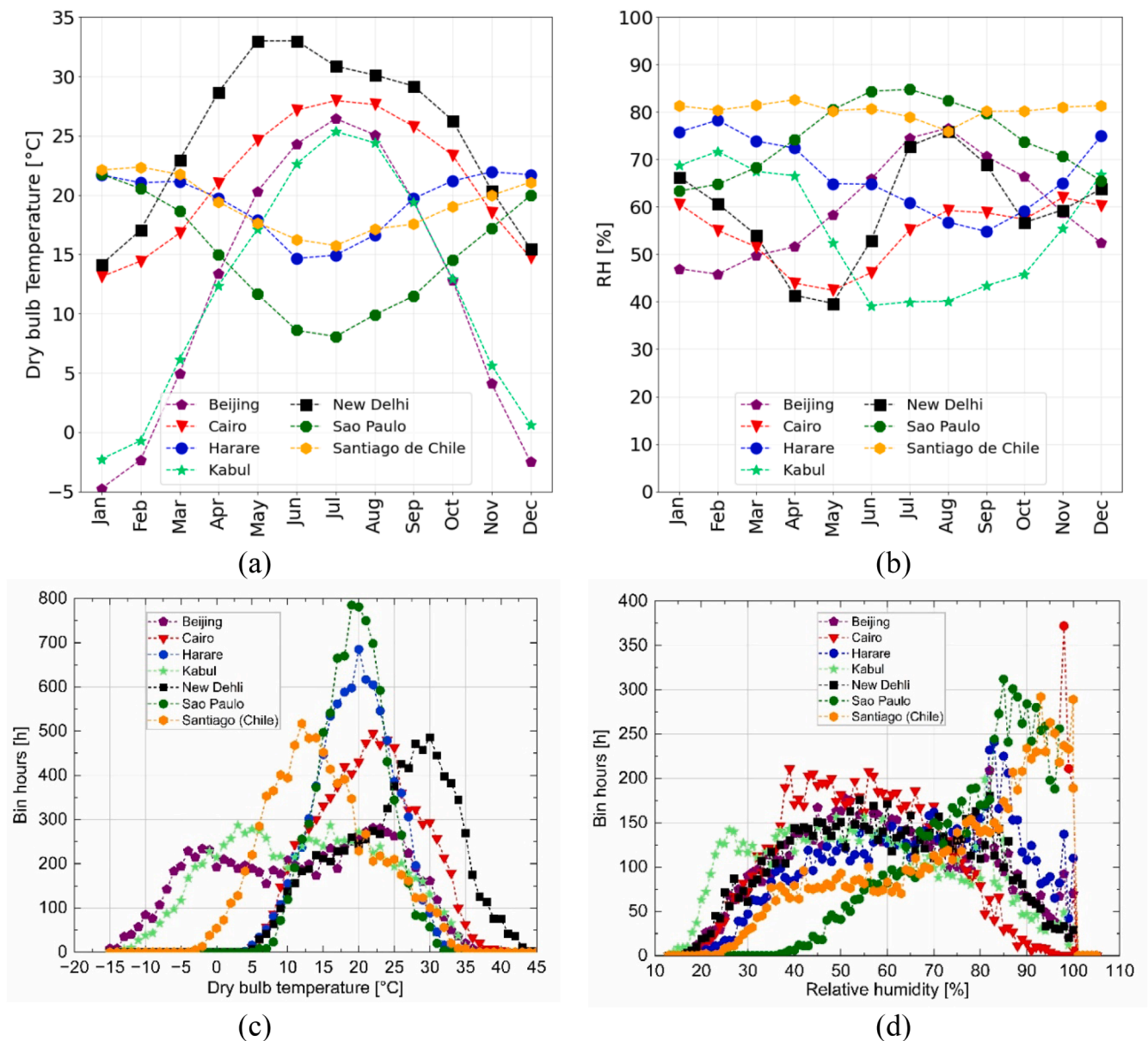


Fig. 3. (a) Monthly average dry bulb temperature and (b) monthly average relative humidity for the seven evaluated cities. Bin hours for (c) dry bulb temperature and (d) relative humidity. Bin hours presents the cumulative frequency of the dry bulb temperature and relative humidity for the seven cities considered in this study.

rate. In this study, the system is designed by considering the same compressor cooling capacity and the same heat exchanger geometries, and different refrigerants and climate conditions.

Regarding the compressors, fixed-speed scroll compressors for air conditioning applications are considered due to their higher efficiency and lower noise emissions. Compressors are selected from the Copeland selection software (Copeland, 2025) using a condensing temperature of 40 °C, an evaporating temperature of 0 °C, a superheating of 10 K, a subcooling of 10 K, a cooling power of 5 kW and a 3-phase/380–420 V (50 Hz) compressor motor. Table 3 summarizes the selected compressors for each refrigerant.

From these refrigerants, R134a, R407C and R410A are phasing-out refrigerants due to their high GWP. On the other hand, refrigerant R1234yf, is a HFO refrigerant with a very low GWP, positioning it as a promising substitute of refrigerant R134a. However, it is a mildly flammable refrigerant, which limits the refrigerant charge to 1.2 kg according to IEC 60,335–2–89 (IEC 60335-2-89, 2019). In the case of R290, it is a natural refrigerant with a low GWP, but highly flammable.

Its flammability limits its applications to refrigeration systems having refrigerant charges lower than 500 g (IEC 60335-2-89, 2019).

According to the results, refrigerant R410A requires the smallest compressor, while refrigerants R134a and R1234yf the biggest ones.

The condenser and the evaporator are sized with the help of the model described in Section 2.6, at a dry bulb temperature of 20 °C, a relative humidity of 60 % and an atmospheric pressure of 1 atm. The targets used to size these heat exchangers are: a water harvesting rate higher than 2.7 L h⁻¹, a system SEC lower than 0.45 kWh L⁻¹, a compressor pressure ratio 2.0 and 2.5, a system COP higher than 4.5 and an electric input power lower than 1.2 kW. The results obtained for the heat exchanger geometries are shared in all systems, in order to only reflect the effect of the different compressors and refrigerants on the performance of the atmospheric water generator under different ambient conditions. Table 4 presents the results obtained for the heat exchanger geometries.

Table 5 summarizes the results obtained for the overall system under the design conditions with the selected compressors and the size of the

Table 2

Köppen classification and brief description of the weather conditions of each city.

City	Köppen classification	Brief description of weather conditions
Beijing	Dwa	Humid continental climate with hot, humid summers and cold, dry winters
Cairo	BWh	Hot desert climate: high temperatures, long hot summers, mild winters with very little rainfall, and overall arid conditions due to stable descending air and high pressure.
Harare	Cwb	Subtropical highland climate, characterized by mild summers, dry winters, and moderate seasonality due to its high altitude on a plateau.
Kabul	BSk	Cold semi-arid (steppe) climate with a warm, dry summer and cold winters.
New Delhi	BSh/Cwa	Hot semi-arid climate (BSh) bordering on a humid subtropical climate with a dry winter (Cwa), characterized by significant variations between hot, rainy summers and cool, dry winters.
Sao Paulo	Cwa	Humid subtropical climate, warm-temperate climate with humid summers and a distinct dry winter season.
Santiago (Chile)	Csb	Warm-summer Mediterranean climate, warm, dry summers and mild, wet winters.

Table 3

Characteristics of the selected scroll compressors: compressor model, compressor swept volume and compressor cooling power at the selected conditions.

Refrigerant	R134a	R290	R407C	R410A	R1234yf
Compressor model	ZR34K3E	ZH04KCU	ZR22K3E	ZP24K5E	YH06K1E
$\dot{V}_s$ [m ³ h ⁻¹]	8.03	5.76	5.34	3.96	8.00
$\dot{Q}_{ev}$ [kW]	4.94	4.81	4.97	5.42	4.82

Table 4

Dimensions of the condenser and evaporator of the VCR systems, with N_{rows} the number of tube rows, $N_{circuits}$ the number of refrigerant circuits, $N_{tubes,rows}$ the number of tubes per row, N_{tubes} the total number of tubes, P_{fin} the fin pitch, e_{fin} the fin thickness, A_a the air side heat transfer area, D_i the internal diameter of the tubes, e_t the tube thickness, S_L the longitudinal pitch and S_T the transversal pitch.

Variable	Condenser	Evaporator
N_{rows}	3	4
$N_{circuits}$	3	3
$N_{tubes,rows}$	21	21
N_{tubes}	63	84
P_{fin} [mm]	3	3.2
e_{fin} [mm]	0.3	0.3
A_a [m ²]	28.4	43.4
D_i [mm]	7.5	7.5
e_t [mm]	1	1
S_L [mm]	26	28
S_T [mm]	30	32
Height [m]	0.630	0.672
Width [m]	0.90	0.96
Depth [m]	0.078	0.112

heat exchangers presented in Table 4. Every system meets the constraints imposed in the system design with refrigerant R410A having the highest cooling power per compressor size (Volumetric Cooling Capacity, VCC), followed by the refrigerant R407C.

2.5. Control strategy

An ideal control is considered for the atmospheric water generator. It

Table 5

Design conditions of the VCR system for each evaluated refrigerant, with $\dot{Q}_{ev}$ the evaporator power, $\dot{W}_{cp}$ the compressor power, r_p the compressor pressure ratio, WHR the Water Harvesting Rate, COP the Coefficient Of Performance, $\dot{V}_a$ the volumetric air flow rate and VCC the Volumetric Cooling Capacity.

Refrigerant	R134a	R290	R407C	R410A	R1234yf
$\dot{Q}_{ev}$ [kW]	5.88	5.52	7.34	6.33	5.62
$\dot{W}_{cp}$ [kW]	1.09	1.08	1.01	1.17	1.20
r_p [-]	2.50	2.18	2.07	2.22	2.41
WHR [L h ⁻¹]	2.97	2.78	3.72	3.21	2.84
COP [-]	5.40	5.10	7.15	5.40	4.68
$\dot{V}_a$ [m ³ h ⁻¹]	779	732	972	834	741
VCC [MJ m ⁻³]	2.64	3.45	4.95	5.76	2.53

controls the air temperature at the evaporator exhaust, which is fixed at a constant value of 4 °C. It is achieved by controlling the axial fan speed, which allows to modify the air mass flow rate circulating through the atmospheric water generator. The ideal control also stops the system if the ambient dew point temperature is lower than 4 °C, in order to avoid frosting formation on the evaporator surface. At dew point temperatures higher than 4 °C there is no risk of frosting, thus the system is allowed to work. The only limitation appears at high dry bulb temperatures, when the refrigerant reaches the critical pressure. Under these conditions, the systems is turned off. The magnitude of this temperature depends on the refrigerant critical temperature and on the system size. The compressor is not controlled, it works at a fixed speed and the electronic expansion valve operates independently, maintaining a constant evaporator over-heating at any working condition.

2.6. Components modelling

The system components are modelled with semi-empirical models and considering a modular approach. Each individual model is then interconnected to define the overall system model. In the next sections, the equations for each model are described.

2.6.1. Scroll compressor

The scroll compressor is modelled using the methodology suggested by Winandy et al. (Winandy et al., 2002; Cuevas et al., 2010). This model considers a refrigerant supply heating-up ($\dot{Q}_{r,su,cp}$), a successive adiabatic isentropic and isochoric compression, electromechanical losses ($\dot{W}_{loss,cp}$), and an exhaust cooling-down ($\dot{Q}_{r,ex,cp}$). The refrigerant supply heating-up is determined with Eq. (1), as a result of the heat transfer with a fictitious isothermal wall representing the compressor crankcase. This heat transfer is modelled with the ϵ -NTU method presented in Eqs. (2) and 3, assuming that the fictitious wall has an infinite thermal capacitance.

$$\dot{Q}_{r,su,cp} = \dot{M}_{r,cp} \cdot (h_{r,su1,cp} - h_{r,su,cp}) = \epsilon_{su,cp} \cdot \dot{C}_{r,su,cp} \cdot (t_{wall,cp} - t_{r,su,cp}) \quad (1)$$

In this equation $\dot{M}_{r,cp}$ is the refrigerant mass flow rate supplied by the compressor, $h_{r,su1,cp}$ and $h_{r,su,cp}$ are the refrigerant specific enthalpies before the heating-up su and after the heating-up $su1$, $\epsilon_{su,cp}$ is the heat exchanger effectiveness, $\dot{C}_{r,su,cp}$ is the refrigerant heat capacity rate, and $t_{wall,cp}$ and $t_{r,su,cp}$ are the temperatures of the fictitious wall and the refrigerant at the supply conditions.

The effectiveness is determined in function of the Number of Transfer Units $NTU_{su,cp}$, as described in Eq. (2).

$$\epsilon_{su,cp} = 1 - \exp(-NTU_{su,cp}) \quad (2)$$

$$NTU_{su,cp} = UA_{ref,su,cp} \cdot \frac{\left(\frac{\dot{M}_{r,cp}}{\dot{M}_{r,ref,cp}}\right)^{0.8}}{\dot{C}_{r,su,cp}} \quad (3)$$

In these equations $UA_{ref,su,cp}$ and $\dot{M}_{r,ref,cp}$ are respectively the overall

heat transfer coefficient and the refrigerant mass flow rate at the reference conditions, both parameters of the model.

The compressor-motor electromechanical losses are modelled with Eq. (4). The first term represents the compressor electromechanical losses under unloaded running conditions and the second term the electromechanical losses under loaded running conditions.

$$\dot{W}_{\text{loss,cp}} = T_{\text{loss0}} \cdot \omega_{\text{cp}} + \alpha \cdot \dot{W}_{\text{in,cp}} \quad (4)$$

where T_{loss0} and α parameters of the electromechanical loss model, ω_{cp} is the compressor angular speed and $\dot{W}_{\text{in,cp}}$ is the internal compressor power.

The internal compression is split into two processes: one reversible adiabatic (isentropic) and other one isochoric adiabatic, described in Eq. (5).

$$\dot{W}_{\text{in,cp}} = \dot{M}_{r,\text{cp}} \cdot [(h_{r,\text{ad,cp}} - h_{r,\text{su1,cp}}) + v_{\text{ad,cp}} \cdot (P_{r,\text{ex,cp}} - P_{r,\text{ad,cp}})] \quad (5)$$

where $h_{r,\text{ad,cp}}$, $v_{\text{ad,cp}}$ and $P_{r,\text{ad,cp}}$ are the specific enthalpy, the specific volume and the refrigerant pressure at the intermediate conditions, $h_{r,\text{su1,cp}}$ is the specific enthalpy of the refrigerant at the compression chamber supplied conditions and $P_{r,\text{ex,cp}}$ is the refrigerant pressure at the exhaust conditions.

The refrigerant intermediate specific volume $v_{\text{ad,cp}}$ depends on the compressor built-in volume ratio $r_{v,\text{in,cp}}$ and the specific volume of the refrigerant supplied to the compression chamber $v_{r,\text{su1,cp}}$, as shown in Eq. (6).

$$r_{v,\text{in,cp}} = \frac{v_{r,\text{su1,cp}}}{v_{r,\text{ad,cp}}} \quad (6)$$

The compressor input power $\dot{W}_{\text{cp}}$ is obtained by summing the electromechanical losses and the internal compression power, as presented in Eq. (7).

$$\dot{W}_{\text{cp}} = \dot{W}_{\text{in,cp}} + \dot{W}_{\text{loss,cp}} \quad (7)$$

The ambient losses $\dot{Q}_{\text{cp,amb}}$ are determined with Eq. (8), and the energy balance of the fictitious wall is defined by Eq. (9).

$$\dot{Q}_{\text{cp,amb}} = UA_{\text{amb}} \cdot (t_{\text{wall,cp}} - t_{\text{amb}}) \quad (8)$$

In this equation UA_{amb} is the compressor to ambient overall heat transfer coefficient, a parameter of the model, and t_{amb} is the ambient temperature.

$$\dot{W}_{\text{loss,cp}} - \dot{Q}_{r,\text{ex,cp}} + \dot{Q}_{r,\text{su,cp}} - \dot{Q}_{\text{cp,amb}} = 0 \quad (9)$$

The refrigerant mass flow rate $\dot{M}_{r,\text{cp}}$ trapped by the compressor is determined with Eq. (10), which depends on the compressor speed N_{cp} , the compressor swept volume $V_{s,\text{cp}}$ and the refrigerant specific volume $v_{r,\text{su1,cp}}$.

$$\dot{M}_{r,\text{cp}} = \frac{N_{\text{cp}} \cdot V_{s,\text{cp}}}{v_{r,\text{su1,cp}}} \quad (10)$$

2.6.2. Condenser

The condenser is modelled with a three-zones model: superheated vapour, two-phase, and subcooled liquid zone, in agreement with the methodology proposed by Cuevas et al. (Cuevas et al., 2009). The model considers the energy balances on both air and refrigerant sides for each zone, as well as the heat transfer between both fluids, as described in Eqs. (11) and 12 respectively.

$$\dot{Q}_{\text{zi,cd}} = \dot{M}_{r,\text{cd}} \cdot (h_{r,\text{su,zi,cd}} - h_{r,\text{ex,zi,cd}}) = \dot{M}_{a,\text{cd}} \cdot (h_{a,\text{ex,zi,cd}} - h_{a,\text{su,zi,cd}}) \quad (11)$$

$$\dot{Q}_{\text{zi,cd}} = \varepsilon_{\text{zi,cd}} \cdot \dot{C}_{\text{min,zi,cd}} \cdot (t_{r,\text{su,zi,cd}} - t_{a,\text{su,zi,cd}}) \quad (12)$$

With $\dot{Q}_{\text{zi,cd}}$ the heat flow of the zone “i”, h_a and h_r are the specific enthalpies of the air and the refrigerant at the supply *su* and exhaust *ex*

conditions in the zone “i”, $\dot{M}_{a,\text{cd}}$ and $\dot{M}_{r,\text{cd}}$ are the air and the refrigerant mass flow rates, $\varepsilon_{\text{zi,cd}}$ is the heat exchanger effectiveness, $\dot{C}_{\text{min,zi,cd}}$ is the minimum heat capacity rate, and $t_{a,\text{su,zi,cd}}$ and $t_{r,\text{su,zi,cd}}$ are the air and refrigerant temperatures at the condenser supply conditions.

A counterflow approach is considered for each zone. The heat exchanger effectiveness $\varepsilon_{\text{zi,cd}}$ in the superheated and subcooled zones is determined in function of the Number of Transfer Units $NTU_{\text{zi,cd}}$ and the heat capacity ratio $\omega_{r,\text{zi,cd}}$ with Eq. (13), while Eq. (14) is used for the two-phase zone effectiveness, which only depends on the Number of Transfer Units $NTU_{\text{tp,cd}}$.

$$\varepsilon_{\text{zi,cd}} = \frac{1 - e^{(-NTU_{\text{zi,cd}} \cdot (1 - \omega_{r,\text{zi,cd}}))}}{1 - C_{r,\text{zi,cd}} \cdot e^{(-NTU_{\text{zi,cd}} \cdot (1 - \omega_{r,\text{zi,cd}}))}} \quad (13)$$

$$\varepsilon_{\text{tp,cd}} = 1 - e^{(-NTU_{\text{tp,cd}})} \quad (14)$$

The overall heat transfer coefficients $UA_{\text{zi,cd}}$ are determined according to Eq. (15).

$$\frac{1}{UA_{\text{zi,cd}}} = \frac{1}{h_{\text{cv,r,zi,cd}} \cdot A_{r,\text{zi,cd}}} + \frac{\ln\left(\frac{D_{o,\text{cd}}}{D_{i,\text{cd}}}\right)}{2 \cdot \pi \cdot k_{\text{zi,cd}} \cdot L_{\text{zi,cd}}} + \frac{1}{h_{\text{cv,a,zi,cd}} \cdot A_{a,\text{zi,cd}} \cdot \eta_{o,\text{zi,cd}}} \quad (15)$$

The correlations used to determine the corresponding convective heat transfer coefficients on the refrigerant ($h_{\text{cv,r,zi,cd}}$) and on the air ($h_{\text{cv,a,zi,cd}}$) sides for each zone and fluid are explained in detail in Section 2.6.4. The heat transfer areas on the refrigerant ($A_{r,\text{zi,cd}}$) and on the air ($A_{a,\text{zi,cd}}$) sides, as well as the diameters $D_{i,\text{cd}}$ and $D_{o,\text{cd}}$ are determined with the values given in Table 4. $k_{\text{zi,cd}}$ is the thermal conductivity of the heat exchanger tubes and $\eta_{o,\text{zi,cd}}$ the overall surface efficiency of the heat exchanger. Finally, $L_{\text{zi,cd}}$ is the length of the heat exchanger occupied by the zone “i”, which is determined by the model.

The pressure drops on the refrigerant side in each zone considers three contributions: the pressure drop due to the friction ($\Delta P_{f,r,\text{zi,cd}}$), the pressure variation due to the density change (acceleration) ($\Delta P_{m,r,\text{zi,cd}}$) and the singular pressure drop due to the U-bend fittings of the heat exchanger ($\Delta P_{s,r,\text{zi,cd}}$), as described in Eq. (16) for the condenser.

$$\Delta P_{r,\text{zi,cd}} = \Delta P_{f,r,\text{zi,cd}} + \Delta P_{m,r,\text{zi,cd}} + \Delta P_{s,r,\text{zi,cd}} \quad (16)$$

For the superheated and subcooled zones, the frictional pressure drop is determined as follows:

$$\Delta P_{f,r,\text{zi,cd}} = f_{f,r,\text{zi,cd}} \cdot \frac{L_{\text{zi,cd}}}{D_{i,\text{cd}}} \cdot \rho_{r,\text{zi,cd}} \cdot \frac{C_{r,\text{zi,cd}}^2}{2} \quad (17)$$

where $f_{f,r,\text{zi,cd}}$ is the friction factor, $\rho_{r,\text{zi,cd}}$ is the refrigerant density and $C_{r,\text{zi,cd}}$ is the refrigerant velocity.

Again, for the superheated and subcooled zones, the pressure variation due to the density change is determined as:

$$\Delta P_{m,r,\text{zi,cd}} = G_{r,\text{cd}}^2 \cdot \left[\frac{1}{\rho_{r,\text{ex,zi,cd}}} - \frac{1}{\rho_{r,\text{su,zi,cd}}} \right] \quad (18)$$

with $G_{r,\text{cd}}$ the refrigerant mass velocity, ρ_r the refrigerant density at the supply *su* and exhaust *ex* of the zone “i”.

And finally, for the superheated and subcooled zones, the singular pressure drop is determined as:

$$\Delta P_{s,r,\text{zi,cd}} = K_U \cdot N_U \cdot \rho_{r,\text{zi,cd}} \cdot \frac{C_{r,\text{zi,cd}}^2}{2} \quad (19)$$

In Eq. (19), the term K_U is the singular pressure drop coefficient for a U-bend fittings while N_U is the number of U-bend fittings contained in the analyzed zone.

On the other hand, in the two-phase zone, the frictional pressure drop $\Delta P_{f,r,\text{tp,cd}}$ is determined by Eq. (20) (Collier and Thome, 1994).

$$\Delta P_{f,r,\text{tp,cd}} = \phi_{\text{lo}}^2 \cdot \Delta P_{r,\text{lo,tp,cd}} \quad (20)$$

Where the term $\Delta P_{r,lo,tp,cd}$ is the pressure drop assuming only liquid state for the refrigerant, while the term ϕ_{lo}^2 is the two-phase multiplier factor.

The pressure variation due to the density variation $\Delta P_{m,r,tp,cd}$ is determined with Eq. (21) (Collier and Thome, 1994), where α_{su} and α_{ex} are the void fraction, and x_{su} and x_{ex} the refrigerant qualities at the supply su and exhaust ex conditions, and ρ_l and ρ_g are the saturated liquid and saturated vapour refrigerant densities.

$$\Delta P_{m,r,tp,cd} = G_{r,cd}^2 \cdot \left[\left\{ \frac{(1-x_{ex})^2}{\rho_l \cdot (1-\alpha_{ex})} + \frac{x_{ex}^2}{\rho_g \cdot \alpha_{ex}} \right\} - \left\{ \frac{(1-x_{su})^2}{\rho_l \cdot (1-\alpha_{su})} + \frac{x_{su}^2}{\rho_g \cdot \alpha_{su}} \right\} \right] \quad (21)$$

Finally, the singular pressure drop $\Delta P_{s,r,zi,cd}$ is determined with Eq. (22) (Collier and Thome, 1994).

$$\Delta P_{s,r,zi,cd} = \phi_{lo,U}^2 \cdot N_U \cdot \Delta P_{r,s,lo,tp,cd} \quad (22)$$

where $\phi_{lo,U}^2$ is the multiplier factor and $\Delta P_{r,s,lo,tp,cd}$ is the pressure drop assuming that the fluid is circulating entirely as saturated liquid.

In this modelling, the pressure drops in the supply and exhaust collectors are neglected. The correlations used to determine the frictional pressure drop on the refrigerant side and on the air side are further described in Section 2.6.4.

2.6.3. Evaporator

The evaporator is modelled similarly to the condenser, but in this case the mass transfer on the air side is added to consider the water condensation. This phenomenon is modelled by using the approach proposed by Ding et al. (Ding et al., 1990), who assume that the coil works either in a totally wet or a totally dry regime. In both cases, the refrigerant side is discretized into two zones: a superheated and a two-phase zone. On the air side, the behaviour of the heat exchanger in a totally dry regime is modelled in the same way as described for the condenser, but for the totally wet regime the effect of water condensation is included, as described in Eq. (23). In this case a fictitious fluid is defined for the moist air.

$$\dot{Q}_{wet,ev} = \dot{M}_{a,ev} \cdot [(h_{a,su,ev} - h_{a,ex,ev}) - (W_{a,su,ev} - W_{a,ex,ev}) \cdot h_{w,ev}] \quad (23)$$

In this equation $\dot{Q}_{wet,ev}$ is the heat flow in wet regime, $\dot{M}_{a,ev}$ is the air mass flow rate, and h_a and W_a are the air specific enthalpy and humidity at the evaporator ev supply su and exhaust ex conditions, and $h_{w,ev}$ is the specific enthalpy of the liquid condensed water.

Eq. (23) is equalled to Eq. (24) to introduce the heat capacity rate of the fictitious air $\dot{C}_{fa,ev}$, where the driving potential for the heat flow is the air wet bulb temperature t_{wb} at the evaporator ev supply su and exhaust ex conditions, instead of the air enthalpy.

$$\dot{Q}_{wet,ev} = \dot{C}_{fa,ev} \cdot (t_{wb,su,ev} - t_{wb,ex,ev}) \quad (24)$$

The fluid-to-fluid heat transfer is calculated with Eq. (25), where the effectiveness $\epsilon_{wet,ev}$ is determined with Eq. (29).

$$\dot{Q}_{wet,ev} = \epsilon_{wet,ev} \cdot \dot{C}_{min,wet,ev} \cdot (t_{wb,su,ev} - t_{r,ev}) \quad (25)$$

$\dot{C}_{min,wet,ev}$ is the heat capacity rate and $t_{r,ev}$ the evaporating temperature.

In wet regime, the overall heat transfer coefficient ($UA_{wet,ev}$) is determined with Eq. (26). To consider the effect of the mass transfer due to water condensation, the air side thermal resistance is corrected by the specific heat ratio $c_{p,fa,ev}/c_{p,a,ev}$ (Threlkeld, 1970).

$$\frac{1}{UA_{wet,ev}} = \frac{1}{h_{cv,a,ev} \cdot c_{p,fa,ev} / c_{p,a,ev} \cdot A_{a,ev}} + \frac{\ln\left(\frac{D_{o,ev}}{D_{i,ev}}\right)}{2 \cdot \pi \cdot k_{ev} \cdot L_{ev}} + \frac{1}{h_{cv,r,ev} \cdot A_{r,ev}} \quad (26)$$

The convective heat transfer coefficients on the air $h_{cv,a,ev}$ and on the refrigerant $h_{cv,r,ev}$ sides are determined as explained in Section 2.6.4.

$c_{p,fa,ev}$ and $c_{p,a,ev}$ are the specific heat of the fictitious air and of the dry air respectively, and k_{ev} is the thermal conductivity of the heat exchanger tubes. The heat transfer areas on the air $A_{a,ev}$ and on the refrigerant $A_{r,ev}$ sides, the tube L_{ev} and the tube diameters $D_{i,ev}$ and $D_{o,ev}$ are determined from the values given in Table 4.

The air dry-bulb temperature at the evaporator exhaust is determined with the ASHRAE hypothesis (ASHRAE Handbook, 2012), which considers that the air exhaust conditions are the result of the mixture of the air in contact with the evaporator surface (saturated) and the air entering to the coil. The coil surface temperature is here considered as constant. Thus, the air properties are determined with Eqs. (27) to 30.

$$h_{a,ex,ev} = h_{a,su,ev} - \epsilon_{c,wet,ev} \cdot (h_{a,su,ev} - h_{c,wet,ev}) \quad (27)$$

$$W_{a,ex,ev} = W_{a,su,ev} - \epsilon_{c,wet,ev} \cdot (W_{a,su,ev} - W_{c,wet,ev}) \quad (28)$$

$$\epsilon_{c,wet,ev} = 1 - e^{(-NTU_{c,wet,ev})} \quad (29)$$

$$NTU_{c,wet,ev} = \frac{1}{R_{a,ev} \cdot \dot{C}_{a,ev}} \quad (30)$$

In these equations, h_c and W_c are the air specific enthalpy and humidity in the evaporator ev at the contact c temperature, $\epsilon_{c,wet,ev}$ is the contact effectiveness, $NTU_{c,wet,ev}$ is the Number of Transfer Units of contact heat exchange, $R_{a,ev}$ is the thermal resistance on the air side and $\dot{C}_{a,ev}$ is the air heat capacity rate.

The actual working power $\dot{Q}_{ev}$ is determined with the assumption proposed by Braun (Braun et al., 1989), which considers the maximum between the dry $\dot{Q}_{dry,ev}$ and wet $\dot{Q}_{wet,ev}$ regime coil power, as indicated in Eq. (31). According to Braun, this assumption introduces an error lower than 5 %.

$$\dot{Q}_{ev} = \text{MAX}(\dot{Q}_{dry,ev}, \dot{Q}_{wet,ev}) \quad (31)$$

Using this set of equations, it is possible to represent the influence of the evaporator geometry on the amount of condensed water.

2.6.4. Convective heat transfer coefficient and friction factor correlations

a) Refrigerant side.

Convective heat transfer coefficient:

The convective heat transfer coefficients for the refrigerant side in single phase (subcooled and superheated zones) are determined with the Baehr and Stephan (Baehr and Stephan, 2006) correlation for the laminar regime, and the Gnielinski correlation (Gnielinski, 1975) for the turbulent regime (Eqs. (32) and 33, respectively).

$$Nu = \frac{\frac{3.657}{(\tanh(2.264 \cdot Gz^{-1/3}) + 1.7 \cdot Gz^{-2/3})} + \frac{0.0499}{Gz^{-1}} \cdot \tanh(Gz^{-1})}{\tanh(2.432 \cdot Pr^{1/6} \cdot Gz^{-1/6})} \quad (32)$$

$$Nu = \frac{\left(\frac{f}{8}\right) \cdot (Re - 1000) \cdot Pr}{1 + 12.7 \cdot \left(\frac{f}{8}\right)^{1/2} \cdot (Pr^{2/3} - 1)} \quad (33)$$

where Nu is the Nusselt number, Gz is the Graetz number, Pr is the Prandtl number, Re is the Reynolds number and f the friction factor.

In the two-phase zone, the convective heat transfer coefficients are determined with the Thome et al. correlation (Thome et al., 2003) and the Shah correlation (Shah, 1976) for condensation (Eq. (34)) and evaporation (Eq. (35)) respectively. In the case of the Shah correlation (Shah, 1976), the two-phase convective heat transfer coefficient $h_{cv,shah}$ uses the single phase convective heat transfer coefficient $h_{cv,l}$ determined with the Dittus-Boelter correlation and an enhancement factor ψ .

$$Nu_{thome} = 0.003 \cdot Re_{l,thome}^{0.74} \cdot Pr_l^{0.5} \cdot f_{if} \quad (34)$$

$$h_{cv,shah} = \psi \cdot h_{cv,l} \quad (35)$$

The Thome et al. correlation (Thome et al., 2003) uses the Reynolds number $Re_{l,thome}$ (Eq. (36)) and the interface friction factor f_{if} (Eq. (40)), which use the variables and equations presented hereafter in Eqs. (37), 38, 39, 41, 42 and 43.

$$Re_{l,thome} = \frac{4 \cdot G \cdot (1-x) \cdot \delta}{\mu_l \cdot (1-\alpha)} \quad (36)$$

$$\alpha = \frac{\alpha_h - \alpha_{ra}}{\ln\left(\frac{\alpha_h}{\alpha_{ra}}\right)} \quad (37)$$

$$\alpha_h = \left[1 + \left[\frac{(1-x)}{x}\right] \cdot \left(\frac{\rho_g}{\rho_l}\right)\right]^{-1} \quad (38)$$

$$\alpha_{ra} = \frac{x}{\rho_g} \cdot \left\{ \left[1 + 0.12 \cdot (1-x) \cdot \left[\frac{x}{\rho_g} + \frac{1-x}{\rho_l}\right] + \frac{1.18 \cdot (1-x) \cdot [g \cdot \sigma \cdot (\rho_l - \rho_g)]^{0.25}}{G \cdot \rho_l^{0.5}} \right]^{-1} \right\} \quad (39)$$

$$f_{if} = 1 - \left(\frac{C_g}{C_l}\right)^{1/2} \cdot \left[\frac{(\rho_l - \rho_g) \cdot g \cdot \delta^2}{\sigma}\right]^{1/4} \quad (40)$$

$$A_l = (1-\alpha) \cdot A \quad (41)$$

$$C_l = \frac{G \cdot (1-x)}{\rho_l \cdot (1-\alpha)} \quad (42)$$

$$C_g = \frac{G \cdot x}{\rho_g \cdot \alpha} \quad (43)$$

In these equations G is the mass velocity, x is the refrigerant quality, δ is the thickness of the annular liquid during condensation, μ_l is the dynamic viscosity of the saturated liquid, α is the void fraction, α_h is the void fraction determined with the homogeneous model, α_{ra} is the void fraction determined with the Rouhani-Axelsson model, ρ_g and ρ_l are the saturated vapour and saturated liquid refrigerant density respectively, g is the gravitational acceleration, σ is the surface tension, C_g and C_l are the velocity of the vapour and liquid phase respectively, A_l is the cross sectional area occupied by the liquid phase and A is the cross sectional area of the tube.

Pressure drop:

The single phase friction factor is determined as follows for laminar (Eq. (44)) and turbulent (Eq. (45)) regimes:

$$f = \frac{64}{Re} \quad (44)$$

$$f = [0.79 \cdot \ln(Re) - 1.64]^2 \quad (45)$$

For the refrigerant side in the two-phase zone, the frictional pressure drop $\Delta P_{f,tp}$ is determined with the Friedel correlation (Friedel, 1979), which uses a multiplier factor ϕ_{lo}^2 and the liquid only pressure drop ΔP_{lo} , according to Eqs. (46) and 47.

$$\Delta P_{f,tp} = \phi_{lo}^2 \cdot \Delta P_{lo} \quad (46)$$

$$\Delta P_{lo} = f_{lo} \cdot \frac{L}{D_i} \cdot \rho_{lo} \cdot \frac{C_{lo}^2}{2} \quad (47)$$

where f_{lo} , ρ_{lo} and C_{lo} are the friction factor, the density and the velocity by considering that the refrigerant circulates only in liquid phase, L is the length of the tube and D_i its diameter.

The two-phase multiplier factor ϕ_{lo}^2 used in the Friedel correlation (Friedel, 1979) is determined with the following equations.

$$\phi_{lo}^2 = \frac{A_1 + 3.24 \cdot A_2}{Fr_p^{0.045} \cdot We_p^{0.035}} \quad (48)$$

$$A_1 = (1-x)^2 + x^2 \cdot \left(\frac{\rho_l}{\rho_g}\right)^{0.91} \cdot \left(\frac{f_{go}}{f_{lo}}\right) \quad (49)$$

$$A_2 = x^{0.78} \cdot (1-x)^{0.24} \cdot \left(\frac{\rho_l}{\rho_g}\right)^{0.91} \cdot \left(\frac{\mu_g}{\mu_l}\right)^{0.19} \cdot \left(1 - \frac{\mu_g}{\mu_l}\right)^{0.7} \quad (50)$$

$$f_{lo} = \frac{0.079}{Re_{lo}^{0.25}} \quad (51)$$

$$f_{go} = \frac{0.079}{Re_{go}^{0.25}} \quad (52)$$

$$C_{lo} = \frac{G}{\rho_l} \quad (53)$$

Most of these variables were defined previously, except the Froude two-phase Fr_p and Weber two-phase We_p numbers. The subscripts go and lo are used to indicate that the corresponding variables are determined by considering that the refrigerant circulates in vapour only or liquid only phase.

In addition, the following dimensionless numbers are considered in the Friedel correlation:

$$Re_{lo} = \frac{G \cdot D}{\mu_l} \quad (54)$$

$$Re_{go} = \frac{G \cdot D}{\mu_g} \quad (55)$$

$$Fr_p = \frac{G^2}{g \cdot D \cdot \rho_p} \quad (56)$$

$$We_p = \frac{G^2 \cdot D}{\rho_p \cdot \sigma} \quad (57)$$

Where the homogeneous density ρ_p and the mass velocity G are determined as:

$$\rho_p = \left(\frac{x}{\rho_g} + \frac{(1-x)}{\rho_l}\right)^{-1} \quad (58)$$

$$G = \frac{\dot{M}}{N_{circuits} \cdot A} \quad (59)$$

$$A = \frac{\pi}{4} D^2 \quad (60)$$

b) Air side.

The convective heat transfer coefficients on the air side are calculated using the correlation presented by McQuiston (McQuiston et al., 2005), which is based on a 4-row heat exchanger, by considering the Chilton-Colburn j factor, as described in Eqs. (61) to 64.

$$JP = Re_D^{-0.4} \cdot \left(\frac{A}{A_t}\right)^{-0.15} \quad (61)$$

Where JP is a parameter used by McQuiston, Re_D is the Reynolds number determined using the tube outside diameter, A is the total external heat transfer area and A_t is the external area without fins. The parameter JP is used to determine the Colburn factor for a heat exchanger with 4 rows of tubes j_4 .

$$j_4 = 0.2675 \cdot JP + 1.325 \cdot 10^{-6} \quad (62)$$

Then, the Colburn factor j_N for a heat exchanger with N_{rows} is determined with Eq. (63).

$$\frac{j_N}{j_4} = \frac{1 - 1280 \cdot N_{rows} \cdot Re_{S_L}^{-1.2}}{1 - 1280 \cdot 4 \cdot Re_{S_L}^{-1.2}} \quad (63)$$

where Re_{S_L} is the Reynolds number determined with the longitudinal pitch. Finally, the air side convective heat transfer coefficient $h_{cv,a}$ is determined with Eq. (64).

$$h_{cv,a} = \frac{j \cdot G_{max} \cdot c_p}{Pr^{2/3}} \quad (64)$$

Where G_{max} is the mass velocity determined at the minimum free-flow area and c_p is the air specific heat.

The air-side condenser and evaporator pressure drops are determined with Eq. (65), presented by McQuiston (McQuiston et al., 2005).

$$\Delta P = \frac{G_{max}^2}{2 \cdot \rho_{a,su}} \left[(1 + \sigma^2) \cdot \left(\frac{\rho_{a,su}}{\rho_{a,ex}} - 1 \right) + f \cdot \frac{A}{A_{free}} \frac{\rho_{a,su}}{\rho_{a,av}} \right] \quad (65)$$

With ρ_a the air density at the supply su , exhaust ex and average av conditions, σ is the ratio between the minimum free-flow area and the frontal area, and A_{free} is the free-flow area.

2.6.5. Fan

The fan power $\dot{W}_{fan}$ is determined with Eq. (66), by using the air-side pressure drop calculated from the condenser and evaporator models ($\Delta P_{a,ev} + \Delta P_{a,cd}$). An overall fan efficiency $\eta_{overall}$ of 20 % is considered, estimated from an axial fan manufacturer (ZIEHL-ABEGG 2025).

$$\dot{W}_{fan} = \frac{\dot{V}_{a,fan} \cdot (\Delta P_{a,ev} + \Delta P_{a,cd})}{\eta_{overall}} \quad (66)$$

2.6.6. Models validation

Each model is validated by comparing their results with the information provided in the manufacturer catalogues.

In the case of the compressor models, the parameters were identified from the Copeland-Emerson catalogues. Table 6 summarizes the identified parameters, which are obtained regarding an overall error of the models lower than 3 % for the predicted input power and refrigerant mass flow rate.

The condenser model is validated with the models LMC3N15–10, 11, 20, 30, 31, 40 and 41 of the manufacturer LU-VE (LU-VE S.p.A., 2025),

Table 6

Identified parameters of the selected scroll compressors, with α loss factor used to determine the variable electromechanical loss, T_{loss0} is the torque used to determine the electromechanical loss at unloaded conditions, $V_{s,cp}$ is the compressor swept volume, $r_{v,in,cp}$ is the compressor built-in volume ratio, $UA_{ref,su,cp}$, $UA_{ref,ex,cp}$ and UA_{amb} are the overall heat transfer coefficients for the supply, exhaust and environment heat exchange respectively, N_{points} is the number of points used to identify the parameters, and $Err_{\dot{M}_r}$ and $Err_{\dot{W}_{cp}}$ are the errors on the prediction of the refrigerant mass flow rate and the compressor input power respectively.

Refrigerant	R134a	R290	R407C	R410A	R1234yf
Compressor model	ZR34K3E	ZH04KCU	ZR22K3E	ZP24K5E	YH06K1E
α [-]	0.189	0.211	0.146	0.190	0.265
T_{loss0} [kN]	0.589	0.307	0.277	0.279	0.501
$r_{v,in,cp}$ [-]	2.443	3.583	2.518	2.363	3.918
$V_{s,cp}$ [m ³]	$4.92 \cdot 10^{-5}$	$3.40 \cdot 10^{-5}$	$3.46 \cdot 10^{-5}$	$2.43 \cdot 10^{-5}$	$4.75 \cdot 10^{-5}$
$UA_{ref,su,cp}$ [kW K ⁻¹]	3.751	4.750	3.215	4.019	3.476
$UA_{ref,ex,cp}$ [kW K ⁻¹]	0.016	$6.8 \cdot 10^{-3}$	0.021	0.024	0.011
UA_{amb} [kW K ⁻¹]	5.21	4.87	5.03	5.42	5.04
N_{points}	78	97	70	65	98
$Err_{\dot{M}_r}$	1	1.1	1.6	2.4	1
$Err_{\dot{W}_{cp}}$	1.1	1.5	1.7	2.7	2.6

at the conditions defined in the Standard BS-EN-327 (BS EN 327, 2014); and the evaporator model with the model 5x-124-R22 of the McQuay manufacturer (McQuay, 1960), at the conditions defined in the Standard AHRI-410 (AHRI Standard 410, 2001). Table 7 shows the maximum errors obtained for each model in the main outputs.

The errors obtained from the models are considered acceptable. The maximum error is obtained in the condensed water $\dot{M}_{w,ev}$, which is then called Water Harvesting Rate (WHR) in the following sections.

2.6.7. Integrated VCR system model

The previously described sub-models of the scroll compressor, condenser, evaporator and axial fan are integrated into an overall VCR system model. The connections between the sub-models are presented in Fig. 4. In this approach, the expansion valve is not modelled, it is assumed that it is able to maintain a constant overheating at the evaporator exhaust.

The system of equations for the global model is numerically solved using the root-optimized method, available in the SciPy library (Virtanen et al., 2020). This method iterates the condensing and evaporating pressures until the numerical residual of each one is lower than 1×10^{-6} .

3. Results and discussion

The performance of the atmospheric water generators is evaluated through their Coefficient of Performance (COP), Specific Energy Consumption (SEC), and Water Harvesting Rate (WHR), which are calculated using Eqs. (67), (68), and (69), respectively.

$$COP = \frac{\dot{Q}_{ev}}{\dot{W}_{cp} + \dot{W}_{fan}} \quad (67)$$

$$SEC = \frac{\dot{W}_{cp} + \dot{W}_{fan}}{WHR} \quad (68)$$

$$WHR = \dot{M}_{a,ev} \cdot (W_{a,su,ev} - W_{a,ex,ev}) \quad (69)$$

3.1. Parametric analysis

Fig. 5 presents the WHR and the SEC as a heat map on the psychrometric chart for the 5 refrigerants under different climate conditions. Refrigerants R407C and R410A allows to operate in a widest range of climate conditions with a good performance: higher WHRs and lower SECs. It is also observed that the system perform better at high relative humidities and lower ambient temperatures, presenting higher WHRs and SECs.

The behaviour of VCR systems and their trends in energy consumption has been described by many other authors (Patel et al., 2020; Zolfagharkhani et al., 2018; Raveesh et al., 2023; Cendoya et al., 2023), who noted that the highest performance of these systems occurs under high relative humidities (lower SEC and high WHR), which is related with a lower sensible heat that must be removed to start condensing water. According to the obtained results, the best operating conditions

Table 7

Maximum error of each model on the prediction of: Refrigerant mass flow rate $\dot{M}_{r,cp}$, refrigerant exhaust temperature $t_{r,ex,cp}$ and input power $\dot{W}_{cp}$ for the compressor; heat flow $\dot{Q}_{cd}$ and air exhaust temperature $t_{a,ex,cd}$ for the condenser; heat flow $\dot{Q}_{ev}$, air exhaust temperature $t_{a,ex,ev}$ and condensed water $\dot{M}_{w,ev}$ for the evaporator.

Model	Maximum error in output variables
Compressor	$\dot{M}_{r,cp} = \pm 3\%$, $t_{r,ex,cp} = \pm 4$ °C, $\dot{W}_{cp} = \pm 3\%$
Condenser	$\dot{Q}_{cd} = \pm 5\%$, $t_{a,ex,cd} = \pm 0.5$ °C
Evaporator	$\dot{Q}_{ev} = \pm 2.5\%$, $t_{a,ex,ev} = \pm 0.5$ °C, $\dot{M}_{w,ev} = \pm 6\%$

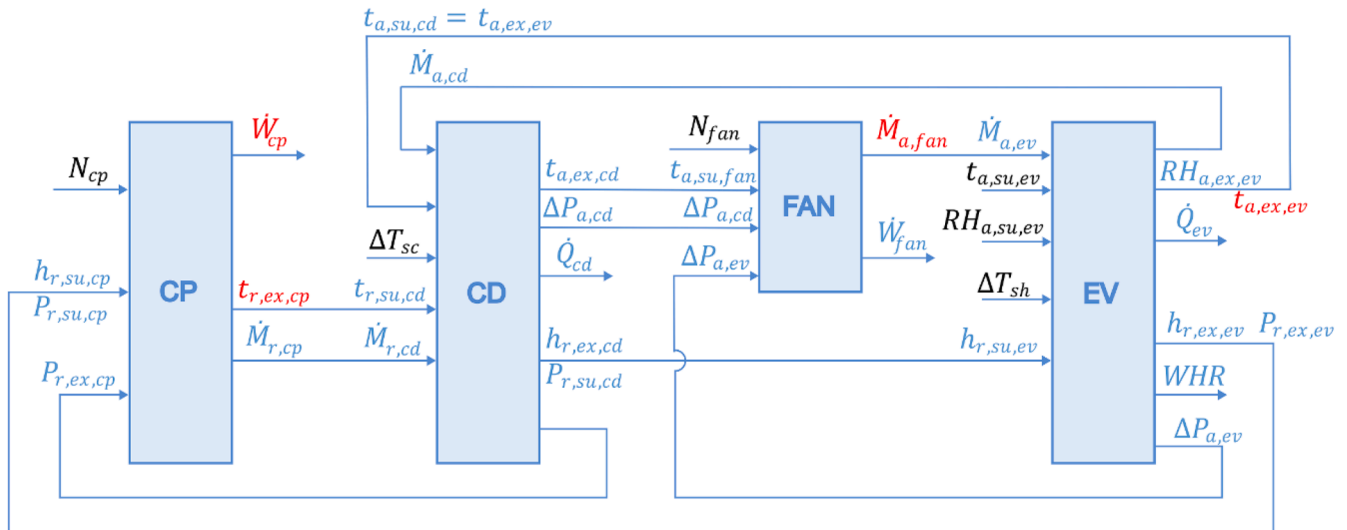


Fig. 4. Integrated VCR system model. The overall model is composed of individual semi-empirical models that are interconnected through the variables indicated in the Figure. This interconnection allows to reduce the number of inputs of the overall model to the ones highlighted in black. Variables highlighted in red must be checked to guarantee a safe working condition of the system.

correspond to medium-low temperatures ($<25\text{ }^{\circ}\text{C}$) with medium-high relative humidities ($>50\%$). In contrast, at high relative humidities ($>80\%$) and high temperatures ($>25\text{ }^{\circ}\text{C}$), the SEC increases due to the control system, which reduces the airflow rate circulating through the system to target a temperature of $4\text{ }^{\circ}\text{C}$ at the evaporator exhaust.

The climate conditions where the atmospheric water generator works efficiently are quite similar for all refrigerants. Fig. 6 presents the behaviour of the refrigerants under three combinations of ambient temperatures and relative humidities: $10\text{ }^{\circ}\text{C} / 70\%$ to 100% , $20\text{ }^{\circ}\text{C} / 30\%$ to 100% and $30\text{ }^{\circ}\text{C} / 20\%$ to 100% . It is observed that all the refrigerants follow similar trends.

Due to the control strategy of the AWG, that maintain a constant air temperature of $4\text{ }^{\circ}\text{C}$ at the evaporator exhaust, when the ambient temperature increases the air mass flow rate circulating through the system decreases. On the other hand, when the relative humidity increases there are two performance variables that go in opposite direction: the potential of water in the air increases as the relative humidity increases and the air mass flow rate decreases as relative humidity increases. Thus, for $10\text{ }^{\circ}\text{C}$ the system presents a higher WHR because it works at higher air mass flow rates. Then, for $20\text{ }^{\circ}\text{C}$ the trend is similar with lower WHRs because of the lower air mass flow rate, but it is observed that at high relative humidities, the WHR tends to a constant value due to the decrease of the air mass flow rate. Then, at $30\text{ }^{\circ}\text{C}$ the impact of the air mass flow rate decrease at high relative humidities is more significant, which explains the WHR decrease at high relative humidities and causing the WHR to reach a maximum at a given relative humidity, which depends on the refrigerant type. The trend of the SEC is a consequence of the WHR trend, due to the definition given in Eq. (68).

Concerning the COP, at low ambient temperatures, i.e. $10\text{ }^{\circ}\text{C}$, and low relative humidities, the air mass flow rate is high, which means a higher input power for the axial fan, and then it decreases as the relative humidity increases, which means a lower input power for the axial fan. On the other hand, at low ambient temperatures the compressor pressure ratio is low, which means lower compressor input power. In fact, at $10\text{ }^{\circ}\text{C}$ and 75% of relative humidity, both compressor and axial fan input powers are of the same order of magnitude. Thus, although the compressor input power increases as the relative humidity increases, the input power of the axial fan decreases more quickly, resulting in a decrease of the total system input power and thus an increase in the COP. It is not observed at $20\text{ }^{\circ}\text{C}$ and $30\text{ }^{\circ}\text{C}$ because at these temperatures the compressor input power is always higher than the input power of the axial fan, because the air mass flow rate at those conditions is lower than

the ones obtained at $10\text{ }^{\circ}\text{C}$.

In these figures, it is also observed that at low relative humidities, the SEC increases exponentially. It is explained for the high sensible load that the evaporator has to manage at those conditions, which reduces the evaporator power available for the latent load. In fact, in dry climates is better to use other kind of technologies, such as sorption technologies (Rao et al., 2022; Feng et al., 2022), or to combine VCR systems with sorption systems (Cendoya et al., 2023).

These results are very influenced by the control system used in this analysis. For example, when the ambient temperature increases, as the evaporator exhaust temperature is fixed at $4\text{ }^{\circ}\text{C}$, the air mass flow rate circulating through the AWG must decrease. That explains the reduction in the collected water, despite having a higher water potential in the ambient air. These results suggest that is better to have a control with a variable air mass flow rate and a variable temperature of the air exhausting the evaporator, in order to maximize the collected water.

3.2. Simulation analysis in different cities

The atmospheric water generator is evaluated under steady-state regime considering the climate conditions of Beijing, Cairo, Harare, Kabul, New Delhi, Santiago de Chile, and Sao Paulo. Fig. 7 presents the results obtained for the annual hourly average SEC, annual hourly average WHR and the number of hours during a year that the system cannot operate (hour_{non}) for the five refrigerants. The number of hours that the system cannot operate includes the hours when the climate conditions present a dew point temperature lower than $4\text{ }^{\circ}\text{C}$, the hours when the condensing pressure exceeds the refrigerant critical pressure and the hours when the compressor pressure ratio is too low (lower than 1.5).

In terms of SEC, refrigerants R407C and R410A presents the best results, with R407C being better in Cairo, Harere, Kabul and Sao Paulo, and R410A in Beijing, New Delhi and Santiago. Refrigerant R290 is in most of cases the third best option.

In the case water harvesting rate, the best performance is obtained with refrigerant R407C, followed by R410A and R134a. The worst performance is obtained with refrigerants R290 and R1234yf.

Concerning the non-operating hours, Beijing and Kabul, being the coldest cities, presents the higher values. It means that the main limitation in the working domain is the low range of ambient temperatures. On the other hand, in New Delhi, being the hottest city, only refrigerant R1234yf is more severely affected. In spite of that, high temperatures

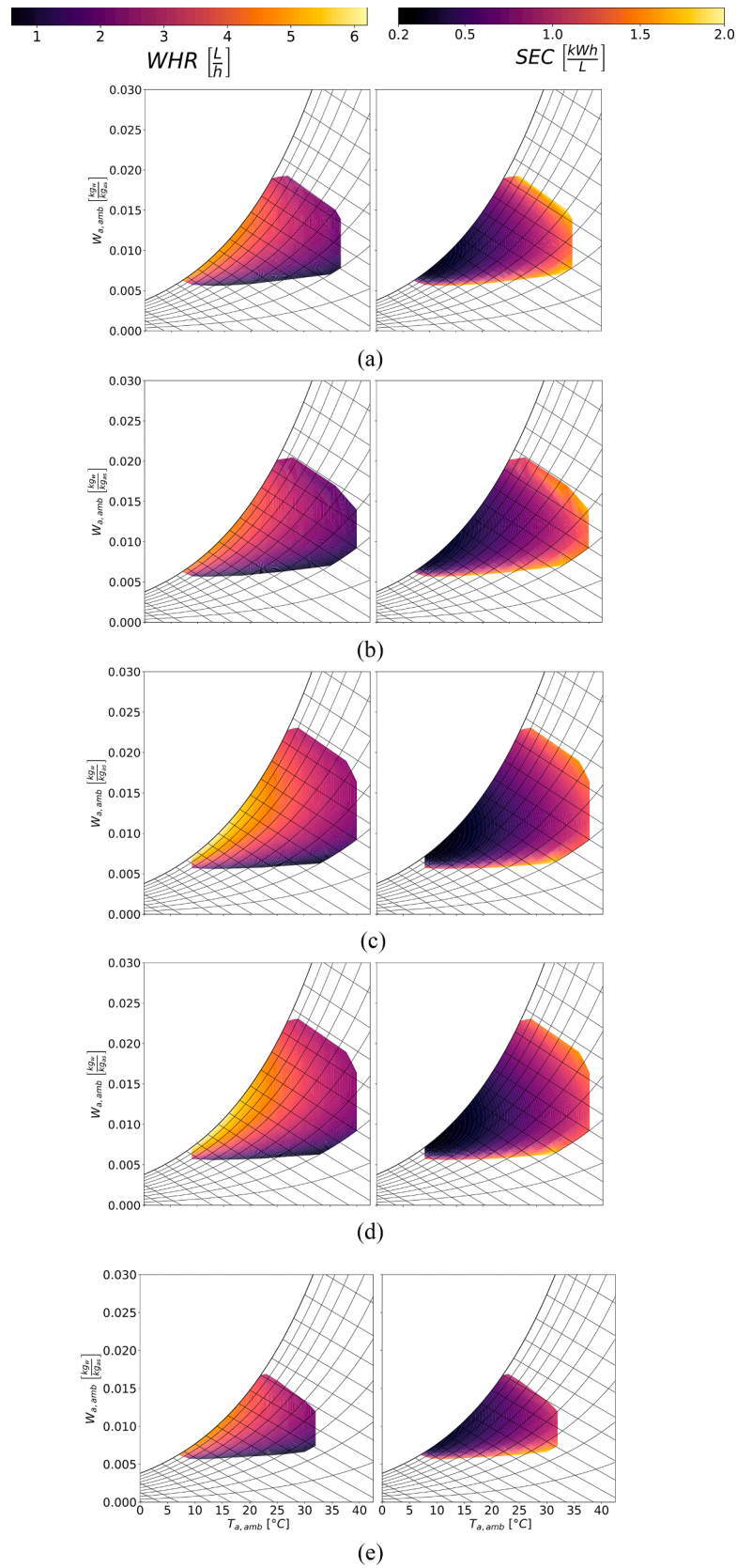


Fig. 5. WHR and SEC for different refrigerants under different climatic conditions, a) R134a, b) R290, c) R407C, d) R410A and e) R1234yf. The area covered for each refrigerant indicates the system working domain and the colours the ambient conditions where the refrigerants considered perform best.

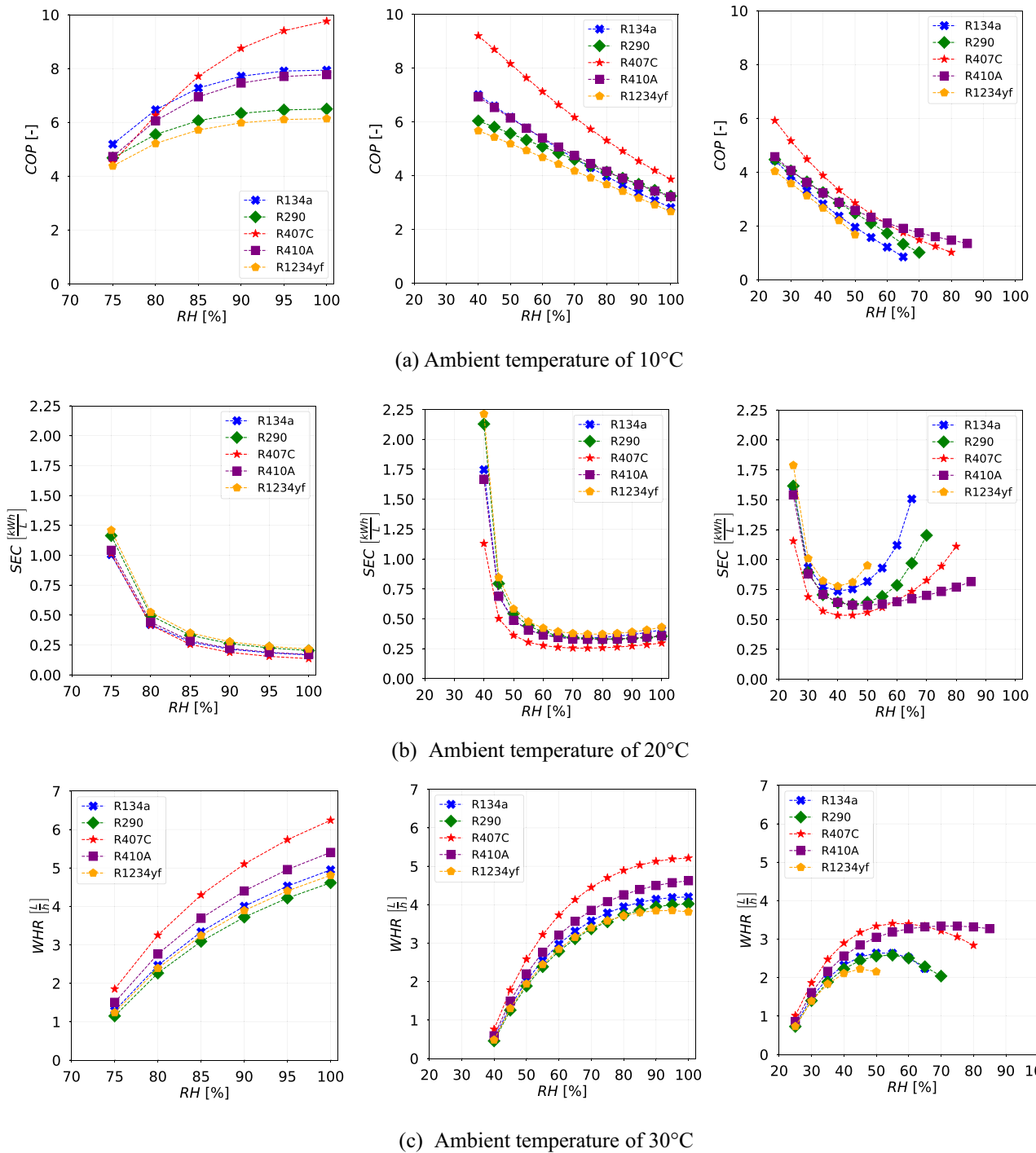


Fig. 6. WHR, SEC and COP (left to right) of R134a, R290, R407C, R410A and R1234yf under three different scenarios: a) ambient temperature of 10 °C and different relative humidities, b) ambient temperature of 20 °C and different relative humidities, and c) ambient temperature of 30 °C and different relative humidities. The column at the left presents the Water Harvesting Rate, the column located in the centre presents the Specific Energy Consumption and the column located at the right the system Coefficient Of Performance.

could be considered as a limitation, which is observed in the behaviour of Cairo and New Delhi.

According to these findings, refrigerant selection has a significant impact on the SEC, Water Harvesting Rate and operating hours. For example, taking Sao Paulo as reference, refrigerant R407C allows to collect 27 % more water than refrigerant R1234yf. In terms of SEC, refrigerant R407C allows to obtain SECs 20 % lower than the ones obtained with refrigerant R1234yf. Thus, refrigerant selection has a significant impact on the system performance.

Another interesting finding obtained from these results is the number of hours that the system does not operate due to the unfavourable ambient conditions. Beijing and Kabul are the cities with the higher number of hours with non-operating hours. The main reason of that are the low ambient temperatures of these cities, compared to the other ones. There is no correlation with the Köppen classification, because Beijing presents a humid continental climate but with cold and dry winters, which penalize the operation of AWGs, and Kabul presents a cold-semi-arid climate, with hot and dry summer, but with cold winters.

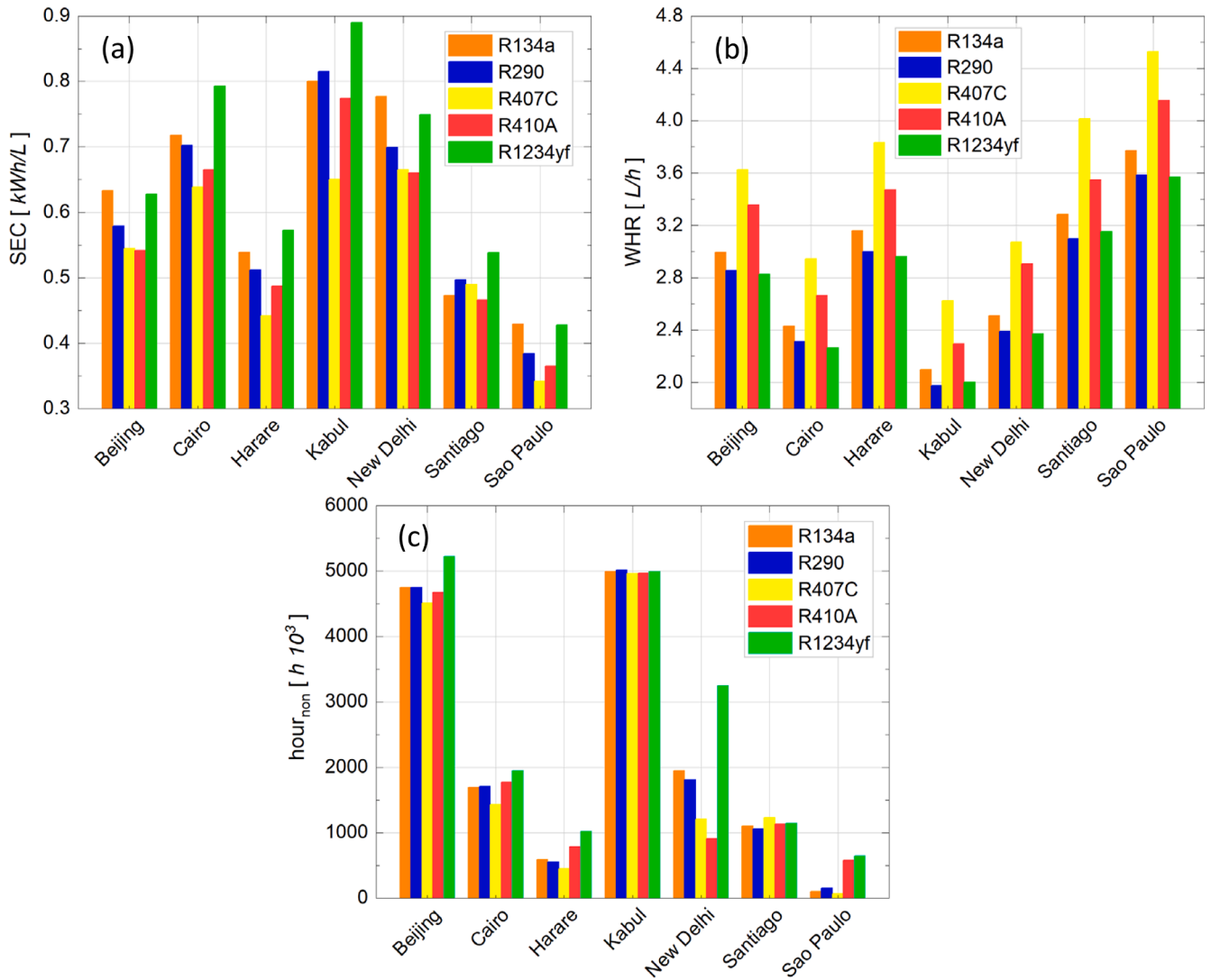


Fig. 7. Annual simulation results for each refrigerant in the cities considered in this study (a) annual hourly average SEC, (b) annual hourly average WHR, (c) non-operating hours.

Both cities have similar non-operating hours, but Beijing have a humid continental climate allowing to collect more water than Kabul and other cities having lower non-operating hours such as Cairo and New Delhi. In conclusion, in terms of water collection, climates classified under the group C by Köppen classification seems to have the better conditions for this technology. Although none of the cities considered in this study are in group A, cities classified in this group should also present favourable conditions for this technology.

Concerning the refrigerant choice, it was observed that for climates with very large amount of hours with ambient temperatures lower than 4 °C, independent of the refrigerant choice, AWGs are not suitable.

In conclusion, and despite the fact that it is a phasing-out refrigerant, refrigerant R407C is the best choice for AWGs in terms of WHR and SEC. In terms of the economic feasibility, according to Ravesh et al. (Ravesh et al., 2023), the system is economically feasible if the SEC is lower than 1 kWh L⁻¹. According to the results, all the cities considered in this study comply, in average, with the economic feasibility.

3.3. Critical analysis

To understand the results obtained from the simulations, the following coefficients and information is presented. Firstly, it is introduced a coefficient that takes into account the thermo-physical prop-

erties of the refrigerants used in the determination of the convective heat transfer coefficients for the condenser (Eq. (70)) and for the evaporator (Eq. (71)).

$$K_{cd} = \frac{k^{2/3} \cdot (\rho \cdot c_p)^{1/3}}{\nu^{0.47}} \quad (70)$$

$$K_{ev} = \frac{k^{3/5} \cdot (\rho \cdot c_p)^{0.4}}{\nu^{0.4}} \quad (71)$$

The results obtained with these equations and the critical pressure and temperature for every refrigerant are presented in Table 8. Concerning the properties that favour the convective heat transfer coefficients, it is observed that the refrigerant with the best to the worst properties are R290, R410A, R407C, R134a and R1234yf. Concerning the critical pressure from the highest to the lowest the classification is R410A, R407C, R290, R134a and R1234yf, and for the critical temperature R134a, R290, R1234yf, R407C and R410A.

Another interesting variable to take into account for the analysis is the compressor isentropic effectiveness, which is shown in Fig. 8. According to the results, near to the design range, the refrigerants that present the highest to the lowest isentropic effectiveness are R407C, R410A, R134a, R290 and R1234yf. It is also observed that the maximum isentropic effectiveness for R290 and R1234yf is located at a value of a

Table 8

Coefficients that takes into account the thermo-physical properties of the refrigerants used in the determination of the convective heat transfer coefficients for the condenser and for the evaporator, and critical pressure P_{crit} and temperature t_{crit} .

Refrigerant	K_{ev} $t_{ev} = 0$ °C $x = 0$	K_{ev} $t_{ev} = 0$ °C $x = 1$	K_{ev} $t_{cd} = 40$ °C $x = 0$	K_{ev} $t_{cd} = 40$ °C $x = 0$	P_{crit} kPa	t_{crit} °C
R410A	160	37	170	73	4901	71.3
R407C	135	29	150	55	4632	86.2
R290	170	43	182	76	4247	96.7
R134a	120	27	130	49	4059	101.0
R1234yf	107	29	117	50	3382	94.7

pressure ratio between 3.3 and 3.5.

After the simulations, the following conclusions are obtained in order to help to the atmospheric water generator designers:

- HFC refrigerants seems to be the most performants refrigerants among the refrigerants evaluated in this study. In terms of harvested water, refrigerant R407C performs better, followed by refrigerant R410A. On the other hand in terms of SEC, the behaviour is almost similar,
- It is observed that de design compressor pressure ratio plays also an important role on the system design. For refrigerants R290 and R1234yf the optimal pressure ratio to maximize the isentropic effectiveness is around 3.3 to 3.5. Thus, it will be interesting for the designers to evaluate the effect of the pressure ratio on the system design, because in this case, by increasing the design pressure ratio will allow to increase the compressor isentropic effectiveness but the volumetric effectiveness will decrease and the compressor input power could to increase or to decrease (on the one hand it will decrease due to the increase of the isentropic effectiveness and on the other hand it will increase due to the increase of the pressure ratio),
- According to the results presented in Fig. 7, cities with low temperatures (Beijing and Kabul) present the higher non-operating hours. On the other hand, hot climates, like the ones of New Delhi and Cairo have also an impact on the non-operating hours, making the system works at condensing pressure higher than the critical pressure. The only way to avoid this is to select a refrigerant with a high critical pressure or oversizing the condenser, but it will cause lower compressor pressure ratios al low ambient temperatures, by

degrading the compressor isentropic effectiveness. For the explored refrigerants and for the design conditions used in this study, the ideal range of temperatures are between 4 °C and almost 33 °C to maximize the operating hours,

- As recommendation, in hot climates the airstream could be pre-cooled before entering the AWG to reduce the non-operating hours due to high condensing pressures and to increase the system performance (higher WHRs and SECs),
- It seems that there exists a combination of air mass flow rate circulating through the atmospheric water generator and air temperature at the evaporator exhaust that maximize the harvested water. According to this, it is also advisable to modulate the compressor capacity, using for example a variable speed compressor,
- It is advisable to select a refrigerant with a high critical pressure to avoid working at supercritical conditions.

Thus, it is recommended to size the Atmospheric Water Generators considering the local climate conditions, because if a system is sized for medium temperatures, i.e. 20 °C, and it is used in climates with high temperatures, for example up to 35 °C, it will experiment very high condensing pressures, because to maintain a low temperature at the evaporator the system must to decrease the air mass flow rate penalizing the working conditions of the condenser. On the other hand, if the system is designed for high ambient temperatures, i.e. 35 °C, and then it is used in a medium temperature climate, i.e. 5 °C to 25 °C, the air mass flow rate must be increased to avoid frosting on the evaporator surface, which will reduce the compressor pressure ratio down to values that could be lower than 2 by penalizing the compressor performance.

Finally, by comparing the results of this study with the ones obtained by Raveesh et al. (Raveesh et al., 2023) in Indian cities, the only study that develop a similar analysis, similar conclusions are obtained: several non-operating hours in cities with cold winters. Concerning the most favourable city for water harvesting of this study (Sao Paulo), the average annual SECs varies between 0.34 kWh L⁻¹ for refrigerant R407C and 0.43 kWh L⁻¹ for refrigerant R134a and R1234yf. In the case of New Delhi, the SEC varies between 0.66 kWh L⁻¹ for refrigerant R410A and 0.78 kWh L⁻¹ for refrigerant R134a. By considering the results of Raveesh et al. (Raveesh et al., 2023), who use refrigerant R134a, the results obtained in this study are within the range informed in their study.

3.4. Environmental impact

In addition to the previous analysis, an environmental assessment is

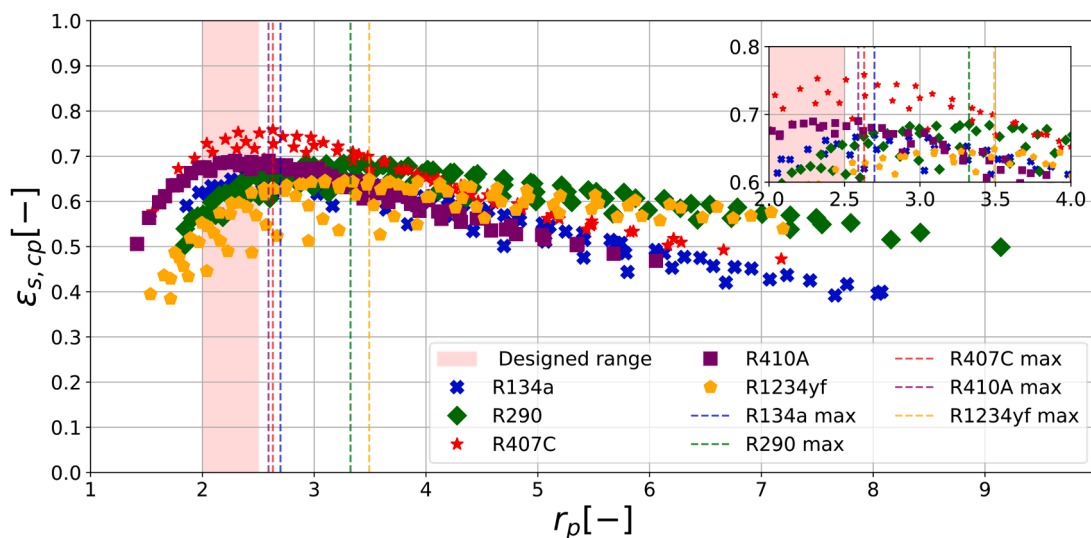


Fig. 8. Compressor isentropic effectiveness against the compressor pressure ratio. The segmented vertical line presents the location of the maximum isentropic effectiveness for each refrigerant and the coloured vertical zone corresponds to the conditions used in the design of the system.

developed using the Total Equivalent Warming Impact (TEWI), which is determined by Eq. (72). The first two terms represent the direct emissions due to the leakages while the third term stands for the indirect emissions, related to the emissions due to the electricity consumption. M_{leak} is the refrigerant leakage, which is considered as 5 % of the refrigerant charge per year (Aguiar et al., 2005); M_{charge} (0.22 kg kW⁻¹) is the total refrigerant charge (estimated from a domestic monoblock air conditioner (Poggi et al., 2008)); and λ (90 %) is the refrigerant recycling factor (AIRAH, 2012). On the other hand, E is the annual electricity consumption obtained from the previous simulations and β is the grid emission factor of the country, obtained from the International Renewable Energy Agency (IRENA) database (IRENA, 2025) for the year 2022 (569 tCO₂ eq GWh⁻¹ in Beijing, 367 tCO₂ eq GWh⁻¹ in Cairo, 365 tCO₂ eq GWh⁻¹ in Harare, 1432 tCO₂ eq GWh⁻¹ in Kabul, 723 tCO₂ eq GWh⁻¹ in New Delhi, 74 tCO₂ eq GWh⁻¹ in Sao Paulo and 355 tCO₂ eq GWh⁻¹ in Santiago (Chile)). Finally, a lifetime n of 15 years is considered for each system (Heikkilä, 2004).

$$TEWI = (GWP \cdot M_{leak} \cdot n) + GWP \cdot M_{charge} \cdot (1 - \lambda) + E \cdot \beta \cdot n \quad (72)$$

The results of the direct and indirect emissions are presented in Fig. 9 for each refrigerant and climate conditions. In addition, this figure shows the annual equivalent CO₂ specific emissions M_l (per litre of harvested water). Excluding the refrigerants R290 and R1234yf, the direct emissions represent less than 10 % of the total TEWI. The environmental profile is mainly related to the regional grid emission factor; with Kabul and New Delhi having the higher emissions and Sao Paulo having the lower ones. Among the refrigerants considered in this study, the environmentally hazardous refrigerants are the R410A, R407C and R134a, whilst the least polluting are the R1234yf and R290. Concerning the specific emissions, Kabul presents the highest emissions and Sao Paulo the lowest ones. The emissions are strongly dependent on the local grid emissions and on the climate conditions, being the warmer climates the ones with the higher energy consumptions and thus with the highest emissions.

4. Conclusions

This study evaluated the effect on the performance of an Atmospheric Water Generator, based on a vapour compression refrigeration

cycle, of five different refrigerants (R134a, R290, R407C, R410A and R1234yf) with their corresponding commercial compressors, different ambient conditions (from a parametric table and climate conditions of seven different cities) and different air mass flow rates circulating through the system. As a useful matter, this work delivers a performance map for each refrigerant, as well as the performance and environmental-based recommendations, and critical analysis for the system design.

The results indicate that at low temperatures the optimum WHR is located at high relative humidities (near to 100 %), whereas at higher temperatures it is located a medium relative humidities. The minimum SEC simultaneously depends on the temperature and relative humidity. While the temperature increases, the optimal conditions that minimize the SEC change from a relative humidity near to 100 % for any temperature to 65 % for 20 °C and to 35 % for 30 °C. Since the optimum relative humidity at a given temperature will not match the optimum SEC and WHR, a trade-off must be developed between these variables.

As a general contribution, a performance psychrometric-heat map for the five Atmospheric Water Generators was presented. Among the evaluated systems, the one using refrigerant R407C presents the best performance across the systems, while the one using refrigerant R410A presents good performance at high temperatures (>30 °C) and relative humidities below 80 %. This map is strongly influenced by the design conditions: 20 °C and 60 % relative humidity. A good design of an Atmospheric Water Generator must maximize the number of operating hours, maximize the system efficiency, maximize the water production, especially during periods of water stress or shortage, minimize energy consumption, etc. Therefore, a single climate design condition is not sufficient to ensure reliable year-round operation.

Regarding the seven selected cities, the system using refrigerant R407C is the one with better performance, while the one using refrigerant R410A is better for warmer cities such as New Delhi or Cairo. On the other hand, R290 also has a good performance, with equivalent CO₂ emissions among the lowest, as well as refrigerant R1234yf. This environmental impact depends strongly on the local grid emission factor than on the refrigerant itself. From the cities analysed, New Delhi and Kabul present the highest emissions due to the higher grid emission factors and higher energy consumption.

One way to improve the system performance is replacing the fixed-speed compressor by a variable speed compressor. This will allow adding a new control variable to optimize water production and to reduce

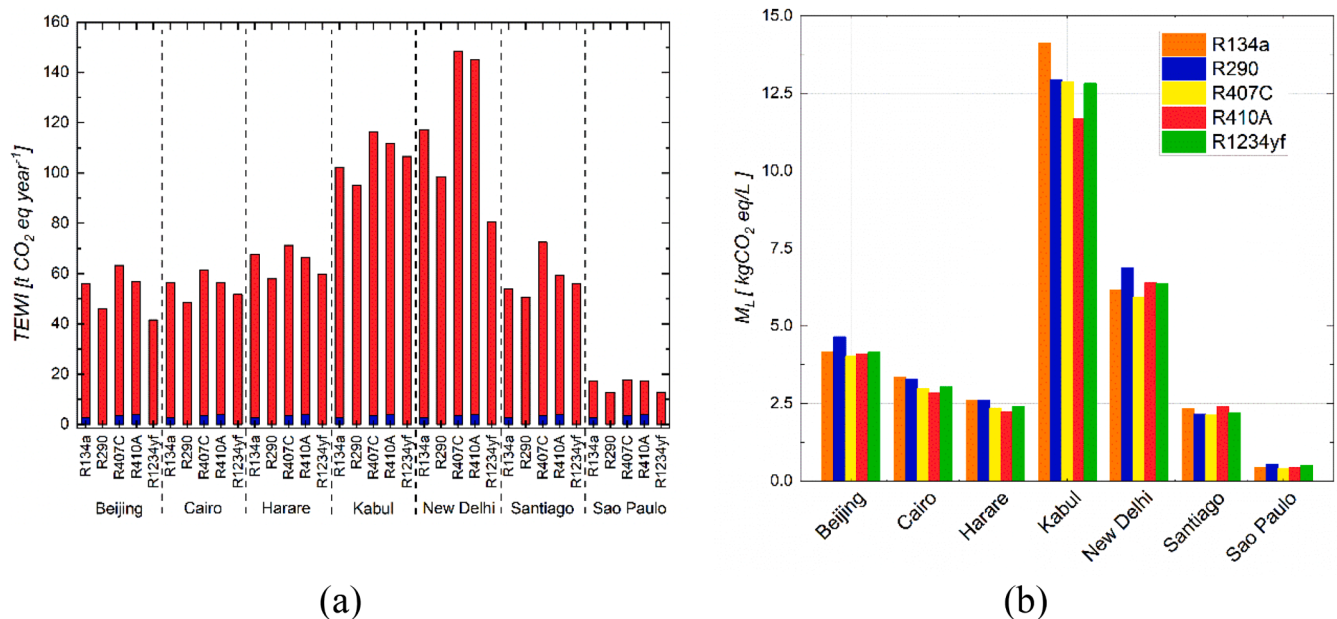


Fig. 9. (a) Direct (blue bars) and indirect (red bars) annual emissions (TEWI), and (b) Equivalent CO₂ emission per litre of collected water for the five refrigerants and the seven cities considered in this study.

the energy consumption. It will also allow to expand the operating map of the system due to this new degree of freedom.

CRedit authorship contribution statement

Aitor Cendoya: Writing – original draft, Validation, Software, Methodology, Investigation. **Daniel Sacasas:** Writing – original draft, Validation, Software, Methodology, Investigation. **Matias Pezo:** Writing – original draft, Validation, Software, Methodology, Investigation. **Cristian Cuevas:** Writing – review & editing, Writing – original draft, Supervision, Methodology. **Enrique Wagemann:** Writing – original draft, Validation, Software, Methodology, Investigation.

Declaration of competing interest

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

Cristian Cuevas reports financial support was provided by National Agency for Research and Development. Aitor Cendoya reports financial support was provided by National Agency for Research and Development. Matias Pezo reports financial support was provided by National Agency for Research and Development. Daniel Sacasas reports financial support was provided by National Agency for Research and Development. Enrique Wagemann reports was provided by National Agency for Research and Development. If there are other authors, they declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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