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CFD based analytical approach of the annual performance of a solar absorption chiller with an underground cold thermal storage and an indirect seawater cooling system.

Keywords: Solar absorption chiller, underground thermal storage, indirect seawater cooling system, evacuated tube solar collector, CFD.

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Introduction

The IPCC Fourth Assessment Report (AR4) concluded that "Most of the observed increase in global average temperature since the mid-20th century is very likely due to the observed increase in anthropogenic greenhouse gas concentrations."

The residential sector represents 27% of global CO2 emissions, as well as 17% of global energy consumption, having great potential on the mitigation of climate change. In this sector, space heating and cooling represent 43% of total primary energy use, making renewable energies an important countermeasure for the reduction of emissions. If implemented properly, renewable energies can also contribute to social and economic development, energy access, a secure energy supply, and reducing negative impacts on the environment and health.

A significant drawback of renewable energies is that most of the time they are not cost effective, making them unattractive for the consumer. Therefore, often innovative and creative ways of using the available resources is necessary.

There are many cooling technologies available in the actuality, but the vast majority of the implemented systems are traditional vapor-compression air conditioners. Among the available options, the solar absorption chiller technology is attractive for regions that have high solar irradiance and are heavily inclined towards cooling loads.

This paper presents a design of a solar absorption chiller system using an indirect seawater cooling system as the heat rejecter / condenser loop, and an underground cold water tank as a thermal storage. A method for estimating the annual energy performance and costs of the system by using ANSYS FLUENT and Excel is also presented. A one-year simulation of the proposed solar absorption chiller system design located in Veracruz, Mexico is performed using the above mentioned method and compared with a water-powered chiller.

Method

Performing annual simulations can be challenging when dealing with highly dynamic processes especially if they are involved in a system-level simulation. In these scenarios, doing a CFD simulation requires high computer loads, not to mention being time consuming, which is inconvenient. An analytical approximation has the advantage to simplify the calculations significantly, as well as being flexible and simple to modify. This is why an analytical approach by using an Excel worksheet was decided in order to do the annual simulation.

In this paper, computer fluid dynamics was used in order to simulate the several components composing a solar absorption chiller in order to obtain some key design parameters and then these were used in analytical functions in order to perform the annual simulation.

The proposed design of the absorption chiller can be seen in figure 1.

It is composed by an array of evacuated tube solar collectors (ETC), a hot water buffer tank, an indirect seawater cooling system and an underground cold thermal storage tank.

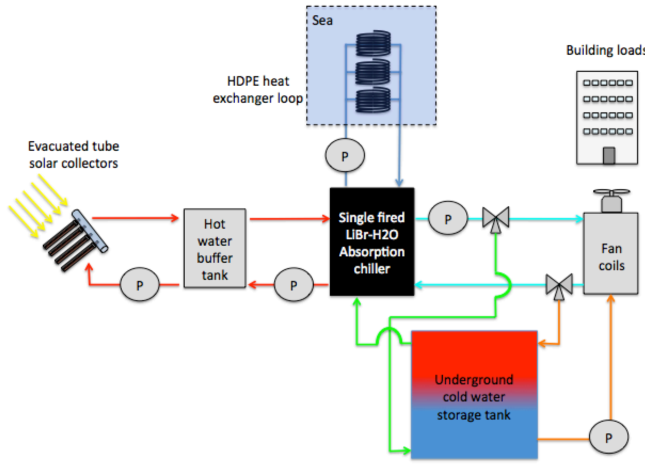

Figure 1. Diagram of the design of the proposed solar absorption chiller system

Weather data of Veracruz, Mexico was obtained through the NSRDB data viewer et al. which is an interactive GIS application from which weather data such as dry bulb temperature, humidity, solar irradiation, wind velocity, etc. can be downloaded. This data can be configured into either 1-hour time steps or 30 minute time steps for one year. The building load conditions were set according to ASHRAE and ACCA et al standards. An analysis of the data was done in order to find out how many cooling load and heating load hours were required. For this, it is necessary to set some conditions.

It is well known by the HVAC industry that when the ambient temperature is in the range of 19.4 and 15.5 °C no heating or cooling load is required due to these ambient temperature yielding a comfort temperature inside the building. Having this in mind, the analysis showed that out of the 8756 hours of one whole year, 8588 required a cooling load, and 0 hours required a heating load. The remaining 172 hours were in the comfort temperature range, therefore this region is complete one sided to cooling loads.

It is important to notice that the rated chiller's cooling capacity can increase or decrease depending on the cooling capacity factor (CCF), which is related directly to the temperature of the heat medium and the temperature of the cooling water. With this in mind, we performed the following calculations.

Evacuated tube solar collector

In the solar collector market, it is of great importance to standardize the way the solar collectors are measured in order to obtain an unbiased result of the collector efficiency of different providers. In order to do this, a separate organization usually tests and evaluates the performance of the collectors using the same standards and conditions. In Europe, the Quality Assurance in Solar Heating and Cooling Technology (QAiST) project was ran in order to develop the European Standards for the solar thermal market for quality assurance of the products and to enhance competitiveness of European manufacturers. In this project, the EN12795 was developed, which is the standard for solar thermal collectors. In this standard, several test conditions and requirements are thoroughly mentioned in order to obtain a comparable result among the several

manufacturers. The main method used in order to evaluate an evacuated tube solar collector is the quasi-dynamic method; therefore, our calculations were based on this methodology.

In order to calculate the performance of the ETC, we first need to identify some key parameters, in particular, the zero loss efficiency, which is the rate of heat transfer from the evacuated tube to the working fluid with no convection losses.

A simulation was done in ANSYS Fluent in order to obtain these parameters. The heat transfer efficiency of our design was of 96% and the solar heat flux to the absorber coat was of 800 W/m2 with a direct irradiation of 1000 W/m2. With this information we obtained a zero loss efficiency of 0.768, which is a fairly common value among ETC in the market.

Indirect seawater cooling system

In order to perform a one year simulation of the performance of the indirect seawater heat exchanger in a mathematical approach, it is necessary to understand the overall heat transfer of the heat exchanger, which in this case is composed by an array of helical coils. The helical coil shape was chosen because it has been proven to provide higher heat transfer rates than a linear pipe configuration due to the bifurcation of the fluid when colliding with the outer wall due to centrifugal forces.

It was decided to use the Logarithmic mean temperature difference method in order to obtain the overall heat transfer coefficient of this specific helical coil heat exchanger.

Fluent was used to do the simulation to obtain the heat flux through the walls of the helical coil under several seawater current velocities. This was done because the overall heat transfer coefficient depends on the properties of the heat exchanger such as flow rate, geometry and other physical properties of the fluid. Sea current velocity and temperature data was obtained in order to know what is the range of the max and minimum current velocities for our simulations. We can observe the effect of the centrifugal forces at the helical coil outlet in figure 3.

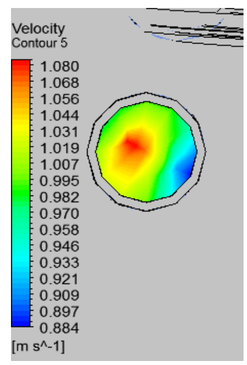

Figure 2. Velocity contour at the outlet of the helical coil

Six simulations were done in order to obtain a curve characterizing the overall heat transfer coefficient in function of the velocity.

Underground thermal storage tank

Because we are using a single UTS tank, it is important to take account for the thermocline losses that occur. In order to do this, we used the method proposed by Bayon et al. In their work, the authors propose an analytical function describing the behavior of a thermocline storage tank with the objective of facilitating annual simulations of this type of thermal storage system. One important detail of this analytical function is that it does not take account for thermal losses to the exterior because molten salt thermal storage tanks are very well insulated and they present minimal heat, therefore, in order to use this function for our research, it is important to first demonstrate that the underground thermal storage tank has negligible heat loss.

Simulation in Fluent was performed to understand the heat flux entering the UTS as a function of time. The heat flux was minimal even at its peak, which with we could assume the tank had insignificant thermal losses. With this the thermocline function was implemented in the excel worksheet.

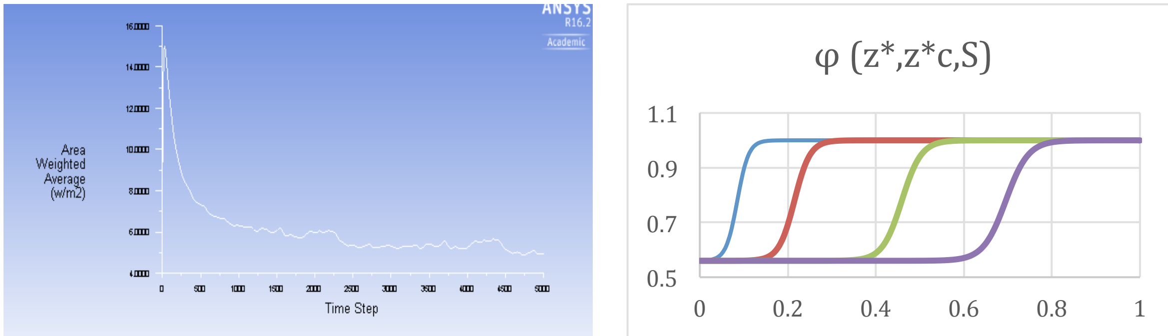

Figure 3. a) Change in area weighted heat flux as a function of time in the underground thermal storage tank, b) dimensionless thermocline temperature in function of the dimensionless height during charge.

Results and discussion

With the above information, an annual simulation of the entire system was done with the created Excel worksheet tool. When using a 570 m2 area of collector, 1500 m2 area of the indirect seawater heat exchanger, 200 m3 of the underground thermal storage and a 60 m3 of volume of the hot buffer tank, the entire system performed as observed in figure 5.

With this configuration, the solar absorption chiller is capable of covering 67% of the annual building load; therefore, during 33% of the time, a backup chiller has to cover it.

We also performed the simulation using a double cold-water thermal storage tank in order to compare it with the thermocline design. This design managed to cover 76% of the annual building load. It is important to mention that with this design, there is an extra 118,684 kWh of heat available for hot water generation. This amount depends greatly on the size of the UTS.

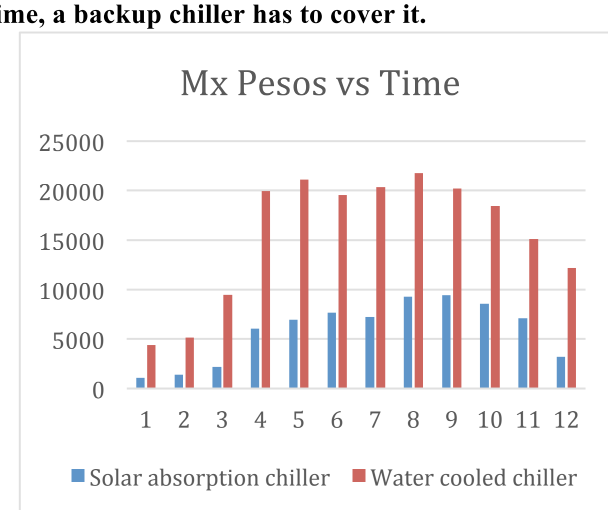

Figure 4. Annual cost comparison between the solar absorption chiller and a traditional water-cooled chiller.

Afterwards, we performed a simulation with a water-powered chiller using a cooling tower and compared it with the same chiller using the indirect seawater cooling system. The chiller with the cooling tower consumes an annual electric power of 132,858.15 kWh and the chiller with the indirect seawater cooling system consumes 148,789.34 kWh. The cooling tower chiller consumes slightly less because the wet bulb temperature is slightly lower than the sea surface temperature throughout the year, though it is important to notice that this comparison does not include the pump's electric demand or the water consumption in the case of the cooling tower, which can be significant.

For future research, it is recommended to include the installation costs, which can be significant in the case of the absorption chiller, in order to do a more accurate comparison, as well as including the additional parasitic loads in the case of a system using a cooling tower. Finally, it is recommended to perform an experimental validation of the annual simulation results.