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Supplementary Information

Addressing Global Water Stress Using Desalination and Atmospheric Water Harvesting: A Thermodynamic and Technoeconomic Perspective

Jordan D. Kocher and Akanksha K. Menon*

George W. Woodruff School of Mechanical Engineering, Georgia Institute of Technology, Atlanta, GA, USA

*Corresponding author. Email: akanksha.menon@me.gatech.edu

Supplementary Note 1: Assumptions (for Cost and Heat Transfer Analyses)

In our analysis of the LCOW of distributed desalination and AWH, we made various assumptions about capital expenditures, operating expenditures, system efficiencies, and operational parameters. These values are given in Table S1 below.

Table S1: Table of assumptions | List of all assumed costs, system performance values, and operational parameters.

System	Variable	Symbol	Unit	Value	Notes
Active Cooling AWH	Vapor Compression Capital Cost	$CAPEX_{VC}$	\$/ton	500[1]	Based on a DOE report that lists the manufacturing costs of a 2-ton air conditioning system as \$690, plus an assumed markup of approximately 50%
	Levelized Cost of Electricity	LCOE	$\$/kWh_e$	0.06[2]	Based on the average industrial retail electricity price
	Vapor Compression Coefficient of Performance	COP		3	
	Evaporator Outlet Temperature (Practical Operation)	T_{evap}	°C	5	We picked a value low enough to yield significant water in most locations, while still being high enough to avoid freezing
	Temperature Difference Between Evaporator and Ambient (Reversible)	ΔT_{evap}	°C	10	
Passive Cooling AWH	Passive Radiative Cooling Surface Capital Cost	$CAPEX_{PRC}$	$\$/m^2$	36.5[3]	Based on the cost of a 1 ft^2 sheet of TPX (polymethylpentene). Note that TPX is the matrix material into which silica spheres are embedded to achieve radiative cooling[4], but this additional cost is not considered here. As a reference, SkyCool's commercial radiative cooling panels are quoted at $\$750/m^2$
	Heat Transfer Coefficient	U	W/m^2-K	10	
Sorbent AWH	Sorbent Capital Cost	$CAPEX_{sorb}$	\$/kg	3[5]	Based on MOF-303[6,7]. MOF-303 is based on aluminum, which is \$3/kg. To provide a generous analysis of the sorbent system, we assume that MOF-303 could approach this \$3/kg value
	Levelized Cost of Heat	LCOH	$\$/kWh_{th}$	0.01[8]	Based on industrial average price of natural gas in the U.S. in 2020
Water Storage	Water Storage Capital Cost	$CAPEX_{WS}$	$\$/m^3$	100[9]	
Coastal Seawater Desalination + Water Transport	Seawater Desalination LCOW	$LCOW_{desal}$	$\$/m^3$	1[10,11]	
	SEC of seawater desalination		kWh_e/m^3	4[12]	Upper end of the range given by Menon *et al.*[12]
	Vertical Water Transport Cost		$(\$/m^3)/m$	5×10^{-4} [13]	Water transport costs are based on research by Zhou and Tol[13]. In their work, Zhou and Tol review previous findings on water transport costs around the globe and settle on 5¢ per 100 m vertical distance and 6¢ per 100 km horizontal distance. Note that our transport values are conservative compared to others that have been used[14].
	Horizontal Water Transport Cost (Canals)		$(\$/m^3)/km$	6×10^{-4} [13]	See above
	Horizontal Water Transport Cost (Tunnels)		$(\$/m^3)/km$	12.48×10^{-4} [13]	Based on a finding[13] that water transport by tunnel is 108% more expensive than by canal
	Horizontal Water Transport Cost (Pipes)		$(\$/m^3)/km$	22.26×10^{-4} [13]	Based on a finding[13] that water transport by pipe is 271% more expensive than by canal
Financial Parameters	System lifetime	n	yr	30	
	Fixed Operations and Maintenance Cost	$OPEX_{fix}$	% CAPEX/yr	2[9,15,16]	
	Discount Rate	r	%/yr	7[17]	

Supplementary Note 2: Criteria for Negligible Sorbent Cost

In our analysis of the sorbent-based AWH system, we found that the capital expenditure of the sorbent has a negligible contribution to the system LCOW. In this section, we quantify the general conditions under which the sorbent cost becomes negligible. Eq. (S1) gives the sorbent capital expenditure contribution to LCOW, where $CAPEX_{sorb}$ is the cost per unit mass sorbent (in \$/kg), CRF is the capital recovery factor (which is 0.08 for a discount rate of 7% and system lifetime of 30 years), $OPEX_{fix}$ is the fixed operations and maintenance cost (2% of the CAPEX per year in our analysis), and $Annual\ Yield_{sorb}$ is the annual yield of water per unit mass sorbent (in m^3 water per kg sorbent).

$$LCOW_{sorb,CAPEX} = \frac{CAPEX_{sorb} \times (CRF + OPEX_{fix})}{Annual\ Yield_{sorb}} \quad \text{(S1)}$$

For a well-designed sorbent AWH system, the sorbent will be capable of harvesting moisture for nearly the entire range of relative humidities that occur in the location where it is deployed. Then, assuming the system cycles once per hour, Eq. (S2) results for the annual yield (in m^3 water per kg sorbent), where W_{max} is the water uptake of the sorbent (in kg water per kg sorbent).

$$Annual\ Yield_{sorb} \approx \frac{24 \times 365 \times W_{max}}{1000} \quad \text{(S2)}$$

When $LCOW_{sorb,CAPEX} \leq \$0.10/m^3$, the sorbent capital cost can be treated as negligible relative to the energy cost, since the energy cost in practical sorbent systems is always at least on the order of \$1/m^3 (and sometimes on the order of \$10/m^3). For $LCOW_{sorb,CAPEX} \leq \$0.10/m^3$ to be true, the following inequality must also be true (based on Eq. (S1) and (S2)):

$$\left(\frac{CAPEX_{sorb}}{W_{max}}\right)_{neg} \leq \frac{0.1 \times 24 \times 365}{1000 \times (0.08 + 0.02)} \tag{S3}$$

Simplifying reveals that the sorbent capital cost is negligible when $\frac{CAPEX_{sorb}}{W_{max}} \leq \$8.70$ per kg of water.

Supplementary Note 3: Energy Consumption for Water Transport

For water transport, we refer to Zhou and Tol[13], who provide a review of the cost associated with water transport from various sources in the literature. In their work, they cite Kally[18], who states that investment (capital expenditure) and operating expenditure have an equal share in the cost of transporting water. As such, we have assumed in our analysis that of the $\$5 \times 10^{-4}$ /m^3 per m of vertical transport, half of the cost comes from energy consumption. Dividing $\$2.5 \times 10^{-4}$ /m^3 per m by an LCOE of \$0.06/kWh gives 4.2×10^{-3} kWh/m^3 per m of vertical transport. In addition, we can compare the energy consumption value that we used to a real-world value. The California River Aqueduct is a water transport system within the U.S. that transports an average of 1.5×10^9 m^3 of water each year[19] over a horizontal distance of 242 mi and a net vertical distance of 1357 ft[20,21], with an annual energy consumption of 2×10^9 kWh[21]. To compare our vertical transport energy consumption to a worst-case value, we can attribute all of the Colorado River Aqueduct energy consumption to the vertical transport portion; this results in 3.3×10^{-3} kWh/m^3 per m of vertical transport. Therefore, the vertical transport energy consumption value that we used in our energy analysis is 30% larger than the worst-case value of the Colorado River Aqueduct.

Likewise, we take our horizontal transport value of $\$13.58 \times 10^{-4}$/m^3 per km, multiply by 0.5, and divide by an LCOE of \$0.06/kWh to get an energy consumption of 11.3×10^{-3} kWh/m^3 per km of horizontal transport. This number can again be compared to real world values. If all of the Colorado River Aqueduct's energy consumption is attributed to horizontal transport, the resulting energy consumption is 3.4×10^{-3} kWh/m^3 per km of horizontal transport. Thus, the horizontal transport energy consumption value that we used is three times greater than the worst-case value of a major real-world water transport system. In addition, another recent work analyzed the cost of water transport and used energy consumption and cost values significantly lower than ours[14]. This means our energy analysis of water transport is very conservative, and the real energy consumption required to transport water is likely far less than the values we calculated in this work.

Supplementary Note 4: Energy Comparison on the Basis of Heat

In Fig. 3 of the main text, reversible AWH was compared to practical desalination (coastal desalination with water transport) on the basis of work input. However, heat-driven AWH systems (*e.g.,* sorption-based) are common, and the energy input to heat- and work-driven systems cannot be directly compared. Instead, the heat input for AWH should be compared to the heat required to produce the work that drives the desalination and water transport processes. This heat required can be calculated by multiplying the least work of separation from Eq. (1) of the main text with a factor of $(1 - T_{amb}/T_S)^{-1}$, where T_S is the temperature of the heat source (assumed to be 200 °C). Meanwhile, the heat required to produce electricity for desalination and water transport is calculated by dividing the specific energy consumption of those processes by the efficiency of the power plant used to generate the electricity (assumed to be 40%). The results of this energy analysis are given in Fig. S1. Heat-driven AWH requires more heat than what is required to generate electricity for coastal desalination with water transport for a vast majority of the global population. These results are remarkably similar to Fig. 3 of the main text, because the factor of $1 - T_{amb}/T_S$ equates to 0.37 when T_{amb} is 300 K, which is very close to the assumed power plant efficiency of 0.4. For T_{amb}, we use the average annual temperature at each location.

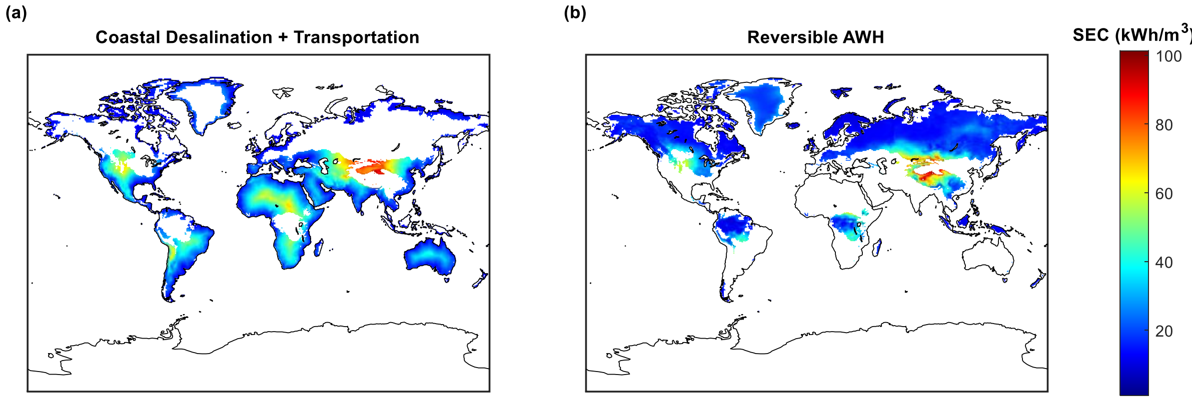

Fig. S1. Analysis of the technology that requires the least amount of heat to produce a unit volume of water in each global location. Contour plot of the specific energy consumption (SEC) of coastal seawater desalination with transportation (a) and reversible AWH (b). The map in (a) is colored only in locations where coastal desalination with transport requires less heat than AWH, while the map in (b) is colored in locations where reversible AWH requires less heat. The area where reversible AWH is more efficient accounts for 41.7% of the global land area, but only 17.4% of the global population and 14.7% of the global water risk weighted population lives in that area.

Supplementary Note 5: AWH Energy Cost Sensitivity Analysis

In the main text, we found that the LCOW of sorbent AWH is dominated by the energy cost. As such, if a significantly cheaper heat source were used, sorbent AWH would become more cost effective. This is quantified in Fig. S2, which shows the LCOW of sorbent AWH in Niger with different regeneration temperatures and different LCOH values. If the LCOH is as low as 0.1 ¢/kWh_{th}, sorbent AWH can reach cost parity with seawater desalination (~$1/$m^3$). However, it might be difficult to find a heat source that can regenerate the sorbent at 135 °C. As such, we investigate two lower regeneration temperatures: 80 °C and 65 °C. At 80 °C, the energy consumption is only 37% higher than at 135 °C, but the lower LCOH would be more feasible at 80 °C. However, at a regeneration temperature of 65 °C, the energy consumption is 130% higher than at 135 °C, indicating this regeneration temperature is too low to become cost effective. Thus, the cost of heat should be balanced with the energy consumption of the system when choosing a regeneration temperature. The optimal temperature will likely be lower than the regeneration temperature that yields peak efficiency, but sufficiently higher than the minimum allowable regeneration temperature.

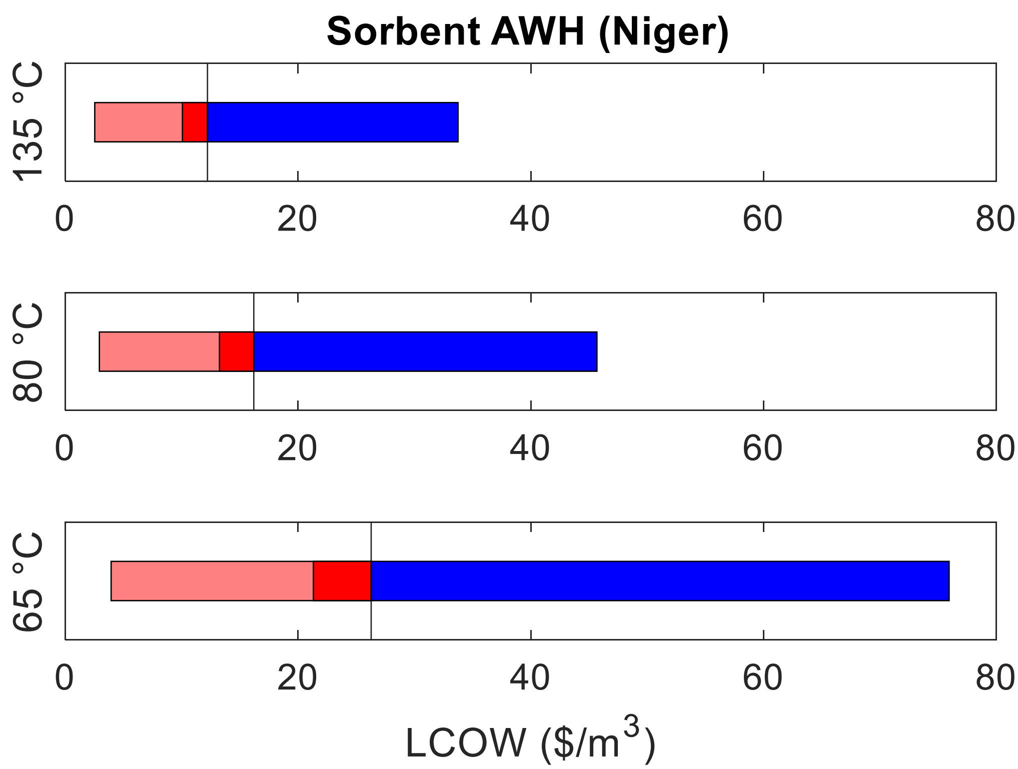

Fig. S2. Sensitivity study of practical sorbent AWH. The light red bar corresponds to an LCOH of 0.1 ¢/kWh, the dark red bar corresponds to an LCOH of 0.8 ¢/kWh, the baseline LCOH value is 1 ¢/kWh, and the blue bar corresponds to an LCOH of 3 ¢/kWh. Three different regeneration temperatures were investigated: 135 °C, 80 °C, and 65 °C. Results are only presented for Niger because the variation between locations is insignificant for sorbent-based AWH.

The heat associated with sorbent AWH is sometimes described as being "free" in the literature, but even solar and waste heat require capital expenditures to collect and transfer that heat from the source to the sorbent. If we use $1/m^3 as a benchmark value for LCOW (due to the comparisons we have drawn between AWH and desalination), then a heat source can reasonably be considered "free" if it contributes less than $0.1/m^3 to the LCOW of AWH (an order of magnitude less than the benchmark LCOW of seawater desalination). For a regeneration temperature of 135 °C, the LCOH would need to be as low as 1×10^{-2} ¢/kWh$_{th}$ to be reasonably considered as "free", while a regeneration temperature of 65 °C would need an LCOH of 7.7×10^{-3} ¢/kWh$_{th}$ or lower to be free. These values are both likely to be impractically small, considering that flat plate solar heaters have an LCOH of 1.3 ¢/kWh_{th}[12], which highlights the importance of considering the LCOH when calculating sorbent AWH costs.

We also quantified how sensitive the LCOW of active cooling AWH is to the LCOE. The results are presented in Fig. S3 and indicate that even with significantly lower electricity costs, active dew harvesting is unlikely to be a cost-effective method of water production.

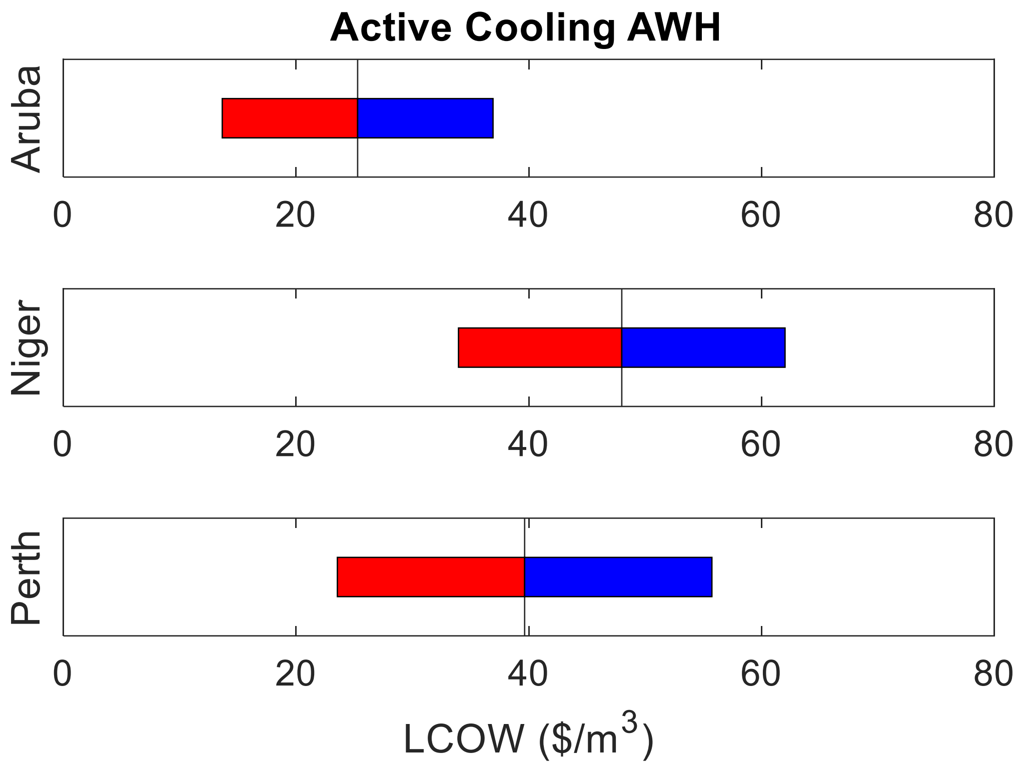

Fig. S3. Sensitivity study of active cooling AWH. Red bars correspond to an LCOE of 3 ¢/kWh, baseline LCOE is 6 ¢/kWh, and blue bars correspond to an LCOE of 9 ¢/kWh.

Supplementary Note 6: Active Cooling AWH Energy Consumption

For the active cooling system, it was assumed that air enters the evaporator at ambient temperature and humidity, and it leaves at a fixed evaporator temperature of 5 °C. To keep the cooling power of the system constant, the mass flow rate of air varies. The air mass flow rate can be found from the following equation:

$$\dot{m}_a = \frac{3.5 \times \dot{Q}_{ton}}{c_{p,a}\left(T_{amb} - T_{evap}\right) + h_{fg}\left(w_{amb} - w_{evap}\right)} \tag{S4}$$

where the factor of 3.5 converts from tons of refrigeration to kW_{th}, $\dot{Q}_{ton}$ is the fixed cooling power of the AWH system (in tons), $c_{p,a}$ is the specific heat of air (in kJ/kg-K), h_{fg} is the enthalpy of vaporization of water, and w is the humidity ratio (in kg water per kg dry air). At the evaporator outlet (subscript *evap*) the humidity ratio is the lesser of two values: the saturation humidity ratio at the evaporator temperature (5 °C) or the ambient humidity ratio. If the evaporator temperature saturation humidity ratio is the lesser of the two, then water condenses and AWH is achieved. If the ambient humidity ratio is the lesser of the two, then the ambient air is too dry to harvest water with the given evaporator temperature.

From the mass flow rate of air, the hourly water production, $\dot{V}_w$, can be found (in units of m^3/ton-h):

$$\dot{V}_{w,active\ cool,h} = \frac{12.6 \times \left(w_{amb} - w_{evap}\right)}{c_{p,a}\left(T_{amb} - T_{evap}\right) + h_{fg}\left(w_{amb} - w_{evap}\right)} \tag{S5}$$

where the factor of 12.6 combines the factors of 3.5, 3600 (to convert from 1/s to 1/h) and 10^{-3} (the specific volume of water to convert from kg to m^3).

The electricity consumed during a given hour is equal to the system cooling power (in kW), multiplied by 1 h, and divided by the system COP (which for simplicity we assume is always 3). The specific energy consumption is equal to the total amount of electricity consumed throughout the year (in kWh), divided by the total yearly water production:

$$SEC_{active\ cool} = \frac{h_{on}}{3.6 \times COP \times \sum_{h=1}^{8760} \frac{\left(w_{amb} - w_{evap}\right)}{c_{p,a}\left(T_{amb} - T_{evap}\right) + h_{fg}\left(w_{amb} - w_{evap}\right)}} \tag{S6}$$

where h_{on} is the number of hours per year that the AWH system is on. The only reason why h_{on} would be less than 8760 is if during certain hours the system would be incapable of harvesting any water (*i.e.*, $w_{evap} = w_{amb}$).

For the reversible active cooling system, the water production (in units of m^3/ton-h) during a given hour of the year is:

$$\dot{V}_{w,active\ cool,rev,h} = \frac{3.5}{W_{min} \times COP_{Carnot}} \tag{S7}$$

where W_{min} is the least work of separation from Eq. (1) of the main text (in units of kWh_e/m^3), and COP_{Carnot} is the COP of a reversible refrigeration cycle. Not only would the refrigeration cycle be internally reversible, but the water production itself would be reversible (requiring less energy than the enthalpy of condensation). Essentially, the refrigeration cycle would reversibly convert electricity to cooling (at a temperature of T_C), and the cooling would then somehow reversibly convert water vapor to liquid water. In addition, we assumed that there were no limitations related to freezing on the reversible active cooling system (meaning the evaporator temperature could be lower than 0 °C). This operation is unrealistic, but it serves as the upper limit for active cooling AWH. By plugging in for the least work and Carnot COP, Eq. (S7) can be rewritten as Eq. (S8), where R_w is in units of kJ/kg-K:

$$\dot{V}_{w,active\ cool,rev,h} = -\frac{12.6}{R_w T_{amb} \ln(a_w) \times \frac{T_C}{T_{amb} - T_C}} \tag{S8}$$

For the reversible analysis, we assumed that the active cooling system would operate at 10 °C below ambient ($T_C = T_{amb} - 10$).

Eq. (S8) reveals that the reversible active cooling system would produce an infinite amount of water during hours when the relative humidity is 100%. As such, we placed a limit on the maximum amount of water that the system would produce during a given hour and varied the limit from 0 m^3 to an infinite amount of water. As the limit approached infinity, all of the year's water supply would be produced solely during times when the relative humidity was 100%. This would be achieved with zero energy input and an infinitesimally sized cooling system (*i.e.*, zero energy cost and zero cooling system capital cost), but it would require a significant amount of water storage. As such, it was found that the minimum LCOW was achieved in the reversible system with some finite limit on maximum water production (*i.e.*, excess water would be discarded during high relative humidity periods). The specific energy consumption across the entire year for the reversible active cooling AWH is then:

$$SEC_{active\ cool,rev} = -\frac{\sum_{h=1}^{8760} \min\left[\left(\frac{T_{amb}}{T_C} - 1\right), \dot{V}_{w,max} \times R_w T_{amb} \ln(a_w)\right]}{3.6 \times \sum_{h=1}^{8760} \min\left[\left(\frac{T_{amb} - T_C}{R_w T_{amb} T_C \ln(a_w)}\right), \dot{V}_{w,max}\right]} \quad \text{(S9)}$$

where $\dot{V}_{w,max}$ is the upper limit on water production.

Supplementary Note 7: Passive Cooling AWH Energy Consumption

For the passive cooling AWH system, there is no energy consumption, so the only quantity that must be determined from the analysis is the hourly water production. In the practical passive cooling system, air is cooled as it flows over a surface, and if the surface is cool enough water will condense out of the air. The surface radiates heat to outer space through the atmospheric window, allowing it to reach sub-ambient temperatures. For simplicity, we assume that the surface is uniform in temperature. The rate of heat transferred from the surface to space via radiation is then:

$$\dot{Q}_{surf} = \varepsilon_{AW} \sigma A_{surf} \left(f_{surf} T_{surf}^4 - f_{space} T_{space}^4 \right) \tag{S10}$$

where ε_{AW} is the emissivity of the radiative cooling surface in the atmospheric window (assumed to be unity), σ is the Stefan-Boltzmann constant, f_{surf} is the fraction of blackbody radiation that a body at T_{surf} emits in the atmospheric window wavelength range, and f_{space} is the fraction of blackbody radiation that a body at T_{space} emits in the atmospheric window wavelength range. It should be noted that Eq. (S10) assumes that the atmosphere is perfectly transparent in the atmospheric window of 8 – 11 μm, which is an unrealistic (but generous) assumption and is thus consistent with us making generous assumptions for the AWH systems to find the lowest possible cost they could achieve.

Even though the surface temperature is spatially invariant, the typical effectiveness-NTU method for heat exchange over a constant temperature surface cannot be used, as this requires the air to have a constant specific heat. Because water begins condensing out of the air at some point on the surface, the air effectively has a specific heat that varies along the length of the surface. To account for this, we discretize the surface into sections, treat the specific heat as constant across each section, and use the effectiveness-NTU method to find the total area required to bring the air to some outlet temperature T_o (Eq. (S11)).

$$\frac{A_{surf}}{\dot{m}_a} = -\frac{1}{U} \times \sum_{T_i = T_{amb}}^{T_o} c'_{p,a,i} \times \ln\left(\frac{T_i - T_{surf} - 1}{T_i - T_{surf}}\right) \tag{S11}$$

In Eq. (S11), we are summing the sections of area needed to drop the air temperature by exactly 1 °C, starting with the incoming ambient air and leaving at T_o. Then, the term $c'_{p,a}$ is the effective specific heat of the air for a 1 °C temperature drop at the given temperature. Initially, $c'_{p,a}$ is just equal to $c_{p,a}$; however, once the air reaches the dew point, $c'_{p,a}$ includes both the sensible heat capacity of the air and the latent heat from any water that condenses during the 1 °C temperature drop:

$$c'_{p,a,i} = c_{p,a} + h_{fg} \times \frac{\left(w_{T_i} - w_{T_{i-1}}\right)}{1\ °C} \tag{S12}$$

Finally, the rate at which the air is being cooled has both sensible (temperature drop) and latent (condensation of moisture) contributions:

$$\dot{Q}_{surf} = \dot{m}_a\left[c_{p,a}(T_{amb} - T_o) + h_{fg}(w_{amb} - w_o)\right] \tag{S13}$$

Using Eq. (S10) – (S13), the surface temperature and ratio of mass flow rate to surface area ($\frac{\dot{m}_a}{A_{surf}}$) can be found for a given outlet temperature T_o by solving the system of equations numerically. The hourly water production (in units of m^3 of water per m^2 of surface area per h) is then:

$$\dot{V}_{w,passive\ cool,h} = 3600 \times \frac{\dot{m}_a}{A_{surf}} \times (w_{amb} - w_o) \tag{S14}$$

In our analysis, we found the hourly water production for various outlet temperatures (effectively sweeping the mass flow rate) for each hour of the year. By choosing the outlet temperature that maximized the water production for each hour, we found the maximum annual water production per unit radiative cooling surface area in each location.

For the reversible passive cooling system, we assume that the surface provides cooling (at a temperature of T_{surf}) to a device that reversibly harvests water from air. The amount of water harvested per unit radiative cooling surface area is equal to the amount of cooling the surface can provide per unit surface area, multiplied by the amount of water that such cooling can reversibly produce:

$$\dot{V}_{w,passive\ cool,rev,h} = -\varepsilon\sigma\left(f_{surf}T_{surf}^4 - f_{space}T_{space}^4\right) \times \frac{1 - \frac{T_{surf}}{T_{amb}}}{R_w T_{amb}\ln(a_w)} \tag{S15}$$

The hourly water production is a function of the surface temperature and reaches a maximum value at some surface temperature between ambient temperature and the temperature of outer space (3 K). In our analysis we found the surface temperature that produced the maximum amount of water for each hour and used that to find the maximum annual water yield per unit surface area in each location.

Supplementary Note 8: Sorbent AWH Energy Consumption

For our analysis, we consider MOF-303 as the sorbent, so we use the energy consumption as determined by Li et al.[22] We found that the energy consumption is relatively insensitive to the ambient temperature in their model, so we use the regeneration heat from their paper (which corresponds to an ambient temperature of 25 °C) for every hour of the year in our analysis. For a regeneration temperature of 135 °C, this corresponds to $Q_{des,h}$ = 1077 kWh/m^3.

During desorption, we assumed that the moisture desorbed from the sorbent is at the inflection relative humidity corresponding to the desorption temperature (again found using the Clausius-Clapeyron relation). The moisture then condenses on a surface maintained at the ambient temperature. The hourly water production (in units of m^3 water per kg sorbent per h) is then:

$$\dot{V}_{w,sorb,h} = \begin{cases} \dfrac{w}{1000}, & RH_{amb} \leq RH_{infl} \\ 0, & RH_{amb} > RH_{infl} \end{cases} \tag{S16}$$

Where the factor of 1/1000 (specific volume of water) converts from kg to m^3. Though this ambient cooling process would require some surface on which the water would condense, we do not include the capital cost of such a surface in our calculation of sorbent AWH LCOW (another generous assumption). It should be noted that Eq. (S16) applies to the reversible sorbent system as well. Then, the specific energy consumption for the entire year is:

$$SEC_{sorb} = \frac{\sum_{h=1}^{8760} \dot{V}_{w,sorb,h}}{\sum_{h=1}^{8760} Q_{des,h}} \tag{S17}$$

The reversible specific energy consumption for a given hour of the year is simply found by multiplying the least work of separation (Eq. (1) of the main text) by a factor of $\left(1-\frac{T_{amb}}{T_S}\right)^{-1}$, where T_S = 473 K (based on an assumed regeneration temperature of 200 °C). Then, the total specific energy consumption across the entire year is simply found by averaging the hourly values:

$$SEC_{sorb,rev} = -\frac{R_w}{8760}\sum_{h=1}^{8760}\frac{\ln(a_w)}{\frac{1}{T_{amb}}-\frac{1}{T_S}} \tag{S18}$$

Supplementary Note 9: AWH Annual Yield and Water Storage Volume

From the hourly water production that was calculated for each hour of the year, the annual yield of water and water storage requirement can be found. To find the annual yield, the hourly water production is simply summed over the entire year:

$$Yield_i = \sum_{h=1}^{8760} \dot{V}_{w,i,h} \tag{S19}$$

where the subscript h corresponds to the given hour of the year. The required volume for water storage, V_{WS}, can be found by taking a cumulative sum of the net water production for each hour of the year and subtracting the maximum value of the cumulative sum by the minimum value. The net water production is simply the hourly water production from the AWH system subtracted by the hourly water consumption. If the rate at which water is consumed is constant, then the hourly water consumption is simply $\frac{1}{8760} \times \sum_{h=1}^{8760} \dot{V}_{w,i,h}$. Then, the required volume for water storage is given in Eq. (S20), where h_{max} is the hour of the year when the sum reaches a maximum value and h_{min} is the hour of the year when the sum reaches a minimum value.

$$V_{WS,i} = \sum_{h=1}^{h_{max}} \left(\dot{V}_{w,i,h} - \frac{1}{8760} \times \sum_{h=1}^{8760} \dot{V}_{w,i,h} \right) - \sum_{h=1}^{h_{min}} \left(\dot{V}_{w,i,h} - \frac{1}{8760} \times \sum_{h=1}^{8760} \dot{V}_{w,i,h} \right) \tag{S20}$$

For the two active AWH technologies (active cooling and sorbent), we explored the effect of implementing a “cutoff relative humidity”, below which the system would not be turned on. This would avoid running the system when AWH is energy intensive, but the penalty is that it shifts the water production to fewer hours of the year, which increases the need for water storage.

As such, there is some optimal cutoff relative humidity, which balances the benefits of avoiding low relative humidities with the benefits of consistent water production. For the reversible and practical active cooling and sorbent AWH systems, we found the optimal cutoff relative humidity and the corresponding minimum LCOW; the LCOW values presented in Fig. 4 and Fig. 5 of the main text correspond to these minimum values. For the reversible active cooling, this means we were optimizing two values: the cutoff relative humidity and the cutoff water production (above which all water would be discarded, so as to avoid excessive storage costs).

Supplementary Note 10: Sorbents vs Desalination with Variable Transport Cost

To complement Fig. 5 of the main text, we present Fig. S4, which shows the LCOW of the cheapest method (coastal desalination with water transport or practical sorbent AWH) depending on the water transport cost. When the water transport cost takes the base value (see Supplementary Note 1), sorbent AWH is cheaper for 0% of the WRW population. When water transport costs take 10× the base value, sorbent AWH is cheaper for 32% of the WRW population.

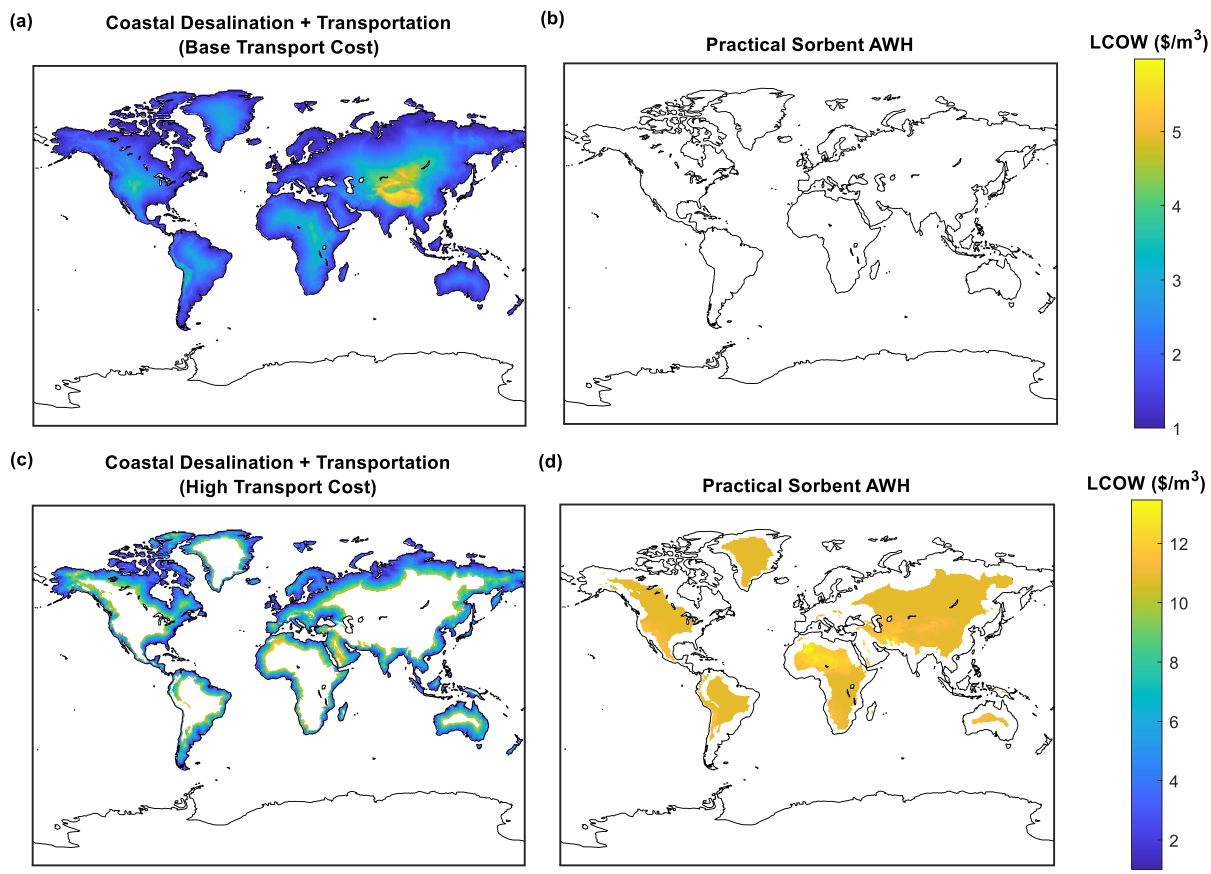

Fig. S4. Contour plot of the LCOW of coastal desalination with transport (a,c) and practical sorbent AWH (b,d). Base water transport costs were used when calculating the LCOW of desalination with transport in (a), while 10× the base water transport cost was used to generate (c). The contour plot in (b) is only filled where practical sorbent AWH is cheaper than coastal desalination with base cost water transport. The contour plot in (d) is only filled where practical sorbent AWH is cheaper than coastal desalination with 10× base cost water transport.

Supplementary Note 11: Fan Power in Passive Cooling AWH

In our analysis of the passive radiative cooling AWH, we considered a convection heat transfer coefficient of 10 W/m^2K between the ambient air and the cooled surface, which would likely have to be achieved by using a fan. The question remains, then, how much power would these fans consume? To answer this, one can first look at field data from air conditioners[23], which shows that the fans in cooling systems have an airflow efficiency of approximately 2 cfm/W (or 1.16×10^{-3} kg of air per J of electrical energy). Considering next the moisture captured by cooling ambient air (25 °C, 70% RH) down to 15 °C (3.3×10^{-3} kg of moisture per kg of air) and multiplying that by the airflow efficiency, a yield of 3.8 kg of water per J of electrical energy results. Rearranging to get energy per unit moisture harvested and converting the units, we get 73 kWh per m^3 of water harvested, which would add roughly $\$5/m^3$ to the LCOW (though this would vary depending on the ambient conditions). In addition, the airflow efficiency for air conditioners is affected not just by the flow resistance through the evaporator, but also the flow resistance through the ducting to deliver the airflow to the building (which would not be present in an AWH system); as such, the airflow efficiency of a standard fan might be more appropriate to use for AWH (which can be one or two orders of magnitude more efficient than in an air conditioner). Due to this uncertainty in airflow efficiency, and because we are trying to make the most generous assumptions possible for AWH (to see if it can reach parity with desalination even under the best assumptions), we neglected the airflow energy consumption.

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