Sigma Expert Solutions LLC

SIMULATION OF A SKID-BASED AUTOMATED FLASH EVAPORATOR SYSTEM

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Disclaimer:

This is a white paper used to propose a concept and does not represent a design calculation. The results presented are based on careful consideration of inputs, review of calculation results, and comparison with other data, but independent check and review has not been performed. Use of this information as a basis for decision making should only be made after independent verification of calculations and independent review.

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1. Introduction

1.1. Background on Problem

As fresh water supplies become increasingly scarce, the ability to generate or directly provide fresh, potable water for consumption is becoming a growing problem. This problem is especially concerning in situations where permanent water purification and generation systems are not feasible, such as during temporary interruptions of a fresh water supply or where access and resources are limited. In both cases, rapid availability of potable water is needed with a minimal amount of installed infrastructure.

1.2. Proposed Solution

A concept called the Skid-based Automated Flash Evaporator System (SAFES) was proposed as part of the American Made Prize Solar Desalination challenge sponsored by the Department of Energy to address the problem statement. SAFES uses multi-stage flash evaporation as its core technology, combined with centrifugal separation, carbon filtration, and UV sterilization to provide clean, potable water from water sources ranging from high salinity RO discharge streams to brackish groundwater sources. Energy for water purification is provided by portable high temperature solar water heating collectors and photovoltaic cells. A complete system containing all necessary equipment fits in a single ISO container for easy transport and storage. The proposal can be seen at www.sigmaexpertsolutions.com/design/desal.

1.3. Scope of this White Paper

To determine the feasibility of SAFES, a simulation was developed of the flash evaporator and auxiliary systems to determine water production rate, energy consumption, electrical power requirements, and space envelope. The proposal for the challenge does not document the simulation in detail, only providing results necessary to validate feasibility. This white paper focuses on the simulation and provides the detail and technical methods used to evaluate this complex thermodynamic system.

The original intent of the simulation was a feasibility assessment of SAFES, but the need to support an iterative design process resulted in a set of scripts with the flexibility to analyze a number of designs that were considered for the final product. Only the results from the SAFES analysis are shown in the proposal, but this white paper leverages the flexibility of the simulation to explore the potential for a miniature flash evaporator system beyond its use in the compact SAFES configuration.

1.4. Results and Conclusions

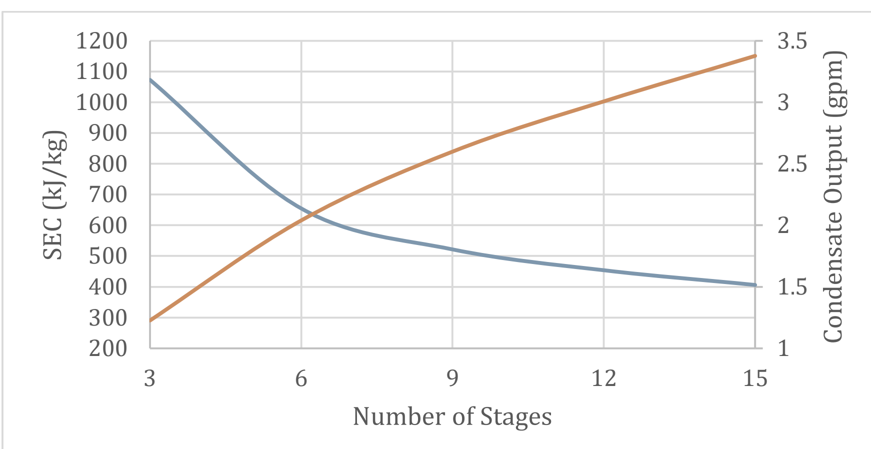
The simulation of the SAFES three stage system showed that a miniaturized flash evaporator is feasible. It produces water using only solar energy with the solar collectors and photovoltaic cells as proposed in the submittal to the American Made Prize Solar Thermal Desalination competition. A complete description of SAFES, some figures of components, and some of the performance data can be viewed in the proposal at the location listed above. SAFES was optimized for portability and ease of use, sacrificing efficiency and performance based on the availability of "free" energy from the sun. As a result, the design has high Specific Energy Consumption (SEC, amount of thermal solar energy consumed per kg water produced).

With the flexibility provided by the simulation, the sources of the inefficiency can be determined, and a more optimized configuration that is not bound by the transportation limitations can be proposed. Scoping studies using the simulation have determined that two factors are the primary drivers of the high SEC: the small number of stages and the low temperature of the brine to prevent scaling. Both were investigated separately.

In the figure below, the case of operating SAFES in Kinston, NC in the month of October with approximately 120m^2 of solar concentrator panels is the user input. This is the same case used for the American Made Prize competition. The SAFES simulation was modified by increasing the number of stages, up to a total of 15 to determine the impact on SEC and condensate generation. Increasing the number of stages with the design outlined in SAFES is completely realistic, as each stage is identical and stack as a unit. Increasing the number of stages only requires stacking more components and redoing some minor piping.

The figure shows that significant SEC savings is realized up until around 6 or 7 stages, with gains in thermal efficiency tapering off after that point. The fifteen-stage case, has an SEC typical for large scale facilities (between 200-400 kJ/kg). These types of facilities typically have upwards of 15 stages of heat recovery and additional heat rejection stages with brine recirculation. Even without these extra efficiency features, the design proposed for SAFES could be modified to approach the thermal efficiency seen in much larger facilities.

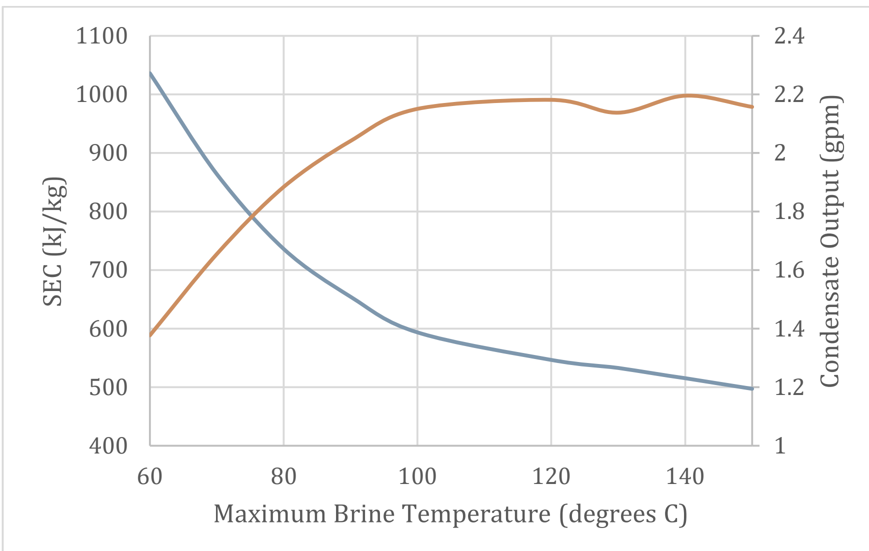
Of additional interest is the impact on water production. Water production increases dramatically as stages are added, with almost a linear increase after around 10 stages. At some point the flow limitation of the system would be reached, and pumping requirements would offset this gain in condensate output.



The next figure shows the impact of changing the maximum brine temperature. Six stages were used with the same environmental conditions previously mentioned. The only parameter changed in the simulation was the maximum allowable brine temperature.

The maximum brine temperature has a great effect on SEC, notably up until approximately 100°C. After that point, there seems to be relative insensitivity. There also seems to be little advantage to temperatures over 100°C for condensate production.

What is not shown in the graph is that the water recovery ratio significantly improves with increasing temperature (amount of condensate made as compared to total system flow) to almost 20%. This seems like a good attribute, but it would result in a higher concentration of contaminants within the stages and lowers the overall flow. This impedes the ability to flush out settled products – a critical aspect of the SAFES design. SAFES incorporates a holding tank with mechanical separation and recirculation, therefore the recovery ratio is not a design consideration other than for minimizing pumping power, which has not been of concern.

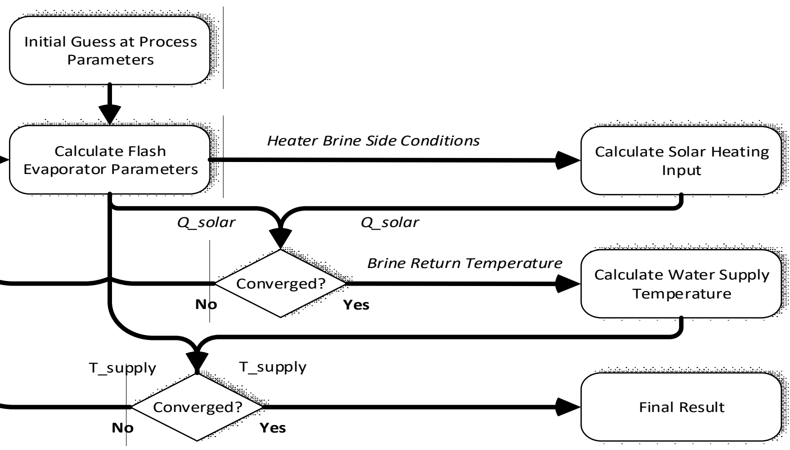


Overall, optimizing the design would require more extensive analysis of the heat exchanger to increase its efficiency and more refined control of the pressure within each stage. This could result in more efficiency, but the work of this white paper indicates that a miniature flash evaporator having six stages with a maximum brine temperature of 90°C is a good balance of efficiency, initial cost investment, low scaling potential, and condensate output. It is still possible to skid-mount such a system (the stack height we be around 2 meters), but the weight would be significant, and the easy portability of SAFES would be compromised. The opportunity to modify SAFES with additional stages in the field after installation could be explored as an “add-on” to significantly increase thermal efficiency and condensate yield.

2. ANALYSIS

2.1. Method of Analysis

SAFES was analyzed by building a simulation of the thermodynamic processes required for flash evaporation and water cooling. As all of these processes are interrelated, a coupled model was built that was iterated to find convergence between all of the interfacing systems at a given operating condition. The basic approach is summarized in the flow-chart below.



Each of the boxes are explained in detail below, along with applicable thermal models used. What is not shown is that the user defines the operating conditions used to evaluation each simulation. A batch input file is used for this purpose and a sample is shown below.

```
#Input file for initial simulation test
#Inputs are in the following order, separated by commas and no space
#T_amb,humidity,incidence,npanels,aperpanel,CFM,atmpress,eff,T_elev,T_heating
#First simulation is After Hurricane Matthew (Kinston, NC October 2016)
25.6,.3,1000,50,2.41,5500,101325,.9,5,90
25.6,.3,1000,75,2.41,5500,101325,.9,5,90
```

The following items are part of the user input:

- T_amb – Dry bulb temperature for the simulation
- humidity – Relative ambient humidity for the simulation
- incidence – Solar incidence. 1000 W/m^2 is used. To determine the water production rate at a specific location, the equivalent full sun hours are used based on this value.
- npanels – The number of modeled solar collectors.
- aperpanel – Surface area of each solar collector.
- CFM – The airflow rate capability of the fan on the evaporator system.
- atmpress – Atmospheric pressure for the simulation

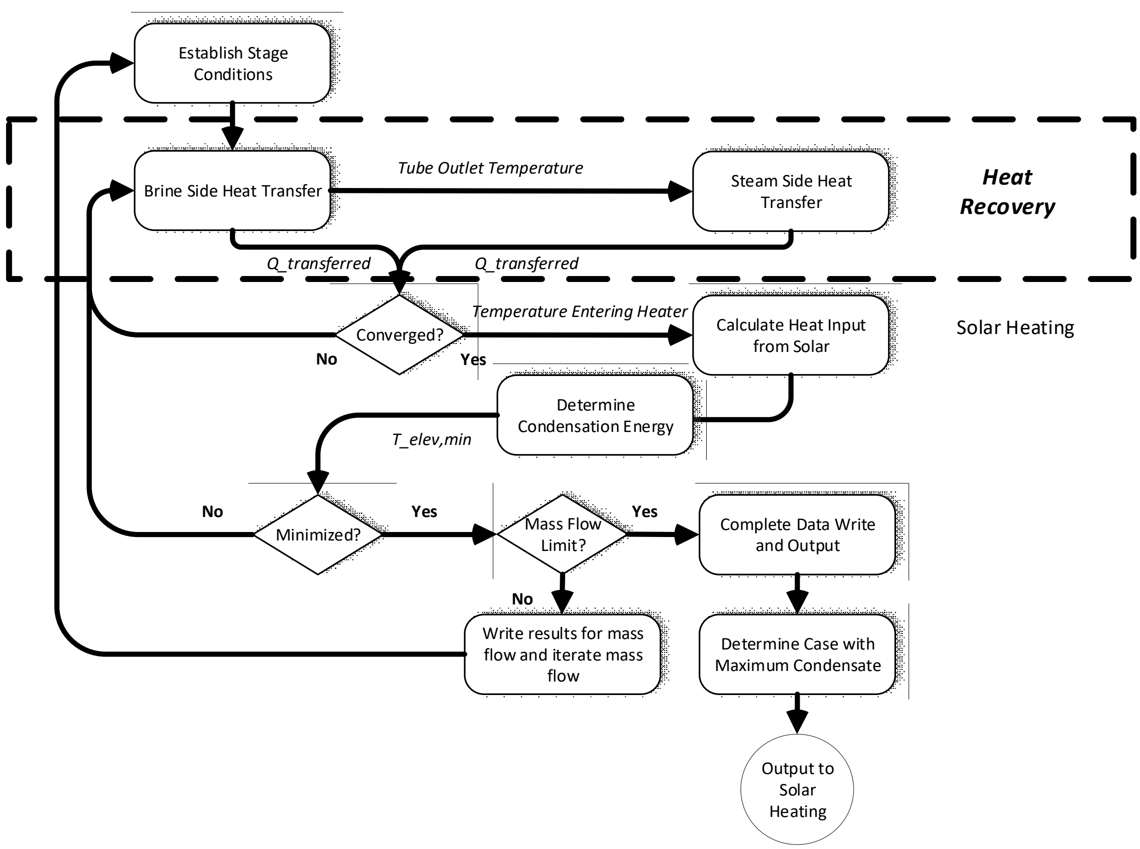
- eff – Maximum anticipated relative humidity achievable by the evaporative cooling system. Used as a proxy for the efficiency of the fill.
- T_elev – The temperature elevation required in a stage to drive the flashing process.
- T_heating – The maximum temperature allowed of the contaminated feedwater prior to entering flash stages to minimize scaling.

2.2. Initial Guess at Parameters

An initial guess at the inlet temperature to the flash evaporator and the amount of heat consumed is made to start the iterations. These values are updated during iteration and their selection only impacts the speed of convergence and has no impact on the final result.

2.3. Calculate Flash Evaporator Parameters

The flash evaporator calculation is performed at a range of feed water flow rates (contaminated inlet water) starting from the lowest viable flow up to where there is a hydraulic limitation on the flow rate. For each flow scenario, an energy balance is achieved between the heat recovery side of the evaporator (inlet water flow) and the heat rejection side (condensing steam). The flowchart below shows this process simplified.



2.3.1. Establish Stage Conditions

Initially the feedwater flow is set to the minimum allowable that can absorb all of the potential heat input from the solar concentrator. The maximum solar energy input is the "irradiance" user input multiplied by the solar panel area and number of panels. This quantity is divided by the heat capacitance of water multiplied by the temperature difference from the maximum allowable brine to the feed inlet temperature.

$$\dot{m} = \frac{Q_{\max\_solar}}{C_p(T_{max} - T_{in})}$$

The mass flow will be incremented after the entire flash evaporation stages have reached equilibrium until a flow limitation is met (addressed later).

The other stage condition established is the pressure for each stage. This is dependent on two factors: the temperature rise in each stage, and the ratio of heat added through recovery vs. solar heating.

To allow for the stage heat transfer surfaces to be the same size, each must transfer approximately the same amount of heat. This implies that the temperature differential on the brine side should be approximately equal for each stage. At the start of a flow case evaluated an initial guess is that each stage only accounts for one degree of temperature rise, with the rest provided through solar heating. This defines the target outlet temperature of each stage. As an approximation, the saturation pressure associated with this outlet temperature is assigned to the applicable stage. The simulations show this approximation is close enough, and has no implication on the correctness, but further refinement would increase the efficiency of each stage by tailoring the stage pressure to account for incomplete heat transfer and variations in heat capacitance. It is anticipated that the gain is minimal, and the extra effort is not justified.

The ratio of heating between recovery and solar is dependent on an energy balance with the condensing steam, which is unknown. The temperature differential is incremented in a flow case until equilibrium conditions are established.

2.3.2. Heat Recovery Section

The heat recovery section is modeled as a single pass shell and tube heat exchanger. The tubes are connected together by a series of plates (fins) to increase the steam side heat transfer area. The brine enters the tube bundle on the inside. Steam condenses on the outside surfaces.

The heat transfer coefficient on the outside is based on a steam condensation correlation for vertically arranged tubes:

$$h_{out,tube\_only} = 0.729\left[\frac{g\rho_l(\rho_l - \rho_v){k_l}^3 h'_{fg}}{N\mu_l(T_{sat} - T_s)D}\right]^{1/4}$$

The surface temperature was estimated as the average of the brine side liquid temperature. The heat transfer coefficient was modified to account for the presence of fins. First the efficiency of the fins was calculated as:

$$\eta_{fin} = \frac{\tanh{(mL)}}{mL}$$

where:

$$m = \left(\frac{2h}{kt}\right)^{1/2}$$

The total efficiency of the outside surface based on total heat transfer area (including fins) was calculated as:

$$\eta_0 = 1 - \frac{A_{fin}}{A_{total}}(1 - \eta_{fin})$$

The heat transfer coefficient on the inside of the tube was determined using the Dittus-Boelter correlation:

$$Nu_D = 0.023 Re_D^{4/5} Pr^{0.4}$$

and

$$h_{in} = \frac{Nu_D k}{D}$$

Combining the interior convection coefficient, and exterior convection coefficient with fin efficiency results in a total convection coefficient for the heat exchanger. Note that it is dependent on temperatures in the brine and steam, so is recalculated every time based on updated values.

This convection coefficient is used to determine how much heat can be transferred from the steam to the brine using the saturation temperature as the hot side and the average brine side temperature as the cold side. The amount of heat gain of the brine side is independently calculated based on the temperature differential from the inlet and outlet, mass flow rate of the brine, and heat capacitance of the water. If the two heating values do not match, the brine outlet temperature is updated and the calculation rerun until convergence is met.

2.3.3. Calculate Solar Heat Input

After heat input from the recovery stages, heat input is performed in the solar heater. The solar heater provides as much heat as it can, up to the maximum brine temperature. For low flows, this results in limiting the solar heating to well below what is available as heat recovery is increased during iteration. For high flows, the availability of solar heating is the limitation. As an initial estimate, 30% of the total incident energy available is assumed available to heat the brine. This value will be compared against the solar heating model discussed in the next block and drive further iterations to establish convergence.

2.3.4. Determine Condensation Energy

Condensation energy is calculated as the water flows back through the heat recovery stages. During the heat recovery calculation, the amount of energy transferred to the brine was calculated. In this calculation, the goal is to optimize the energy recovery, so the outlet temperature of each stage is targeted to be equal to the stage saturation temperature plus some user defined temperature elevation. For each stage, the temperature inlet for the evaporating brine is defined by either output from the solar heater or the previous stage. The outlet temperature is determined by removing the energy from condensation calculated during heat recovery. After calculation of all the stages, it is checked that all stage outlet temperatures are at or above the elevation temperature limit. If the recovery has not been maximized (at least one stage at the elevation limit) the calculation is restarted at the heat recovery section with an updated ratio of heat input from recovery and solar heating until the condition is met.

2.3.5. Check the Mass Flow Limit

The initial mass flow is equivalent to the minimum that would be able to absorb all available solar heat energy if there was no heat recovery. This is iterated up throughout the analysis until a flow limitation is determined. A flow limitation is determined by calculating the flow losses through the stages and comparing that with the pressure differential between the stages.

The flow path through the flash evaporator starts at the top of the stacked assembly, with brine pumped through the tube bundles, down through successive stages and through the solar heater. A centrifugal pump is used for this service, and its power requirement calculated, but this is not considered a limitation.

After exiting the solar heat exchanger, the water enters a plenum that communicates with the stage at the bottom of the stack which has the least vacuum. The evaporating brine is drawn vertically through the stages as pressure decreases (vacuum increases) against flow resistance and vertical head.

The total flow resistance for a stage was developed using the solid model as a basis and breaking it down into a series of pipes and plenums. That calculation is not shown as it is specific to the stage assembly and beyond the scope of this discussion. Combined with the head loss due to elevation, the total differential pressure required to drive the flow was determined. This was compared with the difference in pressure between the stages. Note that a process control system controls pressure at each stage individually, and the design of the flash evaporator has a water seal between each stage to prevent pressure equalization.

For each flow rate, the required pressure and the pressure drop due to flow and pressure head was compared. As the flow rate increases, so does the pressure drop of the system, until a flow limit is reached where the differential pressure between the stages can no longer drive flow. At this point the flash evaporator calculation is terminated. The same check is not performed for the condensate side as the flow losses were found to be significantly lower and not the limiting factor.

2.3.6. Determine Maximum Condensate Condition

The flash evaporator model determines the steady state condition at every potential mass flow through the system. For purposes of this calculation, only the mass flow resulting in the maximum amount of condensate is of interest. This result, and associated flash evaporator conditions are passed to the solar heating calculation.

2.4. Calculate Solar Heating Input

Solar heating input is through a counterflow hairpin style tube in tube heat exchanger. The brine flows through the interior tube and clean, solar heated hot water flows in the external tube. The model uses standard Dittus-Boelter correlations to determine the heat transfer coefficients on both sides of the tube wall to develop the overall heat transfer coefficient (neglecting the thermal resistance of the tube wall).

The calculation uses as an input the heat exchanger inlet and outlet temperatures, the heat transferred to the brine, and the mass flow rate of the brine from the flash evaporator calculation. The overall heat transfer coefficient, and an assumed solar side mass flow (defined as constant in the code), are used to determine the inlet and outlet temperature of the solar heated water side.

The inlet and outlet temperatures, solar panel efficiency, and ambient conditions are used to determine the real heat input potential of the solar panels. The efficiency is based on two difference sources. The first is from a study by the National Renewable Energy Laboratory (NREL), “High Performance Flat Plate Solar Thermal Collector Evaluation”, Figure 15. The other source is the technical information sheet for the Arctic Solar XCPC collector. The NREL data is used only to establish a slope for efficiency as a function of temperatures. In the panels investigated, there were many different results, but this slope was relatively consistent. As an approximation, it was assumed to even be extendable to other types of solar collector technologies. The data point from the XCPC description was vague and stated over 50% efficiency at 200°C. Assuming ambient conditions of 25°C and irradiance value of 1000 W/m^2 we can establish a linear relationship between solar collector efficiency and heated fluid temperature.

The heat input from the solar collectors is compared with the heat input assumed for the flash evaporator. If they do not match, the entire calculation is iterated from the beginning, updating the solar heating.

2.5. Calculate Water Supply Temperature

After convergence of the flash evaporator and solar concentrator models, the brine output of the flash evaporator is sent to an evaporative cooler model. The evaporative cooler consists of a spray header, ducted fan, and some fill over the top of the holding tank of the system. The fan is based on a standard jet duct fan available on McMaster-Carr in terms of airflow and power requirements. The fill is not based on any specific product, but is envisioned as a relatively foul resistant synthetic variant.

The evaporative cooler model determines the amount of heat rejection to atmosphere based on evaporating water to ambient air with given humidity to a final humidity called the "efficiency" in the code. Typically, a value of 90% relative humidity is used as this input to represent the ability for the fill and fan combination to saturate the airflow with water vapor. Additional studies on the design of the fill and its relation to the airstream would be required to establish a more appropriate value.

A simple energy balance is performed to determine the amount of cooling. The energy into the system is characterized by the energy carried by the incoming ambient air and humidity and the brine reject from the flash evaporator. The energy out of the system is the exhausted air at the maximum allowed relative humidity (the "efficiency") and the water going to the holding tank.

$$E_{in} = \dot{m}_{brine} h_{f,brine} + \varpi_{amb} \dot{m}_{air} h_{g,vapor} + \dot{m}_{air} C_{P,air} T_{amb}$$

$$E_{out} = \dot{m}_{to\_sump} h_{f,to\_sump} + \varpi_{eff} \dot{m}_{air} h_{g,vapor} + \dot{m}_{air} C_{P,air} T_{out}$$

The air outlet temperature is initially a guess when performing the energy out equation, and it is iterated until the two equations converge. This results in determining the amount of water lost to evaporation as well as the temperature of the water returning to the holding tank. A simple energy balance on the holding tank with incoming flows of the return from the evaporative chiller, and makeup water (equaling that lost by evaporation in the cooler and in the flash evaporator) at ambient temperature determines the new holding tank water temperature. This is compared with the water temperature used as the inlet to the flash evaporator. If they are the same, the calculation is complete, if not, the flash evaporator inlet temperature is updated, and the entire calculation starts over until flash evaporator inlet temperature converges.

2.6. Final Result

The output of this iterative process is a steady-state condition for the entire system, inclusive of the flash evaporator, solar heater, and evaporative cooler. Process conditions that produce the maximum condensate water are retained from each model. This information can be used to develop the final output.

Final output is tailored around proving feasibility and efficiency of such as system. To that end, the water production rate, energy consumption, as well as information on necessary process conditions are sent to an output file for review. Post processing of this information is performed in a spreadsheet for further data manipulation and to create graphs.

These results are available for every simulation run in the input data file. A snapshot of the output data file is shown below.

```
Results for simulation #1

System Feed (kg/s): ..................................1.1818890450662296
Stage 1 Condensate Flow (kg/s): ......................0.0291230140372427
Stage 2 Condensate Flow (kg/s): ......................0.02956258245670928
Stage 3 Condensate Flow (kg/s): ......................0.030045203913472946
Stage 1 Properties (Temp(C),Press(Pa)): ..............[41.96666666666667, 8252.873333333335]
Stage 2 Properties (Temp(C),Press(Pa)): ..............[56.13333333333333, 16705.92]
Stage 3 Properties (Temp(C),Press(Pa)): ..............[70.3, 31633.399999999994]
Stage 1 Heat Transfer Rate (W): ......................69954.04276239502
Stage 2 Heat Transfer Rate (W): ......................70002.26383543373
Stage 3 Heat Transfer Rate (W): ......................70097.14326155154
Heat Provided by Solar (W): ..........................95106.17326261985
Water Temperature from Pre-treatment (C): ............27.8
Brine Temperature to Pre-treatment (C): ..............48.17057436693803
dP during Heat Recovery (Pa): ........................135.50635632546246
dP during Evaporation (Pa): ..........................8774.191672382629
Temperature into Solar Heater (C): ...................70.3
Temperature out of Solar Heater (C): .................89.50484126585268
dP on the Solar Heater Brine Side (Pa): ..............89.54347562330933
dP on the Solar Heater Process side (Pa): ............623.2476689044441
Heat Rejection Rate (W): .............................100
Temperature to Sump (C): .............................100
Water Evaporated (kg/s): .............................100
Recirculation Pump Power (W): ........................24.08444134805597
Fan Power (W): .......................................373
Booster Pump Power (W): ..............................0
Feed Pump Power (W): .................................0.3732466939654851
Vacuum Pump Power (W): ...............................69.11149433333333
Brine Pump Power (W): ................................209.00774691697538
Condensate Pump Power (W): ...........................120
Control Power (W):....................................25
Pre-treatment Skid Auxiliary Power (W): ..............20
Purification Skid Auxiliary Power (W): ...............50
Total Electrical Power Requirement (W): ..............661.5691823753548
Total Condensate Output (kg/s)/(gpm): ................0.08873080040742493
Total Condensate Output (gpm): .......................1.4064115803138155
```

The file represents the thermodynamic conditions, but also the pumping and electrical system power requirements. These were needed for the feasibility determination. Some of the outputs are not used and have placeholder information. Pumping power for centrifugal pumps is determined using the pump pressure differential, mass flow rate, and an efficiency of 75%. The vacuum pump and condensate pumps are positive displacement. The former is based on a manufacture's data where the minimum pressure of a stage drives power requirements. The latter is the maximum power at maximum condensate flow. Fan power is approximated as the maximum power of the attached motor. Control and auxiliary power is an estimated guess based on the number of controlled valves and powering a micro-controller.