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The Brinesiphon: A Homolog of the Thermosiphon Driven by Induced Salinity and Downward Heat Transfer

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The basis of a novel method for passive solar water heating homologous to the thermosiphon but driven by induced salinity, which causes a fluid to circulate without the need for a mechanical pump and with inverse natural convection (downward heat transfer), is outlined. The *brinesiphon*, like the thermosiphon, operates by harnessing the tendency of a less dense fluid to rise above a denser fluid, resulting in fluid motion through a collector, but with two exceptions: first, the buoyancy is controlled by induced salinity gradients rather than thermal gradients, and second, as a result, natural convection is in the opposite direction than that in the homologous thermosiphon concept; i.e., hot fluid flows down, and cold fluid rises. A *brinesiphon* may be more suitable for solar domestic water heating systems than the thermosiphon because the direction of flow allows downward transfer from a solar collector to a lower storage tank without any type of mechanical pumping system.

Keywords. *Thermosiphon, Salinity gradient, Thermal gradient, Domestic hot water, Water harnessing, Solar energy*

I. INTRODUCTION

Solar water heating (SWH) systems currently represent the most common application of solar energy. There are basically two types of water-heating systems: *forced circulation* systems, also called active solar systems, and *natural circulation* systems, also called thermosiphon or passive solar systems. There is an abundant literature on thermosiphon systems, and the fundamentals can be found in any book on solar heating systems (see, for example, [1] and [2]). However, all thermosiphon systems are based on the fundamental principle of buoyancy driven by a thermal gradient, i.e., harnessing the tendency of a less dense fluid that has been heated to rise above a denser fluid that is colder, resulting in fluid motion through a collector [3]. For this principle to operate, the water must be stored in a tank above the collector (upward heat transfer).

In this work, we outline the basis of a novel method for passive SWH that is homologous to the thermosiphon concept but is driven by induced salinity, which causes a fluid to circulate without the need for a mechanical pump, and, as a result, exhibits inverse natural convection (downward heat transfer). The *brinesiphon*, like the thermosiphon, operates by harnessing the tendency of a less dense fluid to rise above a denser fluid, resulting in fluid motion through a collector, but with two exceptions: first, the buoyancy is controlled by induced salinity gradients and not by thermal gradients, and second, as result, natural convection is in the opposite

direction to that in the homologous thermosiphon; i.e., hot fluid flows down, and cold fluid rises.

Figs. 1 and 2 show simplified sketches of a *thermosiphon* and the proposed *brinesiphon* system, respectively.

II. STATEMENT OF THE CORE IDEA

Let us consider Fig. 2 for the simplest analysis of a *brinesiphon* concept and Fig. 3 for its theoretical treatment. A fluid loaded with a certain salt content is circulated in a closed loop; the working fluid could be salt water or another fluid with a higher vapor pressure, e.g., a salty ammonia solution. This fluid is heated in a solar collector. As a result, a fraction of it is evaporated and sent to a condenser, and the remaining fraction (without evaporation) is sent to the other branch. Thus, we have an evaporated fraction, which is depleted of salt, and another fraction that is enriched in salt. Then, both branches are joined by a semipermeable membrane, which, owing to osmotic pressure, allows the flow of the low-concentration fluid (the evaporated fraction) to pass through the membrane and then dilute the non-evaporated fraction. Consequently, the mixture recovers its initial salinity.

In this simple scheme, two columns with different salinities, and thus different densities, are generated; from this hydrostatic imbalance, buoyancy appears, and a convective flow emerges, as shown in Figs. 2 and 3. Notice that this convection is in the opposite direction from convection driven by thermal gradients; i.e., hot fluid flows down, and cold fluid rises (if the

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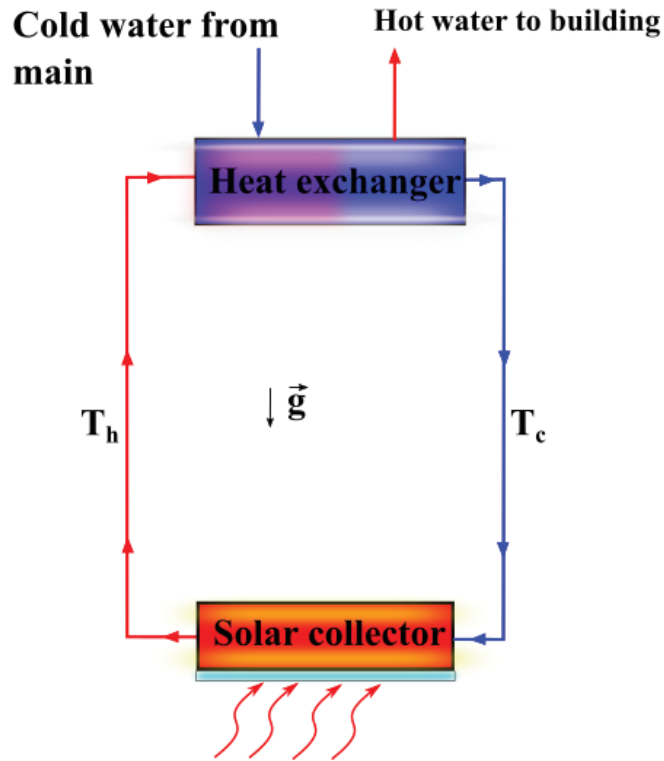


FIG. 1: Sketch of a thermosiphon.

effect of the salinity gradient overcomes the effect of the temperature gradient), so we have downward heat transfer rather than upward heat transfer such as that in traditional thermosiphons. Downward heat transfer can be much more attractive than upward heat transfer for domestic SWH systems because it makes it possible to transport solar heat from a roof to the interior of a house.

A. Calculations

First, let us consider Fig. 3 to develop our theoretical preliminary treatment; for the sake of generality, we take salty water as the working fluid. If we take the density of the warm water (after it is heated in the collector) as the reference density, ρ_o , and the reference salinity as s_o (in wt% of salt), then, after the non-evaporated fraction passes through the evaporator, it is enriched in salt content to a salinity s_2 , and the evaporated fraction is depleted to s_1 . Therefore, the density of the non-evaporated fraction is given by

$$\rho_2 = \rho_o + \nabla_s \rho \Delta s \quad (1)$$

where $\nabla_s \rho$ is the density gradient as a function of salinity ($\text{kg}/(\text{m}^3)(\%)$), and

$$\Delta s = s_2 - s_o \quad (2)$$

is the gain in salinity (percent), where s_2 is the new salinity, and s_o is the initial reference salinity.

This water, which has become heavier owing to its increased salinity, is gravitationally transported to the bottom of the system, where it is cooled and desalinated by mixing with the evaporated fraction, the salinity of which has been depleted to s_1 . For simplicity, we can assume that because of evaporation, the evaporated fraction is totally desalinated, i.e., $s_1 \approx 0$.

After both fractions, i.e., the evaporated and non-evaporated fractions, are mixed at the heat and mass exchanger, a process that is favored by the difference in osmotic pressure (forward osmosis), the salinity recovers to its initial value, s_o . However, because of heat transfer (e.g., for domestic hot water use in a secondary loop), the temperature of the mixture drops. As a result, the new density after the mixture passes through the heat and mass exchanger is given by

$$\rho_1 = \rho_o + \nabla_T \rho \Delta T \quad (3)$$

where $\nabla_T \rho$ is the density gradient as a function of temperature ($\text{kg}/(\text{m}^3)(\text{K})$), and

$$\Delta T = T_c - T_h \quad (4)$$

is the temperature difference between the inlet (hot) temperature T_h and the outlet (cold) bottom temperature T_c .

To move a volume of water around the system against

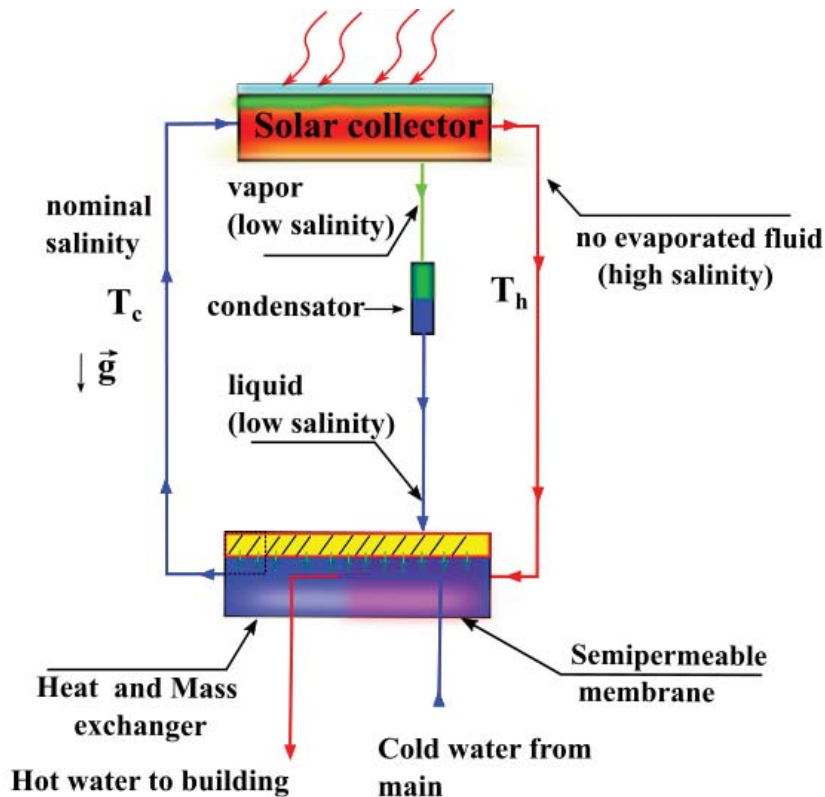


FIG. 2: Sketch of *brinesiphon*.

friction, the hydrostatic difference between the hot and cold water columns should compensate for the friction losses, and thus we have

$$(\rho_2 - \rho_1)gH > \Delta P_f \quad (5)$$

where g is the gravitational force, H is the effective vertical height of the pipes, and ΔP_f is the pressure drop due to the friction losses. This friction pressure drop may be calculated for preliminary assessment by the well-known Darcy–Weisbach equation [7], which is given by

$$\Delta P_f = \frac{8fL\dot{m}_w^2}{\pi^2\rho D^5} \quad (6)$$

where L is the total pipe length, f is the pipe's friction coefficient, \dot{m}_w is the fluid mass flow, D is the diameter of the pipe, and ρ is the average density of the fluid. Thus, Eq.(5) becomes

$$(\rho_2 - \rho_1)gH > \frac{8fL\dot{m}_w^2}{\pi^2\rho D^5} \quad (7)$$

On the other hand, the extractable energy can be expressed as a function of the fluid mass flow as

$$P \approx \dot{m}_w c_p \Delta T_p \quad (8)$$

where P is the power, c_p is the heat capacity, and $\Delta T_p \approx -\Delta T = T_h - T_c$, where T_h and T_c are the hot and cold temperatures, respectively. Thus, Eq.(7) may be rewritten as

$$(\rho_2 - \rho_1) > \frac{16fP^2}{\pi^2\rho D^5 g c_p^2 \Delta T_p^2} \quad (9)$$

Finally, the difference $(\rho_2 - \rho_1)$ in Eq.(7) may be evaluated from Eqs.(3) and (1), and then we obtain

$$(\rho_2 - \rho_1) = \nabla_s \rho \Delta s - \nabla_T \rho \Delta T \quad (10)$$

If we insert this into Eq.(9) and rearrange certain terms, we obtain

$$\frac{s_o}{s_2} = \left[1 + \frac{16fP^2}{\pi^2\rho D^5 g s_o c_p^2 \Delta T_p^2 \nabla_s \rho} + \frac{\nabla_T \rho \Delta T}{\nabla_s \rho s_o} \right]^{-1} \quad (11)$$

By considering the mass balance of salt at the evaporator, we know that

$$\dot{m}_w s_o = [\dot{m}_w - \dot{m}_e] s_2 \quad (12)$$

where \dot{m}_w is the water mass flow entering the evaporator, and \dot{m}_e is the evaporation mass rate; as mentioned

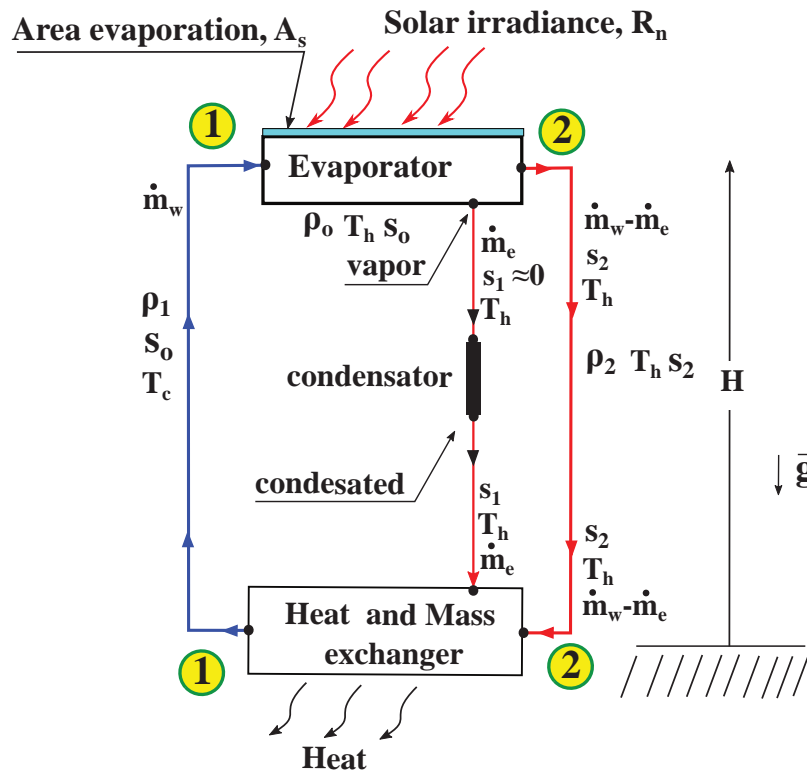


FIG. 3: Physical model for analysis of the *brinesyphon*.

above, the evaporated fluid can be considered to have zero salinity. We can rewrite Eq.(12) as

$$\dot{m}_e = \dot{m}_w \left(1 - \frac{s_o}{s_2}\right) \quad (13)$$

or, by taking into account Eq.(8), we obtain

$$\dot{m}_e = \frac{P}{c_p \Delta T_p} \left(1 - \frac{s_o}{s_2}\right) \quad (14)$$

If we insert this into Eq.(11) and rearrange the terms, we obtain

$$\dot{m}_e = \left[1 - \left[1 + \frac{16fP^2}{\pi^2 \rho D^5 g s_o c_p^2 \Delta T_p^2 \nabla_s \rho} + \frac{\nabla_T \rho \Delta T}{\nabla_s \rho s_o}\right]^{-1}\right] \frac{P}{c_p \Delta T_p u_w} \quad (15)$$

which allows us to calculate the required rate of evaporation as a function of the power.

The most important engineering limiting factor for practical domestic application is the actual surface area needed to generate the required evaporation mass rate, which for realistic domestic designs should be less than approximately 5 m².

The evaporator in the solar collector would need to be a closed-chamber solar evaporator system to facilitate the collection and separation of the vapor and concentrated brine streams. Hence, a rigorous treatment of the evaporation rate would need to be determined by low-pressure

steam boiler analysis. However, in view of several uncertainties of the model, let us use a simple correlation for atmospheric evaporation from water ponds; although it will not be entirely appropriate, it will enable an important assessment of the feasibility of the proposed concept for future research. Many semiempirical formulations of water evaporation as a function of the surface area are available, but the simplest expression, that of Shuttleworth [5], seems preferable.

$$\dot{m}_e = 1.15 \times 10^{-5} \left[\frac{m R_n + \gamma \delta_e (1 + 0.536 u_w)}{\lambda_v (m + \gamma)} \right] \times A_s \quad (kg/s) \quad (16)$$

where m is the slope of the saturation vapor pressure curve (kPa/K), R_n is the net irradiance (MJ/(m²)(day)), u_w is the wind speed (m/s), δ_e is the vapor pressure deficit (kPa), λ_v is the latent heat of vaporization (MJ/kg), A_s is the total surface area (m²), and γ is the psychrometric constant (kPa/K), which is given by

$$\gamma = \frac{0.0016286 \cdot p}{\lambda_v} \quad (kPa/K) \quad (17)$$

The vapor pressure deficit is given by

$$\delta e = (e_s - e_a), \quad \text{or};$$

$$\delta e = (1 - e_r) e_s \quad (18)$$

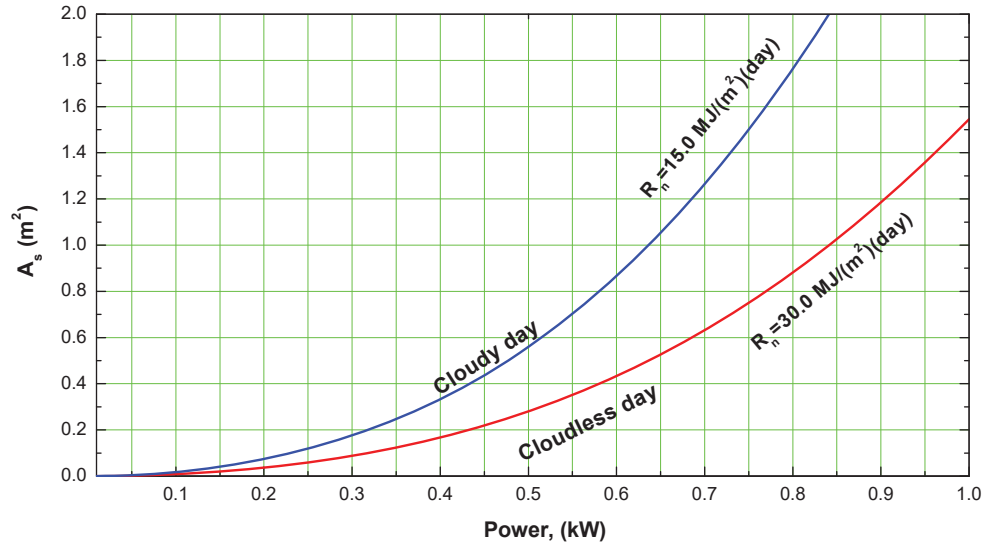


FIG. 4: Required dedicated area of evaporation as a function of power for some values of the solar irradiance.

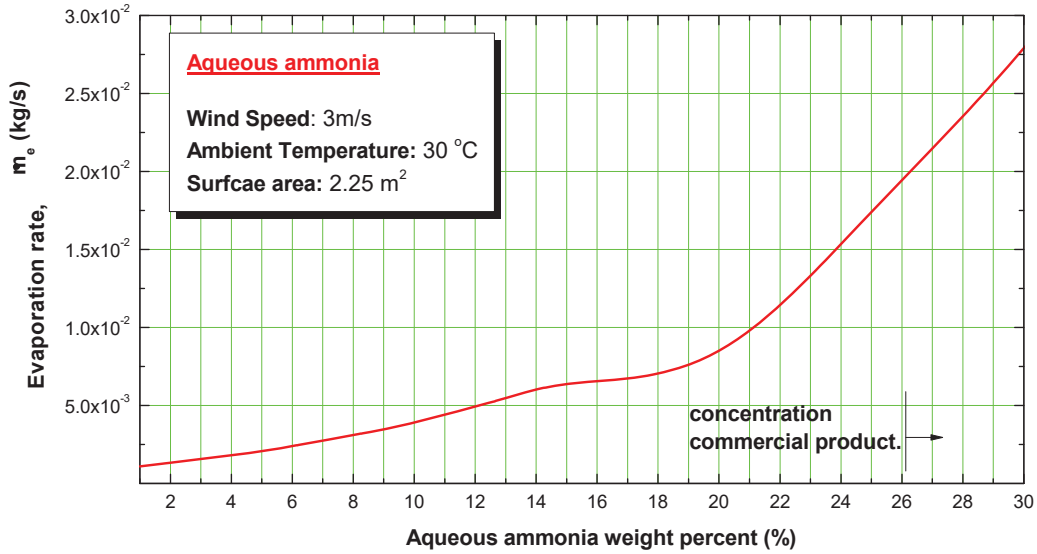


FIG. 5: Evaporation rate of ammonia solutions as a function of weight percent.

where e_s and e_a are the saturated vapor pressure of air and vapor pressure of free-flowing air, respectively. The saturated vapor pressure can be calculated as [6]

$$e_s = 0.13 \exp\left(21.07 - \frac{5336}{T_a}\right) \quad (kPa/K) \quad (19)$$

where T_a is the air temperature (K). Therefore,

$$m = \frac{de_s}{dT_a} = \frac{693.68}{T_a^2} \exp\left(21.07 - \frac{5336}{T_a}\right) \quad (kPa/K) \quad (20)$$

Then, from Eq.(16), the required surface area A_s as a function of the rate of evaporation is

$$A_s \approx \frac{4.34 \times 10^4 \cdot \lambda_v(m + \gamma)}{(mR_n + \gamma\delta_e(1 + 0.536u_w))} \cdot \dot{m}_e \quad (m^2) \quad (21)$$

where the rate of evaporation \dot{m}_e has been calculated previously and is given by Eq.(15).

• Discussion

To obtain some idea of the shape of the curves pre-

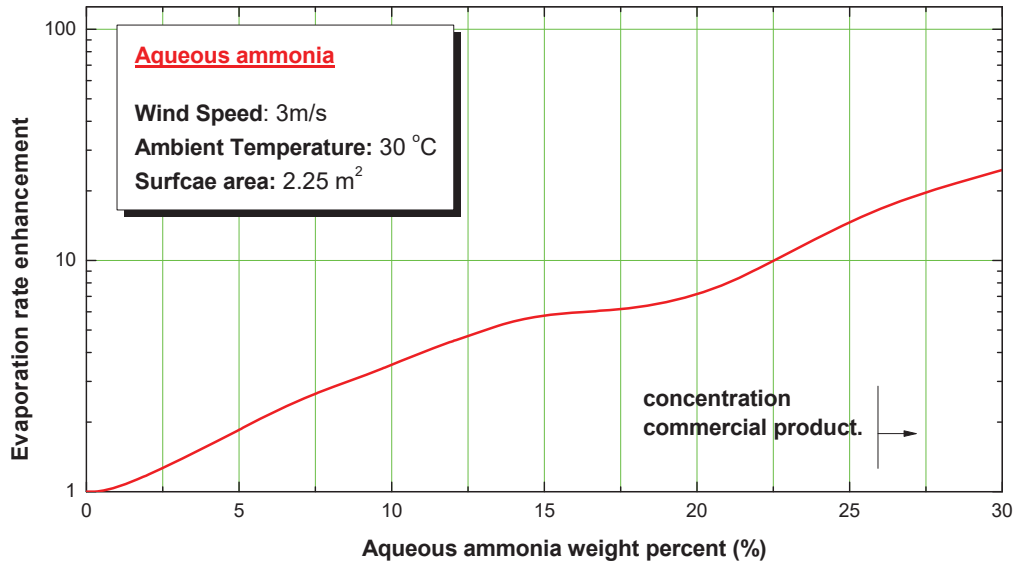


FIG. 6: Evaporation rate enhancement factor for ammonia solutions as a function of weight percent.

dicted by Eq.(21), we assume some typical values of the parameters: diameter $D = 0.02$ m; the reference salinity is similar to that of seawater, i.e., $s_o = 3.5\%$; $g = 9.8$ m/(s²); $c_p = 4.2 \times 10^3$ J/(kg)(K) and $\Delta T_p \approx 10$ K; $\Delta T \approx -10$ K; $\nabla_s \rho = 0.7$ kg/(m³)(%); $\nabla_T \rho = -0.13$ kg/(m³)(K); pipe friction coefficient $f = 0.012$; $H = 2$ m; $T_a \sim 298$ K; $m \sim 0.18$ kPa/K; $\lambda_v = 2.2$ MJ/kg; $e_r \sim 0$ (assuming continuous removal of vapor and maximum evaporation), and thus $\delta_e = e_s$ with $e_s \sim 3$ kPa; $u_w \sim 0.0$ m/s; $\gamma = 7.4 \times 10^{-2}$ kPa/(K), assuming an atmospheric pressure of 100 kPa. The resulting curves are shown in Fig. 4 for clear and cloudy days with 15 to 30 MJ/(m²)(day) (peak summer).

It is easy to see that if water is used, a thermal power close to 800 W could be transported for an average solar irradiance of approximately 20 MJ/(m²)(day) for a practical evaporative surface area of 1 m². This figure could be substantially improved by using a working fluid with a higher vapor pressure, which will result in a substantial reduction of the dedicated surface area needed. For instance, Fig. 5 shows the effect of dilution with a quantity of ammonia on the vaporization rate of water. The enhancement factor of the rate of evaporation is depicted in Fig. 6, and Fig. 7 shows the modified surface area required for evaporation as a function of power using an ammonia concentration of 5% per weight and assuming that the density of the solution is similar than pure water as well as its dependence with salt concentration and temperature. Referring to Fig. 7, for an evaporation area of approximately 0.5 m², we can obtain a value of approximately 800 W.

III. SUMMARY OF RESULTS AND CONCLUSIONS

A novel concept homologous to the thermosiphon but driven by induced salinity gradients, the *brinesiphon*, was proposed, and preliminary calculations were performed. The technology allows downward heat transfer, where hot fluid flows down and cold fluid rises, in contrast to the upward heat transfer in the classical thermosiphon. This type of technology could be more suitable for domestic solar applications in which solar energy is collected on roofs and then needs to be transported to the house.

NOMENCLATURE

A_s	= area of evaporation
c_p	= heat capacity
e_s	= saturated vapor pressure
e_a	= vapor pressure of free-flowing air
δe	= vapor pressure deficit
m	= slope of the saturation vapor pressure curve
\dot{m}	= mass flow rate
\dot{m}_{we}	= mass flow rate of evaporation
\dot{m}_w	= mass flow rate of seawater
\dot{m}_{w2}	= mass flow rate of salinized seawater
\dot{m}_s	= mass flow rate of salt (diluted into seawater)
R_n	= solar irradiance
s	= salinity (%)
T	= temperature
T_a	= air temperature at surface of seawater
T_c	= cold (bottom) temperature of seawater
T_h	= hot (surface) temperature of seawater
$\Delta T = T_c - T_h$	
$\Delta T_p = T_h - T_c$	
P	= power

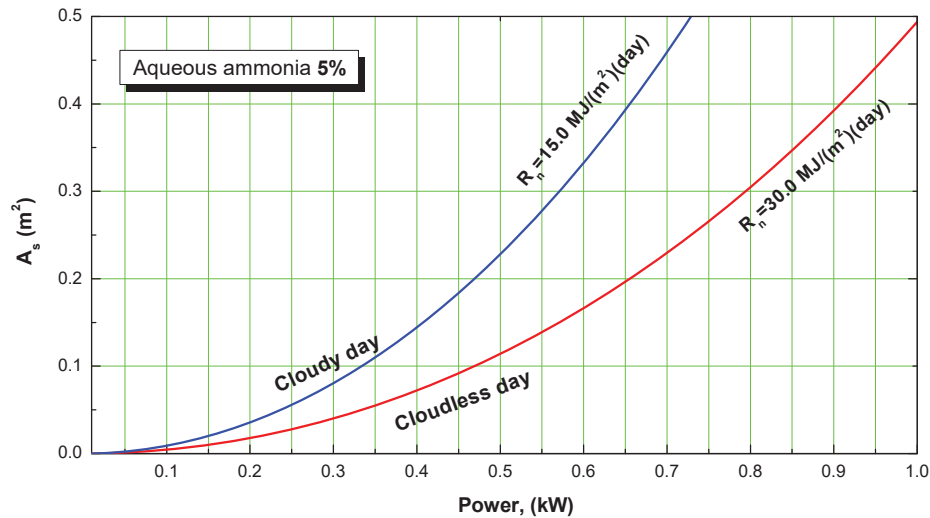


FIG. 7: Required dedicated evaporation area as a function of power for certain values of the solar irradiance with 25% ammonia.

Greek symbols

ρ = density of seawater
 ρ_o = nominal density of seawater
 ρ_2 = density of seawater after salinization
 ρ_1 = density of water at nominal salinization at the bottom of the sea
 γ = psychrometric constant
 λ_v = latent heat of vaporization

Subscript symbols

o = nominal, reference
 c = cold
 h = hot

1 = cold, bottom seawater
 2 = hot, salinized seawater

ACKNOWLEDGMENTS

This research was supported by the Spanish Ministry of Economy and Competitiveness under a Ramon y Cajal fellowship grant: RYC-2013-13459.

IV. REFERENCES

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- [1] Goswami D. Y. Principles of Solar Engineering. CRC Press. 2015
 [2] Kalogirou S.A. Solar Energy Engineering: Processes and Systems. Elsevier, 2013
 [3] Close, D.J. 1962. The Performance of Solar Water Heaters with Natural Circulation. Sol. Energy 6: 33.
 [4] Penman H.L. 1948. natural evaporation from open water, bare soil and grass. Proc. Roy. London A (194), S. 120-145.
 [5] Shuttleworth W.J. 2007. Putting the "vap" into evaporation. Hydrol. Earth Syst. Sci., 11, 210-244
 [6] Merva G.E. 1975. Physio-engineering Principles. AVI Publishing Company, Westport, CT.
 [7] Rouse, H. (1946). Elementary Mechanics of Fluids. John Wiley & Son