Stand-Alone Solar Desalination Plant

F. PICCININNI*, G. S VIRK†, T. SCIALPI*
* Department of Envinromental Engineering and Suistanable Development
Technical University of Bari
ITALY
http://www.poliba.it
† Massey University
NEW ZEALAND

Abstract: - The paper presents research to design and develop an efficient stand-alone modular desalination system able to provide sufficient non potable water to satisfy the needs of an average family of four living in Mediterranean regions. The system comprises a solar thermal collector for providing thermal energy for the desalination process and photovoltaic (PV) elements to provide electrical energy for the pumping operations. The analysis given here starts by presenting the functional scheme of designing a multi-stage flash (MSF) desalination thermal plant where the saltwater is warmed to about 90°C and supplied to several chambers in turn, where the pressure reduces sequentially from one space to the next. It is clear from the analysis that the plant requires large surfaces to provide sufficient evaporation of the seawater; in view of this, an alternative approach based on using several spray-nozzles together with only one flash chamber is proposed. This substantially improves the heat exchange between the seawater and the internal air for the evaporation of the saltwater. The electric power to pump the seawater supply through the solar collector and through the spray nozzle as well as the pumping requirements of the fresh water and brine extracting pumps will be supplied by PV panels. The surface area of the thermal collector will be about 6 m^2 with an optimal tilt angle for summer.

Keywords: Stand-alone desalination, MSF, spray-nozzle, spray evaporator, modelling, TRNSYS program.

1. Introduction

Providing fresh water to people living in large regions of Europe is increasingly becoming a major challenge due to global warming. The research presented here focuses on this important issue and is aimed at developing an efficient on-site desalination system for domestic applications where seawater is close at hand. The approach is to use conventional flash evaporation techniques coupled with renewable energy technologies integrated into the house to provide the thermal and electrical energies for local desalination of the seawater. In this way, more effective local management of thermal and electrical energies as well as water resources, may be achieved. The proposed study appears to be appropriate for the Mediterranean coastal zones, where climatic conditions are not very favourable. In particular, South Italy is classified by the Climate Carte, edited by CNDL 2000, as sub-humid and subject to severe droughts. This classification means that, in Mediterranean coastal zone of Italy, the arid conditions are not only due to the climatic conditions but are caused by the high water usage of the citizens and their activities. Fresh water demand for industrial, agricultural and societal uses is very high because the urgency of the water shortage has not yet been widely accepted or appreciated.

In order that fresh water continues to be provided for all these commercial and general societal needs as required, desalination of seawater is fast becoming a major strategy for long-term human survival in many parts of the world. Among all the seawater desalination processes, the multistage flash (MSF) process offers the most effective approach (4) and because of this, it is adopted in our research to design a low-cost *stand-alone* modular desalination system. The aim is to develop a small automatic system that can fulfil the potable needs of a family of four leaving in such coastal Mediterranean regions. The system also comprises renewable solar thermal energy technologies to power the MSF process as well as PVs to power the electrical energy for the system In MSP systems the saltwater is warmed up to about 90°C and supplied to several chambers in turn, where the pressure reduces sequentially from one chamber to the next (1). It is normal practise that the cold seawater is pre-heated by circulating it through a heat exchanger to condense the water vapour evaporated from the warm seawater to recover and re-cycle the thermal energy. In order to further warm up the temperature of the seawater it is possible to use various types of solar collectors; in particular evacuated-tube thermal solar collectors are known to give some of the highest output temperatures (11). It is well known that such a MSF plant will require fairly large dimensions (1) which is not practical for the domestic application that is being considered here. For this reason, a standard MSF desalination plant is not normally considered suitable for small stand-alone solar desalination situations. We propose to reduce the dimensions of the flash chambers using several spray-nozzles in order to increase the heat

and mass exchange between the seawater and the internal air to improve the evaporation of saltwater so to need only one flash chamber. All the electrical needs are supplied by a PV panel.
The other aspects needed for the standalone domestic desalination system are:
1) the development of an integrated sensors system;
2) the inclusion of a simple yet effective computing system for automatic monitoring and control of all the important variables of the plant and to allow the performance of the plant to be optimised. A simple PLC system is felt to be appropriate for our application to control all the thermodynamic variables (mass flow, temperatures, and pressures) as well as determine the appropriate control strategies to be applied for operating the valves and the water pumps.
The surface area of the thermal collector needed is expected to be about 6 m^2 with the tilt angle set for optimal summer operation. A spray-type evaporation chamber will be mounted below the thermal collector and the other components of the plant. Only three pipe are connected to plant: seawater supply, fresh water and brine discharge.

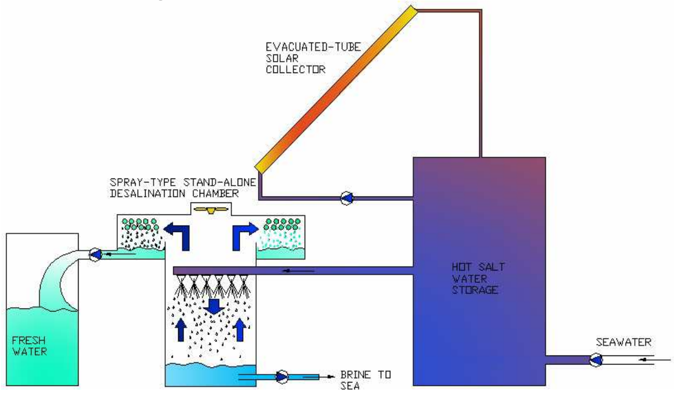

Fig. 1: Schematic of the stand-alone solar desalination plant.

2. Solar spray-type desalination design

In order to reduce the dimensions of the overall MSF solar desalination plant, we propose to use
1. Evacuated-tube solar collectors to maximise the exit seawater temperature, and
2. Spray-nozzles inside the flash evaporation chamber to increase the heat transfer process in the flash evaporation chamber.

A schematic of the proposed desalination plant is shown in Figure 1: the seawater passes first through the condenser tube inside the flash chamber, where water, evaporated in the evaporation section of the unit, is condensed, at the same time preheating the seawater. Subsequently, the seawater is directed either to an evacuated tube solar collector and then it is accumulated in a salt water storage tank from where it is directed to a set of spray-nozzles in the flash chamber so that the spraying can be carried out. The evaporated water will humidify the dry air introduced inside the chamber by a fan and as it passes through the condenser, it condenses generating fresh water.

3. Theory and model

3.1 Flash chamber thermal analysis
The flash distillation process involves boiling seawater and condensing the vapour to produce distilled water. It works on the principle that seawater will evaporate as it is introduced into the flash chamber with lower pressure than saturation pressure. It then condenses and cools down to a saturation temperature equivalent to the chamber pressure. This whole process is normally referred to as a stage, and the stage energy equations include balances for the flashing brine, the preheater/condenser unit as well as the overall energy balance. The energy balance for the flashing brine is:

$$VB_i\, \lambda_{Vi} = B_{i-1}\, Cp_B \left(T_{B\,i-1} - T_{B\,i}\right)$$

where the term, λ_{vi}, is the latent heat of the flashing vapour and is calculated at the vapour temperature, T_{vi}. This temperature is lower than the stage temperature T_Bi given by the thermodynamic losses, where

$$T_{B\,i} = T_{V\,i} + \Delta T_{LOSS} \text{ and } \Delta T_{LOSS} = BPE + NEA + \Delta t$$

The boiling point elevation, BPE, is dependent on the brine salinity and its boiling temperature. On the other hand, the non-equilibrium allowance, NEA, gives a measure of the flashing process thermal efficiency. The non-equilibrium allowance depends on the stage flashing range, Δt, the stage saturation temperature, T_{vi}, the flow rate of the brine recycle, the stage width, and the height of the brine pool. From the energy balance it is possible to evaluate the quantity of the flashing vapour:

$$W_i = \frac{VB_i}{B_{i-1}} = \frac{Cp_B \left(T_{B\,i-1} - T_{B\,i}\right)}{\lambda_{Vi}}$$

where the numerator is the brine enthalpy difference in the chamber, so the obtained vapour can also be calculated as follows:

$$W_i = \frac{HB_{i-1} - HB_i}{\lambda_{Vi}}$$

After its evaporation, the temperature of brine decreases ($T_{Bi}<T_{Bi-1}$) till it reaches the saturation temperature, corresponding to the value of the internal stage pressure. The temperature depression is caused by pressure losses in the demister pad and around the condenser tubes. The flashing vapour has a higher level of energy than the liquid phase, so, although the latent heat increases, the sensible heat decreases because the global energy of the system brine-vapour should remains constant. The mass balance in the evaporative section is given as:

$$B_i = B_{i-1} - VB_i\,,$$

which means that the outlet brine from the stage is less than the inlet brine in a quantity that is equal to the product vapour. The outlet brine has an higher value of salinity; in fact the salt mass is constant but the outlet brine mass flow rate is reduced with respect to the inlet brine mass flow rate in the evaporative section. The brine inlet in the stage can be considered to occur in an adiabatic, open and rigid system, so the enthalpy is constant.

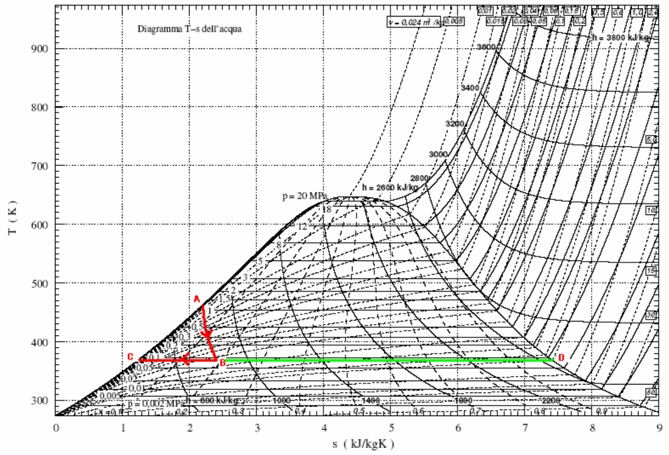

Fig. 2: Water temperature- entropy diagram: transformations of the evaporative chamber.

The transformation A→B is isoenthalpic and the process B→C represents the condensation of the produced vapour. The fraction between the segment BC and the segment CD represents the released vapour from the brine in the evaporative chamber.

3.2 Flash chamber dynamic mathematical model

A number of simplifying assumptions are adopted in the model development. These assumptions have a negligible effect on the accuracy of model predictions; however, they facilitate mathematical development, the solution procedure and reduce the overall computational effort. These assumptions are:

1. The distillate product is salt free; this assumption is valid since the boiling temperature of water is much lower than that of the salts;
2. The thermal losses with external environment are perfectly unimportant because the system is well isolated and works at low temperatures;
3. In the unit, the liquid phase is perfectly mixed so that it is possible to consider its characteristics (as temperature, concentration, specific enthalpy) equal to these of the same outlet liquid;
4. The steam released from the brine has not taken away any water drops;
5. The sub-cooling of the condensate or the superheating of the heating steam has a negligible effect on the system energy balance;
6. The power requirements for pumps and auxiliaries are not considered in the system analysis because it not influences the prediction of the performance ratio;
7. The heat losses to the surroundings are negligible, because the heavy insulation and the relatively low temperatures.

Evaporative stage: mass balance, salinity and enthalpy of brine

$$\frac{dM}{dt} = B_{in} - B_{out} - VB$$

$$\frac{d(M \cdot CB_{out})}{dt} = B_{in} \cdot CB_{in} - B_{out} \cdot CB_{out}$$

$$\frac{d(M \cdot HB_{out})}{dt} = B_{in} \cdot HB_{in} - B_{out} \cdot HB_{out} - VB \cdot HVB$$

$$M = \rho_B \cdot AS \cdot hB$$

Distillate product in the stage

$$D_{out} = VB$$

$$T_{D\,out} = T_{B\,out} - \Delta T_{LOSS}$$

$$\Delta T_{LOSS} = BPE + NEA + \Delta T_p$$

Distillate product in the stage

$$D_{out} = VB + D_{in}$$

$$T_{D\,out} = T_{B\,out} - \Delta T_{LOSS}$$

$$\Delta T_{LOSS} = BPE + NEA + \Delta T_p$$

Heat exchange pipes in the evaporative stage: mass balance, salinity and enthalpy

$$F_{in} = F_{out}$$

$$CF_{in} = CF_{out}$$

$$M_T Cp_F \frac{dT_{Fout}}{dt} = F_{in}(HF_{in} - HF_{out}) + 3600\,U\,A\,\Delta T_{ml}$$

$$VB(HVB - H_{Dout}) = 3600\,U\,A\,\Delta T_{ml}$$

$$M_T = \rho_F \cdot L \cdot AT$$

The thermodynamic properties, as enthalpy, specific heat, density and overall heat transfer coefficient, are calculated using the experimental equations.

3.3 Condenser pipe in the evaporative chamber

The vapour in the upper part of the evaporative stage, in contact with the condenser pipe, in which cold seawater is circulating, condenses giving up the following thermal energy Q_{cv}:

$$Q_{cv} = VB \cdot \lambda_v$$

The thermal energy lost by steam is absorbed by inlet condenser seawater, that has an increase of temperature, so its absorbed thermal energy is:

$$Q_{aF} = B \cdot CpF(T_{Fout} - T_{Fin})$$

$$Q_{aF} = Q_{cv} = Q$$

Knowing the exchanged thermal energy Q, the temperature differences is given from the following equation:

$$\Delta T = T_{Fout} - T_{Fin} = \frac{Q}{B \cdot Cp_F}$$

Using the method of logarithmic mean temperature differences, it is possible to evaluate the dimensions of the heat exchanger (condenser).

The effectiveness of the condenser is defined:

$$\varepsilon = Q/Q_{max}$$

where

Q_{max} = the maximum condenser exchangeable heat

Each heat exchanger has an efficiency, that is a function of the geometry and of the type of heat exchanger. The relationships of the heat exchanger efficiency include the dimensionless term UA/C_{min} that is called NTU. So, in case of phase change, the efficiency is:

$$\varepsilon = 1 - e^{-NTU}$$

3.4 Calculation of the condenser pipe overall heat transfer coefficient U

The seawater inside the pipe exchanges heat by forced convection, because seawater is pumped in by a circulation pump. The steam, that is in the external side of the condenser, exchanges heat by natural convection. The total resistance in the heat exchange is sum of the following resistances:

- external pipe natural convection, R_o;
- condenser wall resistance, R_w;
- internal pipe forced convection, R_i ;
- external and internal pipe sediment/corrosion resistance, R_{fo} and R_{fi} ;
- non condensable gas resistance, R_{nc} .

So, the condenser pipe *overall heat transfer coefficient U* is $1/(R_0 + R_w + R_i + R_{fo} + R_{fi} + R_{nc})$

.

3.5 Hydraulic spray-nozzle: droplet modelling

Spray nozzles are carefully engineered to deliver specific performance under certain operating conditions. Their performance is affected by the nozzle type, spray pattern, capacity, operating pressure, construction material, droplet velocity, and spray distribution, angle and impact. In a spray-type flash chamber there are three heat and mass transfer regions that need to be modelled, the <u>*spray region*</u> below the spray nozzles, the <u>*fill zone*</u> with compact film type fill and the <u>*rain region*</u> below the fill. In the spray and rain regions the water flows in droplet form. This has been represented with Lagrangian particle tracking with coupled heat and mass transfer between the droplets and the continuous phase. The change in droplet temperature and mass are found through equations (1) and (2), where N_i, the molar flux of mass and h the heat transfer coefficient are evaluated through empirical correlations. The letter "p" concerns the particle.

$$m_p c_p \frac{dT_p}{dt} = hA_p\left(T_\infty - T_p\right) + \frac{dm_p}{dt} h_{fg} \quad (1)$$

$$m_p\left(t + \Delta t\right) = m_p\left(t\right) - N_t A_p M_{w,t} \Delta t \quad (2)$$

The energy (Q) and mass (M) transfer are coupled with the continuous phase through the following equations:

$$Q = \left[\frac{m_p}{m_{p,0}} c_p \Delta T_p + \frac{\Delta m_p}{m_{p,0}}\left(-h_{fg} + \int_{T_{ref}}^{T_p} c_{p,t}\, dT\right)\right]\dot{m}_{p,0}$$

$$M = \frac{\Delta m_p}{m_{p,0}} \dot{m}_{p,0}$$

$m_{p,0}$, Δm_p and $m_{p,0}$ are the initial mass of the particle, the average mass of the particle in the cell and the initial mass flow rate of particles in the trajectory respectively. The water evaporated $m_{evap}(n)$ across fluid layer n is determined using equation (4), where $w_{sat,Tw}$ is the specific humidity of saturated air evaluated at the water temperature (kg/kg) and $w_{ave,fluid}$ is the average specific humidity in the fluid zone.

$$m_{evap} = h_d A\left(\omega_{sat,Tw} - \omega_{ave,fluid}\right) \quad (3)$$

3.5 Evacuated-tube collectors analysis

At steady state the useful energy output of a collector is the difference between the absorber solar radiation and the thermal loss. If I_β is the incident solar radiation on a solar collector tilted surface, T_p the average temperature of black plate, T_a the ambient temperature, q_p and q_a respectively the lost thermal energy and the absorbed thermal energy, it is possible to write:

$$q_p = A_c U\left(T_p - T_a\right) \text{ and } q_a = A_c\left(\tau\alpha\right)I_\beta$$

U-values can vary from a maximum of 8 W/m^2°C (very simple solar collectors) to a minimum of 2 W/m^2°C (very sophisticated one). For energy balance, and a steady state system it's a zero energy variation so the useful thermal energy, q_u, transferred to thermal fluid is:

$$q_u = q_a - q_p = A_c\left(\tau\alpha\right)I_\beta - A_c U\left(T_p - T_a\right)$$

Because this equation has a problem in the calculation of the plate temperature T_p, it is preferred to use the Bliss equation (4), that calculates the useful thermal energy through the inlet solar collector temperature, T_i:

$$q_u = A_c F_R\left[\left(\tau\alpha\right)I_\beta - U\left(T_i - T_a\right)\right] \quad (4)$$

q_u can also be written as function of the thermal fluid temperature difference in the black absorbing plate, and it is possible to write the energetic balance equation is:

$$q_u = \dot{m}\, c_p\left(T_u - T_i\right) = A_c F_R\left[\left(\tau\alpha\right)I_\beta - U\left(T_i - T_{amb}\right)\right]$$

Knowing F_R, $(\tau\alpha)$ and U and fixing (during project phase) the value of $\Delta T = T_u - T_i$ and the value of mass flow rate, m, it is possible to estimate the total surface area of solar collector, A_c.

4. Empirical considerations

4.1 Applicability of the proposed desalination plant

The following case study can be applied to all residential houses of the coastal towns, where there is a surplus of population in summer periods, increasing the consumption of potable water, not exclusively for the potable uses but also for all the other uses, as for the bidet, the wc, the garden, the dishes, the washing, the laundry, the car, etc. For the following research application, we have considered the case of Capo San Vito, a seaside small town near Taranto (Puglia, Italy). It has 1,000 people in winter and 5,000 people in summer. Considering the non-potable use of water, the surplus of non-potable water needs for Capo San Vito, the summer population is 112 m^3/d.

4.2 TRNSYS simulations

In order to design the test solar collector system, extensive simulation studies have been carried out. The performances of the energy supply plant have been investigated by the dynamic simulation code TRNSYS. It simulates the hourly performances by means of several FORTRAN subroutines, called types, which are linked together in order to model a thermal system. Simulations have been performed using Taranto's average climatic data, that are the input of weather Subroutine (type T89d).

The evacuated-tube solar collector thermal behavior is simulated in TRNSYS, utilizing type 71, while the hot seawater storage tank is simulated by type 4 (Tank) units. The hot seawater load to the desalination plant was simulated by type 14 (Load profile) and the Daily load equation. The circulation pump is simulated by type 3 blocks.

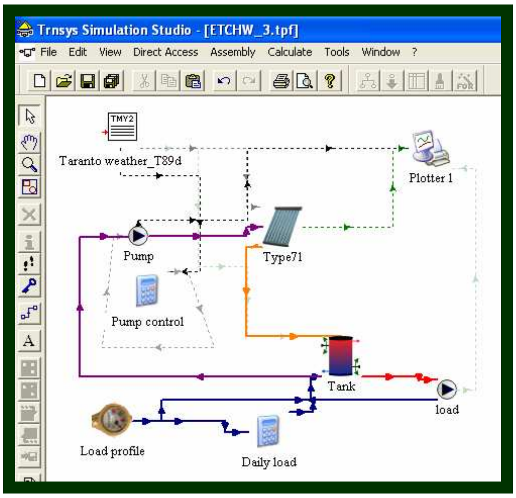

Fig. 3: The TRNSYS Simulation Studio

We have used the TRNSYS Simulation Studio to assess the evacuated-tube solar collector and the hot seawater storage tank, in order to obtain seawater at a temperature of about 90°C, to send in the spray-type flash chamber for the desalination. For four people family, 112 l/day, the hot seawater storage volume should be one cubic meter.

The results of the simulations are shown in the following Figure 4, in which are reported the outlet temperature of the evacuated-tube solar collector as function of its superficies versus the solar radiation. From these results, we finally assess the optimal solar collector superficies for our application, about 6 m^2.

4.3 Calculation results

In order to satisfy the summer surplus water needs, after several calculations, we have found the diameter of cylindrical flash chamber is 0.40 m

The evaporated mass flow all over the summer season (from June to August) is shown in Figure 5.

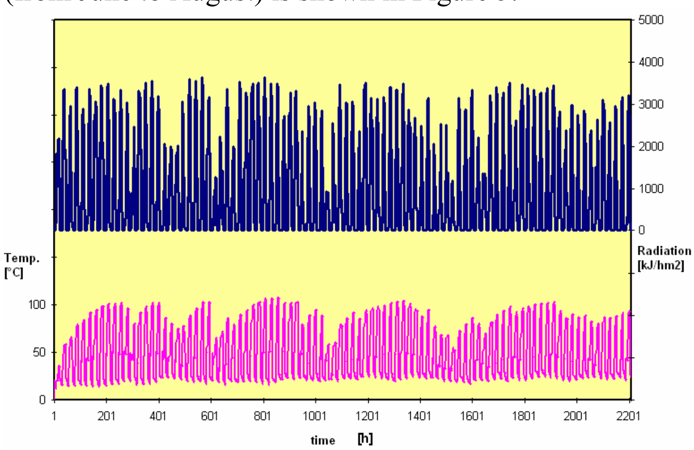

Fig. 4: Solar radiation (upper graph) and solar collector temperatures (lower graph)

The produced fresh water in the so dimensioned desalination plant is shown in the Figure 6: its variability principally depends on the climatic weather data, as the solar radiation, and also on the outlet temperature of the solar collector. During the summer season, it is calculated that the desalination plant can satisfy the 103% of no potable water need.

We modelled the designed spray-type desalination plant, considering the superficies of the evacuated-tube solar collector to satisfy the non-potable water need. From several calculations, the system can be modelled by an exponential equation,

$$\%\ W.N. = 0.2537\ S^{3.3949}$$

where

% W.N. is the percentage of satisfied water need [%];

S is the superficies of the evacuated solar collector [m^2].

5. Conclusions

The designed solar spray-type desalination system confirms the characteristics of a stand-alone, user friendly and PV powered thermal plant, as proposed.

Comparing the proposed desalination plant with the traditional three stage flash desalination plant it's possible to assert that the spray-type evaporator system is compact and much smaller in dimension.

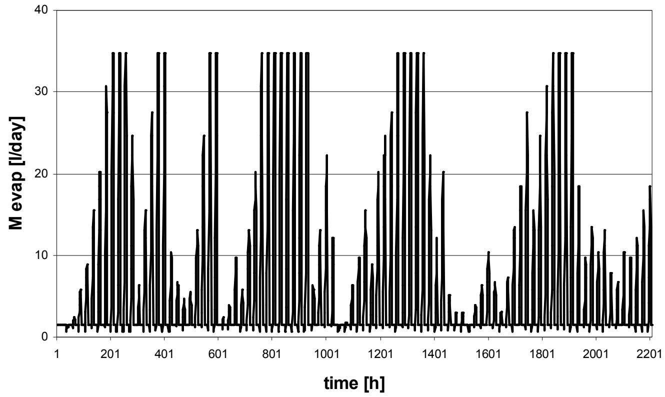

Figure 5: Evaporated mass flow in the spray-type flash evaporator

Besides, its components can be easily divided, compatibly with the external residential house's architecture.

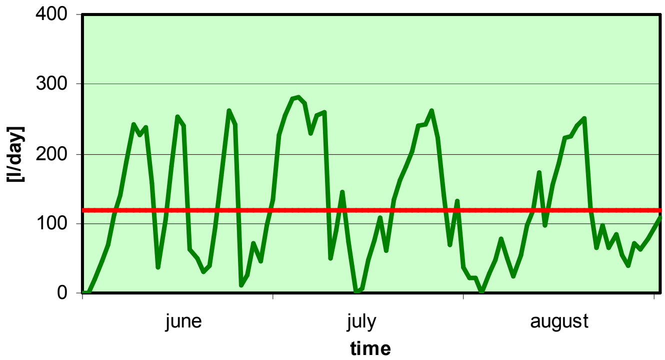

Fig. 5: Fresh water produced in summer (wavy line)

The simulations and the calculations made for dimensioning the two kinds of desalination plants are based on a seaside small town, but they can be easily applied also to other coastal towns. The heating of saltwater is supplied by 6 m^2 of evacuated-tube solar collector for both desalination plats. The 80 W electric power for seawater supplying, solar collector crossing, fresh water and brine extracting pumps will be supplied by PV panels.

This research has proposed to reduce the dimensions of flash chambers, using 9 spray-nozzles and one flash cylindrical evaporator. That improves the heat exchange properties of seawater with internal air for the evaporation of saltwater. The system is also composed by a hot saltwater storage tank of 1 m^3 and a fresh water storage tank of 0.5 m^3. The spray-type desalination plant satisfies the full summer non-potable water needs. Nowadays an experimental apparatus is building up in order to validate the calculations.

References

[1] Hisham T. El-Dessouky*, Hisham M. Ettouney, Yousef Al-Roumi, *Multi-stage flash desalination: present and future outlook,* Chemical Engineering Journal 73 (1999) 173-190

[2] Efat Chafik, *A new seawater desalination process using solar energy,* Desalination 153 (2002) pp 25-37.

[3] Isakson P., L.O. Eriksson, *A dynamic solar collector model for TRNSYS*, revised version of paper presented at Nordic Solar Energy R&D Meeting, Borlange. (1991).

[4] Khawla AbdulMohsen Al-Shayji, *Modeling, simulation and optimization of large-scale commercial desalination plants*, – Dissertation submitted to the Faculty of the Virginia Polytechnic Institute and State University

Nomenclature

A	*heat exchange area (m^2);*
AS	*Evaporative chamber area (m^2);*
AT	*Transversal area of heat exchanger tube (m^2);*
B	*brine flow (kg/h);*
CB	*brine salt concentration (kg/kg);*
CF	*Recirculating cooling water brine concentration (kg/kg);*
cp	*specific heat at constant pressure (kJ/kg K);*
D	*Distillate flow rate (kg/h);*
F	*seawater flow (kg/h);*
g	*Gravitational constant (m/ s^2);*
hB	*brine level in the evaporative chamber (m);*
HB	*Enthalpy of brine (kJ/kg);*
HF	*Enthalpy of recirculating cooling water (kJ/kg);*
HS	*Enthalpy of steam entering the brine heater (kJ/kg);*
HVB	*Enthalpy of vapour (kJ/kg);*
L	*heat exchanger tube length (m);*
M	*Mass of brine (kg);*
NTU	*number of thermal unit;*
P	*pressure (Pa);*
S	*Steam flow rate in the brine heater(kg/h) ;*
T	*temperature (° C);*
U	*Overall heat transfer coefficient (kW/m^{2}K);*
VB	*Vapour flow rate (kg/h);*
Z	*Orifice section (m^2);*
Δl	*difference in height between the levels in two consecutive stages (m);*
ΔT_{lm}	*difference of average logarithmic temperature;*
ΔT_{LOSS}	*temperature loss (° C);*
μ	*flux coefficient;*
ρ	*density (kg/ m^3);*

Subscripts

B	*brine;*
BH	*brine-heater;*
D	*distillate;*
F	*seawater in preheating;*
i	*ith stage;*
in	*input;*
m	*average between input and output:;*
out	*output;*
s	*vapour in the brine-heater;*
T	*heat exchanger tube;*
v	*brine vapour.*