

Low-Power Air Conditioning Technology with Cold Thermal Energy Storage

Leila Dehghan*, Ahmad Fakhar

Department of Mechanical Engineering, Faculty of Engineering, Azad University of Kashan, Iran

*Corresponding author: lleiladehghan1385@yahoo.com

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Abstract Air conditioning of buildings is responsible for a large percentage of the greenhouse and ozone depletion effect, as refrigerant harmful gases are released into the atmosphere from conventional cooling systems. The vapor compression refrigeration is one of the many refrigeration cycles and is the most widely used method for air-conditioning of buildings. On the other hand, solar thermal energy can be used to efficiently cool in the summer. Single, double or triple iterative absorption cooling cycles are used in different solar thermal cooling system designs. Absorption chillers operate with less noise and vibration than compressor-based chillers, but their capital costs are relatively high. In this study, a system is proposed as a combination of the aforementioned systems and the power consumption is minimized using cold thermal energy storage (CTES).

Keywords: *building cooling system, cold thermal energy storage*

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1. Introduction

Cooling of buildings can be achieved at very different energy consumption, ranging from zero energy for purely passive over low-energy consumption for earth heat exchange up to high electrical energy requirements for active compressor chillers. The application of different systems depends strongly on the cooling load, which has to be removed. If a building cannot be cooled by passive means such as night ventilation or earth heat exchange alone, active cooling technologies have to be employed. Today, the dominant cooling systems are electrically driven compression chillers, which have a world market share of about 90%. The average coefficient of performance (COP) of installed systems is about 3.0 or lower and only the best available equipment can reach a COP above 5.0. To reduce the primary energy consumption of chillers, thermal cooling systems offer interesting alternatives, especially if primary energy neutral heat from solar thermal collectors or waste heat from cogeneration units can be used. The main technologies for thermal cooling are closed-cycle absorption and adsorption machines, which use either liquids or solids for the sorption process of the refrigerant. Absorption chillers today are available in the range of 5 to 20,000 kW. In the last few years some new developments have been made in the medium-scale cooling range of 10 to 50 kW for water/lithium bromide and ammonia/water absorption chillers [1,2].

Kalogirou et al [3] presented modelling and simulation of an absorption solar cooling system. The system is modelled with the TRNSYS simulation program and the

typical meteorological year file containing the weather parameters of Nicosia, Cyprus. Florides et al [4] presented the modelling, simulation and total equivalent warming impact of a domestic-size absorption solar cooling system. The system consists of a solar collector, storage tank, a boiler and a LiBr-water absorption refrigerator. Experimentally determined heat and mass transfer coefficients were employed in the design and costing of an 11 kW cooling capacity solar driven absorption cooling machine which, from simulations, was found to have sufficient capacity to satisfy the cooling needs of a well-insulated domestic dwelling. The system was modelled with the TRNSYS simulation program using appropriate equations predicting the performance of the unit. Assilzadeh et al presented a solar cooling system that has been designed for Malaysia and similar tropical regions using evacuated tube solar collectors and LiBr absorption unit. The modeling and simulation of the absorption solar cooling system is carried out with TRNSYS program. The typical meteorological year file containing the weather parameters for Malaysia is used to simulate the system. A comparative study was carried out by Fong et al [5] for the five types of solar cooling systems for a typical office in the subtropical Hong Kong. The results were worked out with the emphasis of suitable system control and operation in response to the year-round changing climatic and loading conditions. Based on the best year-round total of primary energy consumption, the order of the five types of solar cooling systems is: solar electric compression refrigeration, solar absorption refrigeration, solar adsorption refrigeration, solar solid desiccant cooling, and solar mechanical compression refrigeration. Tsoutsos et al [6] the performance and economic evaluation of a solar heating and cooling system of hospital in Crete, is studied

using the transient simulation program (TRNSYS). The meteorological yearfile exploited the hourly weather data where produced by 30-year statistical process. The required data were obtained by Hellenic National Meteorological Service. Vidal et al [7] carried out an

study on the hourly simulation of an ejector cooling cycle assisted by solar energy. The system is simulated using the TRNSYS program and the typical meteorological year (TMY) file that contains the weather data from Florianópolis, Brazil.

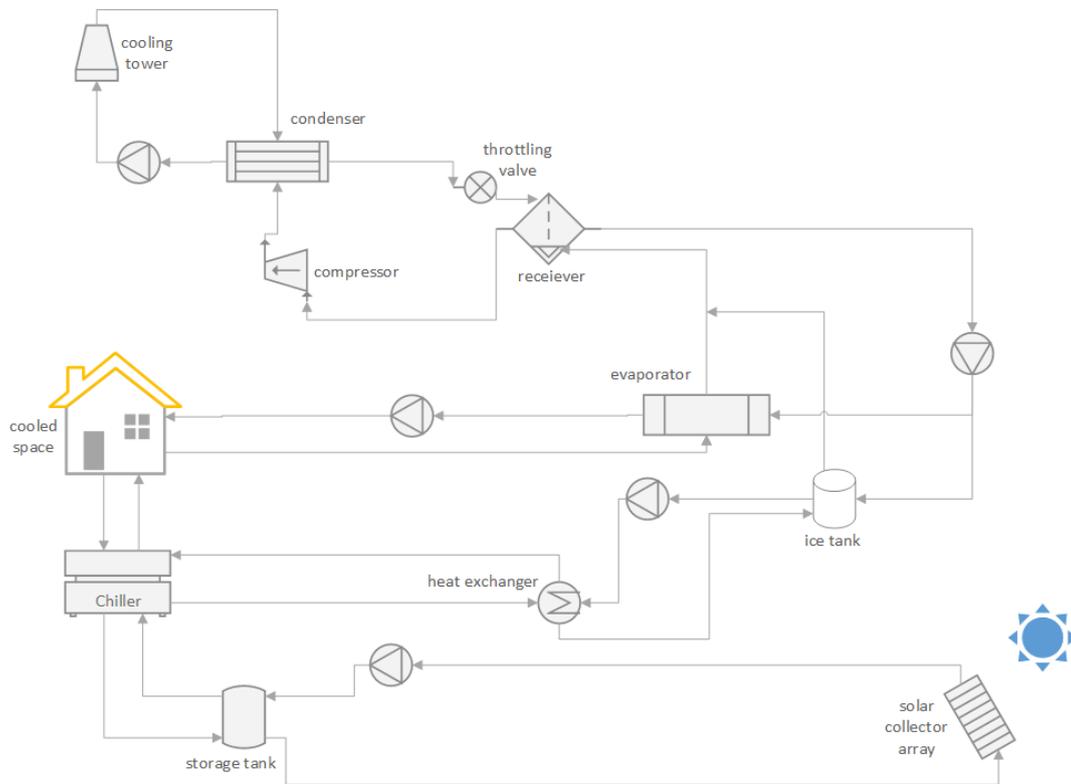


Figure 1. Schematics of the proposed cooling system by eliminating the adsorption chiller cooling tower

The ejector cycle uses R141b as the working fluid and a one-dimensional ejector is modelled in EES (Engineering Equation Solver). A full simulation model was developed by Eicker [8] for absorption cooling systems, combined with a stratified storage tank, steady-state or dynamic collector model and hourly resolved building loads. The model was validated with experimental data from various solar cooling plants. Sparber et al [9] reported that till 2007 there were 81 installed large scale solar cooling systems, eventually including systems which are currently not in operation. 73 installations are located in Europe, 7 in Asia, China in particular, and 1 in America (Mexico). 60% of these installations are dedicated to office buildings, 10% to factories, 15% to laboratories and education centers, 6% to hotels and the left percentage to buildings with different final use (hospitals, canteen, sport center, etc). They also cited that 56 installations are belong to absorption systems and the overall cooling capacity of the thermally driven chillers amounts to 9 MW 31% of it is installed in Spain, 18% in Germany and 12 % in Greece. Bong et al [10] designed and installed solar absorption chiller in Singapore. The system included 7 KW absorption chiller, heat pipe collectors with a total area of 32 m², a hot water storage tank, an auxiliary heater and a 17.5 KW cooling tower. They cited that the overall average cooling capacity provided was 4 KW, solar fraction of 39% and COP of 0.58. Balghouthi et al [11] accomplished a simulation using TRNSYS program in order to select and size different components of solar absorption chiller. They reported that solar absorption cooling systems were suitable for Tunisian's condition.

2. Description of the Proposed Systems

Figure 1 depicted two proposed cooling systems. In this scheme, the cooling tower for the adsorption chiller is replaced with a heat exchanger in which the water is cooled by recirculating the hot water in the ice storage tank.

In the design the ice is produced by pumping very cold liquid refrigerant HCFC-22 in commercial application through an array of pipes immersed in a tank of water. The tank is used as the heat sink for the adsorption chiller. The selected heat source for the chiller is the thermal energy from the sun. Technically, the design resembles process refrigeration. System components, that is, the compressor and condenser, pressure receivers, refrigerant pumps, evaporators, and ice tanks, are individually selected for the application and integrated to provide a reliable refrigeration system. Unlike direct-expansion systems, which rely on additional heat-transfer surface area to separate refrigerant vapor from liquid refrigerant, ice-on-pipe systems use a low-pressure receiver and a method called liquid overfeed to accomplish this. Chilled water or ice is produced by pumping cold liquid refrigerant to a chiller evaporator or an ice tank at a rate faster than that required to evaporate it. A two-phase solution of refrigerant liquid and vapor results, and is returned to the low-pressure receiver. This higher refrigerant flow rate is the reason for the system being referred to as liquid overfeed [12].

The refrigerant returned to the low-pressure receiver is nearly saturated. Refrigerant liquid that does not boil off

in the evaporator is returned for a second pass. Note that the open or atmospheric design of the system dictates the use of a heat exchanger to separate ice water from the building cooling water loop. The cooling loop is normally a closed system [12].

2.1. Vapor Compression Cooling Cycle

The cycle used in the design called the ideal vapor-compression refrigeration cycle, and it is shown schematically in the above figure. The vapor-compression refrigeration cycle is the most widely used cycle for refrigerators, air-conditioning systems, and heat pumps. It consists of four processes: Isentropic compression in a compressor, Constant-pressure heat rejection in a condenser, Throttling in an expansion device, Constant-pressure heat absorption in an evaporator. In an ideal vapor-compression refrigeration cycle, the refrigerant enters the compressor at state 1 as saturated vapor and is compressed isentropically to the condenser pressure. The temperature of the refrigerant increases during this isentropic compression process to well above the temperature of the surrounding medium. The refrigerant then enters the condenser as superheated vapor at state 2 and leaves as saturated liquid at state 3 as a result of heat rejection to the surroundings. The temperature of the refrigerant at this state is still above the temperature of the surroundings.

The saturated liquid refrigerant at state 3 is throttled to the evaporator pressure by passing it through an expansion valve or capillary tube. The temperature of the refrigerant drops below the temperature of the refrigerated space during this process. The refrigerant enters the evaporator at state 4 as a low-quality saturated mixture, and it completely evaporates by absorbing heat from the refrigerated space. The refrigerant leaves the evaporator as saturated vapor and reenters the compressor, completing the cycle. In a household refrigerator, the tubes in the freezer compartment where heat is absorbed by the refrigerant serves as the evaporator. The coils behind the refrigerator, where heat is dissipated to the kitchen air, serve as the condenser.

2.1.1. High Pressure Side

On the high-pressure side of the system, the cold refrigerant vapor that collects at the top of the low-pressure receiver is drawn off by the compressor. After compression, the pressurized (and now hot) vapor passes to the condenser, where cooling tower water circulating through the shell causes the refrigerant to condense. The liquid refrigerant, still at high pressure, exits the condenser and enters a high-pressure receiver, where it is stored for later use. Refrigerant flow from the high-pressure receiver is regulated by a refrigerant metering device to ensure that a minimum liquid level is maintained in the low-pressure receiver.

This coil is actually a series of steel pipes immersed in a tank of water. Cold refrigerant (HCFC-22) is then pumped through these pipes to freeze the water that surrounds them. Bubbles flow around the steel pipes to agitate the water in the tank, sometimes by injecting air at the bottom. The rising air bubbles promote dense, even ice formation during the freezing cycle and uniform melting when the tank is discharged.

2.1.2. Low Pressure Side

The low-pressure receiver plays a critical role in the systems. It separates the two-phase refrigerant solution returning from the ice coil (or chiller evaporator) into liquid and vapor. Gravity induces this separation, causing the liquid refrigerant and oil to settle at the bottom of the receiver while pure refrigerant vapor collects at the top. As the compressor draws this vapor from the receiver, the liquid level falls. To ensure that there is always sufficient liquid in this vessel, a liquid level control adds refrigerant from the high-pressure receiver as needed.

Liquid overfeed systems require a separate oil return/recovery system. This is because the preferred compressor type (helical rotary/screw) expels significant amounts of oil into the discharge line. Entrained in the refrigerant, the oil makes its way through the condenser and high-pressure receiver, eventually ending up in the low-pressure receiver. There, the oil collects at the bottom of the tank (along with the liquid refrigerant) and cannot return to the compressor through the suction line. A separate oil-recovery system is needed to capture, distill, and return the oil to the compressor.

2.2. Solar Cooling System

When there is a source of inexpensive thermal energy at a temperature of 100°C to 200°C is absorption refrigeration. In this design, we used the most widely used absorption refrigeration system is the ammonia–water system, where ammonia (NH_3) serves as the refrigerant and water (H_2O) as the transport medium. Other absorption refrigeration systems include water–lithium bromide and water–lithium chloride systems, where water serves as the refrigerant. The latter two systems are limited to applications such as air-conditioning where the minimum temperature is above the freezing point of water.

It should be noted that this system looks very much like the vapor-compression system, except that the compressor has been replaced by a complex absorption mechanism consisting of an absorber, a pump, a generator, a regenerator, a valve, and a rectifier. Once the pressure of NH_3 is raised by the components in the box (this is the only thing they are set up to do), it is cooled and condensed in the condenser by rejecting heat to the surroundings, is throttled to the evaporator pressure, and absorbs heat from the refrigerated space as it flows through the evaporator. So, there is nothing new there. Here is what happens in the box: Ammonia vapor leaves the evaporator and enters the absorber, where it dissolves and reacts with water to form $\text{NH}_3\cdot\text{H}_2\text{O}$. This is an exothermic reaction; thus heat is released during this process. The amount of NH_3 that can be dissolved in H_2O is inversely proportional to the temperature. Therefore, it is necessary to cool the absorber to maintain its temperature as low as possible, hence to maximize the amount of NH_3 dissolved in water. The liquid $\text{NH}_3\cdot\text{H}_2\text{O}$ solution, which is rich in NH_3 , is then pumped to the generator. Heat is transferred to the solution from a source to vaporize some of the solution. The vapor, which is rich in NH_3 , passes through a rectifier, which separates the water and returns it to the generator. The high-pressure pure NH_3 vapor then continues its journey through the rest of the cycle. The hot $\text{NH}_3\cdot\text{H}_2\text{O}$ solution, which is weak in NH_3 , then passes through a regenerator, where it

transfers some heat to the rich solution leaving the pump, and is throttled to the absorber pressure [13].

3. System Analysis

3.1. Vapor Compression Cooling

As described previously, a refrigeration system is considered which is operating between the pressure limits of 1.6 Mpa and 200 kpa with refrigerant-134a as the working fluid. The refrigerant leaves the condenser as a saturated liquid and is the flash chamber operates at 0.45 Mpa. Part of the refrigerant evaporates during this flashing

process, and this vapor is mixed with the refrigerant leaving the low-pressure compressor. The mixture is then compressed to the condenser pressure by the high-pressure compressor. The liquid in the flash chamber is throttled to the evaporator pressure and cools the refrigerated space as it vaporizes in the evaporator. The mass flow rate of the refrigerant through the low-pressure compressor is 0.11 kg. It is assumed that the refrigerant leaves the evaporator as a saturated vapor and the isentropic efficiency is 86% for the compressor. The results for this system and the one stage vapor compression system is shown in Table 1. It is clear that the choice of the overfeed system can increase the supply for the cooling load.

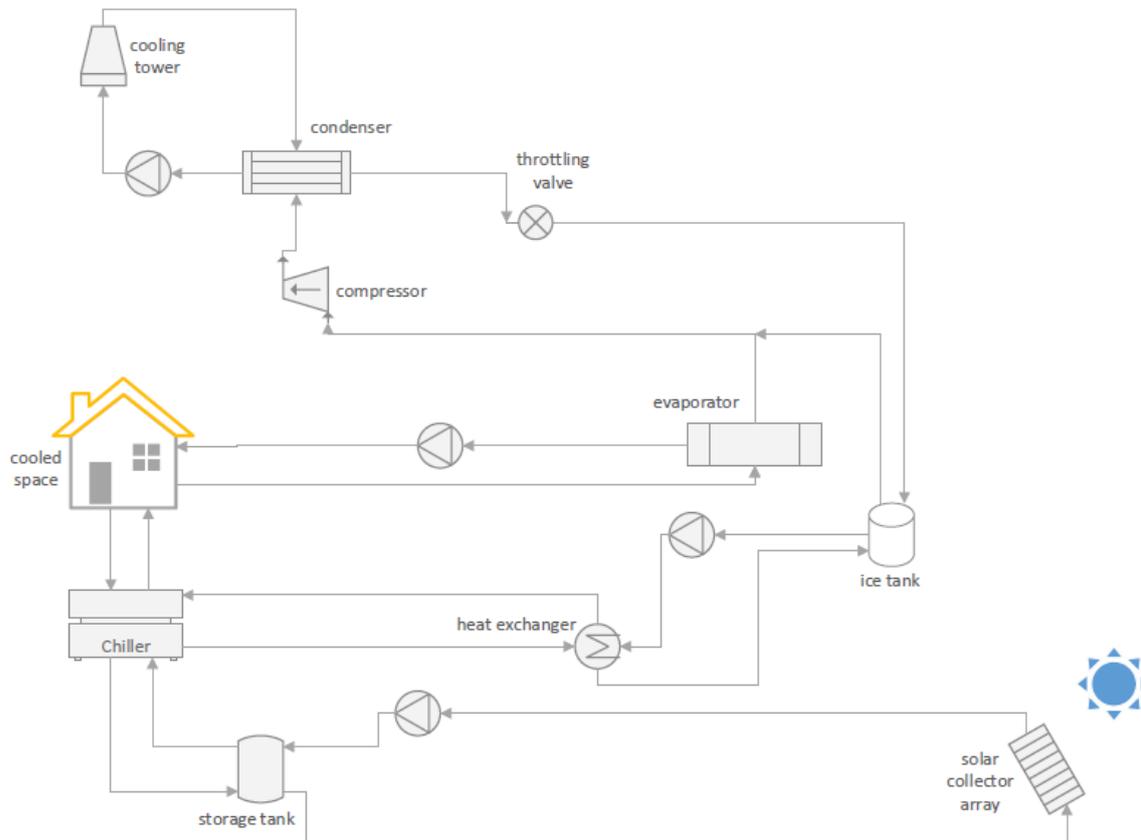


Figure 2. Single stage cooling system integrated absorption chiller

Table 1. Comparison of the One stage and overfeed systems

	Input power	Heat removal	COP
One stage	7.45 kW	19.33 kW	2.59
Overfeed	8.60 kW	18.54 kW	2.16
% changes	+15.4	-4.1	-16.6

3.2. Storage Analysis for Minimum Power

A MATLAB code is developed to simulate the overall system with the purpose that the power consumption of the system be minimized. It is achievable while the cold storage tank utilization is maximized. The thermal load can be found in Table 2. The simulation result is illustrated in Figure 3 where the energy delivery of the heat exchanger, evaporator and ice tank are compared.

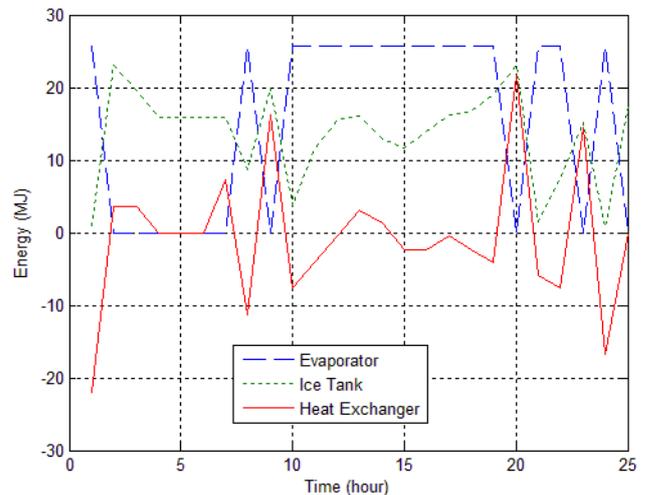


Figure 3. Simulation result for minimum system power consumption

Table 2. Thermal Load

Hour	Load
1	1
2	1
3	1
4	0
5	0
6	2
7	4
8	4.5
9	5
10	6
11	7
12	8
13	7.5
14	6.5
15	6.5
16	7
17	6.5
18	6
19	6
20	5.5
21	5
22	4
23	2.5
24	0

4. Conclusion

A MATLAB code is developed to simulate the daily performance of a vapor compression conditioning system integrated a cold thermal energy storage (ice storage). To minimize the overall power consumption of the conditioning system, cold storage side of the system should be used as much as possible because it just uses a small pump. Under certain load condition, the performance is simulated for the transferred energy through evaporator, ice tank and heat exchanger.

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