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Performance study of hybrid cooling systems for the utilization in buildings

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Résumé

Cette thèse est une contribution à la réduction de la consommation d'énergie primaire et à une meilleure utilisation des sources d'énergie renouvelables dans le cadre des systèmes de refroidissement utilisés dans le bâtiment.
Après un état de l'art sur les systèmes de refroidissement, un modèle dynamique d'un système de rafraîchissement solaire à base de machine à absorption est développé et simulé. Ensuite, un facteur d'efficacité ($\mathcal{EF}$) pour comparer la pertinence de ce système dans différentes régions du monde est défini. Dans la troisième partie, la notion des systèmes de refroidissement hybride -une méthode efficace contribuant à la réduction de la consommation d'énergie primaire- est présentée. Puis, les systèmes hybrides de refroidissement sont classés en catégories et sont comparés avec les systèmes de refroidissement individuels. Ensuite, un schéma permettant de sélectionner le meilleur système de refroidissement hybride dans des conditions données est proposé. Dans la dernière partie, une méthode de dimensionnement d'un système hybride est établie. Cette approche est fondée sur des critères économiques et vise une optimisation des énergies renouvelables disponibles localement. Ainsi, le dimensionnement est réalisé en tenant compte de la région spécifique d'utilisation. Pour ce faire, un système de refroidissement hybride, conçu pour une maison standard, est modélisé puis simulé en utilisant le logiciel Trnsys. Finalement, et pour illustrer la méthode proposée, la problématique de dimensionnement est considérée pour deux régions différentes du globe ; à savoir Marseille-France et Beyrouth-Liban. Le but est d'évaluer les performances de la méthode, à travers des données effectives, pour diverses conditions climatiques, prix des composants et tarif d'électricité.

Mots clés : systèmes de refroidissement, énergie renouvelable, gestion, efficacité énergétique, système hybride, simulation.

Abstract

This thesis is a contribution towards the reduction of primary energy consumption and a better use of the renewable energy sources within the cooling system for building use.
After a state of the art of the cooling machines for building use, a dynamic model for a solar absorption cooling system is developed and simulated. Then, an effectiveness factor ($\mathcal{EF}$) for the comparison of solar absorption chiller suitability in different locations is defined. In the third chapter, the concept of hybrid cooling system -an efficient method contributing to the reduction of primary energy consumption- is presented. Hybrid cooling systems are categorized and reviewed, with the improvement achieved compared to standalone technologies. Then, a scheme for the selection of the best hybrid cooling system for given conditions is proposed. In the last part, an optimal sizing method that defines, in a specific region, a hybrid cooling energy system, economically feasible with maximum renewable energy share is presented. Thereby, the sizing method is performed taking into account the region where it will be used. For this purpose, a hybrid cooling system, used for a standard residential house, is designed. The system is modeled and simulated using a transient system simulation program, called Trnsys. Finally, the problem of sizing is studied for different case studies ; namely Marseilles-France and Beirut-Lebanon. The aim is to assess the proposed method according to diverse climatic conditions, component prices and electricity costs.

Keywords : cooling systems, renewable energy, energy management, effectiveness, hybrid system, simulation.

Table des matières

Table des figures

Liste des tableaux

General introduction

Nowadays, greenhouse gas emissions and their contribution to global warming have been brought to the forefront of international debate. There is a long-run relationship between greenhouse gas emissions and the energy consumption [1]. According to the International Agency of Energy (IEA), the energy consumption increased at a rate greater than 15% during the 30 years from 1973 to 2004 [2], which caused an increase of CO_2 emissions of 5%. In order to cut down this critical situation and to fight global warming, a number of nations have signed the Kyoto Protocol of the United Nations. It is a legally binding agreement that puts the obligation to reduce current emissions on developed countries, which are considered historically responsible for the current levels of greenhouse gases in the atmosphere.

The total contribution from buildings towards energy consumption, both residential and commercial, has increased and exceeded the other major sectors, namely industrial and transportation. It consists about 40% of the total energy consumption in developed countries [2]. Emission reductions from buildings can be achieved either by controlling emissions from the energy supply or by decreasing energy consumption that could be realized through improved building design, increased energy efficiency and conservation, and using other mechanisms which reduce energy demand in buildings [3]. Also, it was found that the use of energy efficiency measures drives out the greatest portion of emission reductions and stabilizes the global climate [2].

Among building services, the energy use of heating ventilation and air- conditioning (HVAC) systems is significant. It consists about half of the total energy consumption in buildings, and around $10 - 20\%$ of the total energy consumption in developed countries [2]. For instance, its share of total residential energy consumption is about 45% in Morocco or in Tunisia [4], 55% in China and 70% in Thailand [5]. Besides, HVAC systems are matter of debate due to the high energy consumption and the use of refrigerant with high global warming potential. For instance, HFC and CFC were prohibited by Kyoto protocol due to their high global warming effect. Recently, numerous efforts were devoted in order to develop a clean technology with low energy consumption to achieve the comfort level -in terms of temperature and humidity- in a building. Accordingly, the development of cooling and heating efficient energy systems using free energy sources is accelerating rapidly, such as cooling systems that use solar energy, biomass, geothermal energy, etc. In addition, in order to enhance its efficiency, HVAC systems could be designed based on the hybridization of production and/or sources of energy. Thus, numerous hybrid systems were proposed and studied in the literature. It is found that a hybrid cooling system properly selected could achieve a great reduction in energy consumption and improvement of the coefficient of performance.

This thesis is a contribution to the study of the performance and implementation of renewable energy-based cooling systems for building use. The main aim is the reduction of energy consumption and the efficient use of renewable energy resources available. More precisely, the works of the present thesis focus on three main objectives.

First of all, a detailed analysis of a solar cooling system is proposed. It consists of a solar absorption cooling system, and the study is performed taking into consideration the energy potentials available in different regions of the world.

The second objective is to perform a review of the hybrid cooling systems existing. Currently, a great number of hybrid systems and different technologies is proposed in the literature. Consequently, based on a detailed study, a description of the advantages and disadvantages of the different solutions studied in the literature is presented. Furthermore, a strategy for the selection of a suitable cooling system is proposed.

The third objective of the thesis is the optimization of hybrid cooling systems based on renewable energy. The main purpose is to determine, for a given region of the world, a cooling system with minimum energy consumption and maximum share of renewable resources. More precisely, once a hybrid technology is adopted, the main purpose is to define an optimal size taking into consideration a certain number of inherent constraints that could be the energy potential, the financial costs of the system designed and the area available in a building to install the said system.

The present thesis consists mainly of four chapters.

The first chapter is dedicated for the generalities about cooling systems. The aim is to present the most common processes used in cooling systems for buildings. For this purpose, the cooling processes are categorized into main classes according to the principle of operation. Then, a detailed description of each cooling machine is presented. The detailed description consists of the operating principle, the state of the art, and a list of the advantages and disadvantages of each cooling system.

The second chapter is devoted for the analysis of the operation of solar absorption machine. The aim is to propose a method for the analysis and the comparison of its use in different regions of the world. More precisely, the studied system consists of a single-effect $LiBr/H_2O$ absorption chiller directly connected to the solar collector without heat storage. In order to describe the behavior of solar absorption chiller, a dynamic model based on heat balance equations for solar collector and generator is described and solved using the finite volume method. Then, the model is validated using experimental results. A number of simulations of the dynamic model are performed for different sites (namely representative cities) of the world and taking into consideration the energy potentials during typical days of the year. It should be noted that the considered sites are selected on different latitude lines in order to determine a characteristic of the world. Based on the series of simulation performed, an effectiveness factor noted $\mathcal{EF}$ is defined. This annual factor is calculated taking into consideration the cooling load, the initial and the operational costs for each site.

The third chapter consists of a detailed review of the existing hybrid cooling systems. More precisely, the systems are categorized into five main categories according to the base cooling technology used, namely vapor compression, absorption, adsorption, desiccant-evaporative and multi-evaporator cooling. For each category, a description of the different configurations is presented, and the advantages of each hybridization are reviewed. Then, a selection scheme according to operation parameters is proposed. The scheme allows to determine a cooling system clean with low pollution emissions and ensuring a reduction in primary energy consumption in a building.

The last chapter focuses on the sizing problem of cooling systems. More precisely, we consider a typical house and we suppose that the energy production, to satisfy the cooling need, is ensured by a hybrid cooling system based on renewable energy and the electric grid. The topology of the hybrid energy production system is determined based on the selection scheme proposed in the third chapter. The objective is twofold. On one hand, the aim is to design an optimal hybrid system that reduces the energy consumption from the grid. On the other hand, the optimal configuration should be performed taking into consideration the area available in the building, the energy potential and the economic costs for the considered site. Therefore, the problematic of optimization is formalized as a mathematical problem under constraints. Then, the results are obtained and discussed for the two case studies selected, namely Marseilles-France and Beirut-Lebanon.

The last part consists of a general conclusion and the perspectives of the works performed in the present thesis.

Introduction générale

De nos jours, les émissions de gaz à effet de serre et leur contribution au réchauffement climatique de la planète sont placées au premier plan de l'actualité internationale. Il existe une relation à long terme entre les émissions de gaz à effet de serre et la consommation d'énergie [1]. Selon l'Agence Internationale de l'Energie (AIE), la consommation d'énergie a augmenté avec un taux supérieur à 15% au cours des 30 années entre 1973 et 2004 [2], ce qui a provoqué une augmentation des émissions de CO_2 de 5%. Afin de réduire ces émissions critiques et de lutter contre le réchauffement de la planète, un certain nombre de pays ont signé le Protocole de Kyoto des Nations Unies. Il s'agit d'un accord juridiquement contraignant qui impose la réduction des émissions dans les pays développés, considérés comme historiquement responsables des gaz à effet de serre dans l'atmosphère.

L'énergie consommée dans les bâtiments résidentiel et commercial est en nette augmentation, elle a même dépassé celle consommée dans les autres grands secteurs que sont l'industrie et le transport. Elle représente environ 40% de la consommation totale d'énergie dans les pays développés [2]. La réduction des émissions provoquées dans les bâtiments peut être obtenue, soit en contrôlant les émissions provenant de la production énergétique, ou en diminuant la consommation d'énergie grâce à une meilleure conception des bâtiments, en améliorant l'efficacité énergétique et en utilisant de nouveaux mécanismes qui réduisent la demande d'énergie dans les bâtiments [3]. En outre, il est prouvé que l'utilisation des mesures d'énergies efficaces conduit à une réduction importante des émissions et à un climat stable [2].

Parmi les services dans les bâtiments, l'énergie utilisée par les systèmes de climatisation est significative. En effet, elle constitue environ la moitié de l'énergie totale consommée dans les bâtiments, et environ de 10 à 20% de l'énergie totale dans les pays développés. A titre d'exemple, le pourcentage de l'énergie consommée par les systèmes de climatisation représente environ 45% au Maroc ou en Tunisie [4], 55% en Chine et 70% en Thaïlande [5]. Ainsi, les systèmes de climatisation font l'objet de débats à cause de leur grande consommation d'énergie et, en particulier, de part leur utilisation de fluides frigorigènes avec un potentiel de réchauffement global. A titre d'exemple, les fluides HFC et CFC polluants ont été interdits suite au protocole de Kyoto. Récemment, de nombreux efforts ont été consentis afin de développer de nouvelles technologies propres et à faible consommation d'énergie pour assurer le confort au niveau de la température et de l'humidité dans le bâtiment. Plus précisément, une activité importante concerne la conception de systèmes de climatisation qui soient efficaces et fondés sur l'utilisation des sources d'énergies renouvelables ; c'est-à-dire l'énergie solaire, la biomasse, la géothermie, etc. En outre, et afin d'augmenter leur efficacité, ces systèmes de climatisation peuvent être conçus sur la base d'une hybridation des dispositifs de production et/ou des sources d'énergie. Ainsi, de nombreux systèmes combinant des différents procédés de refroidissement ont été proposés et étudiés dans la littérature. En effet, un système de refroidissement hybride bien sélectionné offre une grande réduction de l'énergie consommée et une amélioration du coefficient de performance.

Cette thèse est une contribution à l'étude des performances et à la réalisation des systèmes de refroidissement à base d'énergie renouvelable pour une exploitation dans le bâtiment. Le but est la réduction de la consommation d'énergie et l'utilisation efficace des ressources énergétiques renouvelables disponibles. Plus précisément, les travaux de thèse s'articulent autour de trois objectifs.

Tout d'abord, une analyse fine du fonctionnement d'un système de production du froid, à base d'énergie solaire, est proposée. Il s'agit d'un système conçu à partir d'une machine à absorption et l'analyse est effectuée en tenant compte des potentiels énergétiques disponibles dans des différents endroits du globe.

Le deuxième objectif est la réalisation d'une synthèse sur les systèmes de refroidissement hybride existants. Actuellement, dans la littérature, un grand nombre d'hybridations et de technologies différentes sont proposées. Ainsi, le but de cette synthèse, fondée sur une analyse approfondie de la littérature, est la description des avantages et des inconvénients des différentes solutions présentées dans la littérature. En outre, une stratégie en vue de la sélection d'une solution adaptée, est proposée.

Le troisième objectif de la thèse est l'optimisation des systèmes de refroidissement hybrides à base d'énergie renouvelable. Le but principal est de déterminer, pour une région du globe donnée, un système de refroidissement à consommation d'énergie minimale et un taux maximal de participation de ressources renouvelables. Plus précisément, et après un choix adapté d'une topologie d'hybridation, le but principal consiste à définir un dimensionnement optimal tenant compte de certaines contraintes inhérentes qui peuvent être le potentiel énergétique, le coût financier de la conception du système et l'espace disponible dans le bâtiment en vue de son installation.

La thèse est composée de quatre chapitres.

Le premier chapitre est dédié à des généralités sur les systèmes de refroidissement. Le but est d'introduire les différents procédés de production du froid les plus couramment utilisés dans le bâtiment. Pour ce faire, ces procédés sont classés, selon leurs principes de fonctionnement, en différentes catégories principales. Ensuite, une description détaillée de chacun des dispositifs est présentée. Cette description est constituée d'un schéma de fonctionnement, d'un état de l'art exhaustif et d'un recensement des avantages et des inconvénients caractérisant le dispositif considéré.

Le deuxième chapitre est consacré à l'analyse du fonctionnement des machines à absorption. Le but est de proposer une méthode d'analyse et de comparaison en vue de l'exploitation de ces machines dans de différents endroits du globe. Plus précisément, le système étudié est constitué d'un refroidisseur à absorption (à savoir, $LiBr/H_2O$ a simple-effet) directement relié à des capteurs solaires et sans stockage de la chaleur. Pour décrire le comportement du système, un modèle dynamique fondé sur les équations du bilan thermique, pour les capteurs solaires et les générateurs, est établi puis résolu à l'aide de la méthode des volumes finis. Le modèle ainsi obtenu est validé par utilisation de résultats expérimentaux. Ensuite, une série de simulations du modèle dynamique est réalisée pour de nombreux sites (à savoir des villes types) du globe en considérant le potentiel énergétique pour des journées types. Il convient de préciser que les sites considérés ont été choisis sur différentes latitudes afin d'établir une caractérisation globale. La série de simulations ainsi réalisée a permis de définir un facteur d'efficacité, noté par la suite par $\mathcal{EF}$. Ce facteur d'efficacité annuel est calculé en tenant compte des besoins en refroidissement du site considéré et des coûts initiaux opérationnels du dispositif pour chaque site.

Le troisième chapitre est constitué d'une revue détaillée des systèmes de refroidissement hybride existants. Plus précisément, les systèmes sont regroupés en cinq catégories principales selon la technologie de base utilisée, à savoir une machine à compression, une machine à absorption, une machine à adsorption, un "desiccant-evaporative" ou un multi-évaporateur. Pour chacune des cinq classes, une description des différentes configurations est donnée, puis, une synthèse des avantages de chaque hybridation est présentée. Ensuite, un schéma de sélection selon des paramètres de fonctionnement est proposé. Plus précisément, ce schéma permet de choisir un système de refroidissement propre et assurant une réduction de la consommation d'énergie dans un bâtiment.

Le quatrième chapitre traite du problème de dimensionnement des systèmes de refroidissement. Plus précisément, on considère une maison type et on suppose que la production énergétique, pour satisfaire les besoins en refroidissement, est assuré conjointement par un système individuel hybride basée sur des sources renouvelables et par le réseau électrique. La topologie du système hybride de production énergétique est supposé être fondée sur le schéma de sélection proposé dans le troisième chapitre. L'objectif est double. D'une part, le but est de concevoir un système hybride optimal permettant de minimiser l'énergie consommée issue du réseau. D'autre part, la réalisation de la configuration optimale doit être effectuée en tenant compte de l'espace disponible dans le bâtiment, du potentiel énergétique et du coût économique relatifs au site considéré. Pour ce faire, la problématique de dimensionnement est formalisée comme un problème mathématique d'optimisation sous contraintes. Ensuite, une solution est obtenue et discuter dans le cas des sites choisis, à savoir Marseille-France et Beyrouth-Liban.

La dernière partie est constituée d'une conclusion générale et des perspectives à ces travaux de thèse.

1.Generalities on individual cooling systems

Nomenclature

Abbreviations

ABSC	*Absorption Cooling System*
ADSC	*Adsorption Cooling System*
COP	*Coefficient Of Performance*
CT	*Cooling Tower*
DA	*Diffusion-Absorption*
DE	*Double Effect ABSC*
DEC	*Direct Evaporative Cooler*
DBT	*Dry Bulb Temperature*
DSC	*Desiccant System*
EC	*Evaporative Cooler*
EJC	*Ejector Cooling System*
ER	*Einstein Refrigerator*
GAX	*Generator Absorber Heat Exchanger*
GCHP	*Ground Coupled Heat Pump*
H_2	*Hydrogen*
HGCHP	*Hybrid Ground Coupled Heat Pump*
HPG	*High Pressure Generator*
HPGAX	*High Pressure Generator Absorber Heat Exchanger*
IEC	*Indirect Evaporative Cooler*
LD	*Liquid Desiccant System*
LiBr	*Lithium Bromide*
LPGAX	*Low Pressure Generator Absorber Heat Exchanger*
MPG	*Middle Pressure Generator*
NCR	*Nocturnal Cooling Radiator*
NH_3	*Ammonia*
ORC	*Organic Rankine Cycle*
OTSDC	*One-Rotor Two-Stage Rotary Desiccant Cooling System*
PFHE	*Plate–Fin Heat Exchanger*
PV	*Photovoltaic*
SD	*Solid Desiccant System*
SE	*Single-Effect ABSC*
SIEC	*Semi-Indirect Evaporative cooler*
SSLC	*Separating Sensible and Latent Cooling*
TSDC	*Two-Stage Desiccant Cooling System*

VCC	*Vapor Compression Cooling System*
WBT	*Wet Bulb Temperature*

Variables

A	*Area* $[m^2]$
COP	*Coefficient of performance* $[-]$
E	*Energy* $[KJ]$
ES	*Energy saving* $[\%]$
HR	*Heat of condensation rejected* $[\%]$
I	*Solar radiation* $[W/m^2]$
L	*Length* $[m]$
$\dot{m}$	*Mass flow rate* $[kg/s]$
P	*Pressure* $[bar\ or\ Pa]$
PS	*Power saving* $[\%]$
Q	*Capacity* $[W\ or\ KW]$
RH	*Relative humidity* $[\%]$
T	*Temperature* $[^oC\ or\ K]$
t	*Time* $[s\ or\ min]$
V	*Volume* $[m^3\ or\ L]$
Vel	*Velocity* $[m/s]$
$\dot{W}$	*Power* $[W\ or\ KW]$
ϵ	*Exergy efficiency* $[\%]$
ϕ	*Diameter* $[mm]$
η	*Efficiency* $[\%\ or\ -]$
ν	*Entrainment ratio of the ejector* $[-]$
ω	*Air humidity ratio* $[g/kg]$

Subscripts

a	*Ambient*
ab	*Absorber*
ad	*Adsorber*
av	*Average*
c	*Cooling*
ca	*Cooling air*
cd	*Condenser*
cl	*Solar collector*
cp	*Compressor*
cs	*consumption*
d	*Desorption*
dc	*Discharge*
dh	*Dehumidification*

ds	*Desiccant solution*
el	*Electric*
ev	*Evaporator*
exp	*experiment*
gc	*Gas cooler*
gn	*Generator*
go	*Geothermal*
h	*Heating*
hs	*Heat source*
hw	*Hot water*
i	*Inlet*
l	*Latent*
max	*Maximum*
min	*Minimum*
o	*Outlet*
op	*Optimal*
pc	*Pre-cooling*
ph	*Pre-heating*
rg	*Regenerator / Regeneration*
rs	*Rich solution*
sc	*Sub-cooling / Sub-cooler*
set	*Set point*
sim	*Simulation*
sk	*Sink*
sp	*Supply*
th	*Thermal*
to	*Total*
w	*Water*
ws	*Weak solution*
wt	*Waste*

Symbols and abbreviations used in the table

Type of study

E	*Experiment*
S	*Simulation*

Obtained results

$\uparrow$	*Increasing or Increase*
$\downarrow$	*Decreasing or Decrease*
$\rightarrow$	*Implies*
$>$	*Greater*

1.1 Introduction

The rapid growth of the world's population and the raised comfort level have imposed a significant increase in energy demand. Consequently, the CO_2 emissions augmented, and climatic change is increasing at a considerable rate, with an important global temperature rise. Subsequently, the demand for cooling and air conditioning has increased continuously throughout the last decades.

Among building services, the energy use of heating, ventilation, and air conditioning (*HVAC*) systems is significant. It consists about half of the total energy consumption in buildings in developed countries, and around $10 - 20\%$ of the total energy consumption in developed countries [2]. The most common cooling systems used all over the world are based on vapor compression chiller, absorption chiller and adsorption chiller. Besides, desiccant dehumidifier and evaporative cooler are used to control the relative humidity in buildings.

Actually in the literature, numerous works were dedicated to the study of cooling systems. Solar cooling is the most common system reviewed recently ([6] [7] [8], etc.). Thanks to pollution-free working fluids, solar thermal cooling is found more suitable than solar electric cooling, notably in areas where solar energy is always available. In addition, higher capacity and better coefficient of performance (*COP*) [a] could be achieved using solar hybrid cooling systems which are mostly based on absorption chiller and mainly $LiBr/H_2O$ machine [8]. Despite the great benefit of renewable energy-based cooling system, the heat pump or vapor compression chiller remains the widest used to cover both heating and cooling needs.

The present thesis focuses on renewable energy-based cooling system, and its contribution in the reduction of primary energy consumption and the dependency on grid electricity. Two kinds of cooling were studied : (1)the first consists of an absorption chiller connected directly to a solar collector and (2)the second consists of a hybrid cooling system connected to different types of renewable energy systems.

This chapter is dedicated to the generalities on cooling systems used for building cooling, with the recent studies and different types of energy sources driving each machine. It consists of general information about cooling systems, and represents the base of the study addressed in the present thesis.

The remaining sections of this chapter are organized as follow. In the main sections, the cooling systems are categorized into main categories according to the

a. The COP of a machine is defined as the ratio of energy produced to the work required principle of operation. Then, the cooling systems of each category are presented. A detailed description with the state of the art, advantages and disadvantages for the most common cooling systems used in *HVAC* are stated.

1.2 Electrically driven cooling system

The main cooling machines powered by electrical energy are : vapor compression chiller, compressor driven metal hydride cooling system, thermoelectric cooler and vacuum cooler. Vapor compression chiller is the most electrical system used for air conditioning. The compressor driven metal hydride cooling system is investigated in the literature ; however, its application is very limited. Besides, thermoelectric cooler is used for equipment used by military, medical, telecommunications organizations, etc. and vacuum cooler for food cooling.

1.2.1 Vapor Compression Chiller (*VCC*)

1.2.1.1 Principle

VCC consists of four main components as shown in fig.1.1 : a compressor, a condenser, an expansion valve and an evaporator. In the evaporator, a low pressure, low temperature liquid is converted into saturated vapor or even superheated, thus the heat from the zone to be cooled is absorbed. Then, the vapor leaving the evaporator is compressed to a high pressure and temperature vapor in the compressor. Next, the hot compressed vapor is condensed and transformed to a sub-cooled liquid in the condenser, thus releasing heat to the surrounding environment. Finally, the high pressure and temperature liquid, leaving the condenser, runs through an expansion valve where it is expanded and cooled before moving to the evaporator. *VCC* can be classified, according to the type of refrigerant, into two main categories : subcritical and trans-critical. In the second type, carbon dioxide (CO_2) is the refrigerant often used [9].

1.2.1.2 Advantages

- The size of the machine is the smallest for a given refrigeration capacity compared to other heat pumps ;
- The system can be employed over a large range of temperatures from -40^oC to 7^oC [10] ;
- The *COP* of a *VCC* is quite high [11] ;

- The mass flow rate in a *VCC* is relatively low.

1.2.1.3 Disadvantages

- A *VCC* is noisy due to moving parts in the compressor ;
- The refrigeration capacity of the evaporator in a *VCC* drops rapidly with lowered evaporator pressure ;
- At partial load, a poor performance is obtained ;
- A *VCC* is highly dependent on electrical energy ;
- It is essential to superheat the vapor refrigerant leaving the evaporator to avoid liquid hammer in the compressor ;
- Refrigerant leakage in a *VCC* is the major problem for environment pollution.

1.2.1.4 State of the art

One of the most important advantages of *VCC* is the possibility of being used easily either for cooling or heating [12]. A lot of works can be found in the literature concerning the investigation of *VCC*. The operating conditions of *VCC* were studied in numerous works ([13] [14]). Besides, different technologies concerning *VCC* were introduced and studied, recently. The method of separating sensible and latent cooling (*SSLC*)was investigaed in [15]. It consists of two parallel *VCC* cycles, where the first removes the sensible cooling load and the second deals with the latent load. On the other hand, the use of two *VCC* cycles in series, where the first cycle is used to subcool the second one is studied in [16], [17] and [18]. It is found that the use of two *VCC* cycles either in parallel or in series could achieve an improvement on the *COP* and the cooling capacity of the system.

In order to reduce the dependency of *VCC* on grid electricity, different methods were proposed such as ice storage, solar driven *VCC*, thermally activated cooling system, etc. In [19], the authors determined the cost-optimal design of a *VCC* combined with an ice storage tank for various electricity tariff schemes. A thermally activated cooling system consisted of an organic rankine cycle (*ORC*) and a *VCC* was studied in [20] and [21]. Two methods to improve the *COP* of the system were proposed [21] : the first concept is adding a secondary recuperator in the *ORC* and the second one is using different working fluids in the power and cooling cycles. The authors in [22] studied the possibility of using solar electricity to drive the compressor of a *VCC*.

A ground heat exchanger is usually used as a heat rejecter for *VCC*. A new heat rejection system consisted of a ground coupled heat pump (*GCHP*) coupled with nocturnal cooling radiator (*NCR*) was proposed and simulated in [23]. Based on life cycle cost comparison, it is noted that the hybrid ground coupled heat pump (*HGCHP*) is suitable for buildings with high cooling load, even in humid subtropical climates or for buildings with limited surface land areas. Besides, the possibility of using a *HGCHP* in parallel with cooling tower - was studied under different conditions [24]. The systems with ground heat exchanger show a great reduction in operation cost and an improvement of the *COP*. However, the installation of such system requires a great surface which limits its application only for rural buildings or in areas less occupied.

Different comparative studies based on *VCC* cycle have been done. In [25], the authors compared the thermodynamic performance of four small-capacity portable coolers of different cooling technologies namely thermoelectric cooler, Stirling, and two *VCC* with two different compressors types which are reciprocating and linear compressors. The results showed that the Stirling and the reciprocating *VCC* have the maximum overall thermodynamic efficiency. However, the overall efficiency of the (*TEC*) was the lowest due to high internal and external irreversibilities. An economic study for three diffrent cooling system was performed in [26]. They compared the overall cost for three air-conditioning systems located in Alexandria-Egypt, namely a single effect $LiBr/H_2O$ absorption chiller, a double effect $LiBr/H_2O$ absorption chiller and a *VCC*. They found that the total cost of the double-effect absorption chiller is lower than that of the single-effect absorption chiller and *VCC*. Moreover, the total cost of the single-effect one is higher than that of *VCC*. The comparison of a solar driven rotary desiccant cooling system and a *VCC* installed in Berlin and Shanghai showed that the desiccant cooling system is less electricity consumer and more fresh air producer [27]. On the other hand, the initial cost of the *VCC* is lower than that of a desiccant system, while the use of low-grade temperature reduces the operating cost of desiccant systems. According to [6], the capital investment of solar electric cooling system which consists of a photovoltaic (PV) system connected to a *VCC*) is expected to be the lowest in 2030, when compared to solar thermal cooling systems. This is due to greater *COP* of *VCC* and the cost reduction of *PV* technology. In addition, the solar electric cooling will be the most environmentally friendly since it has lower projected CO_2 emission values than solar thermal technologies.
The summary of studies conducted on *VCC* is presented in Table 1.1.

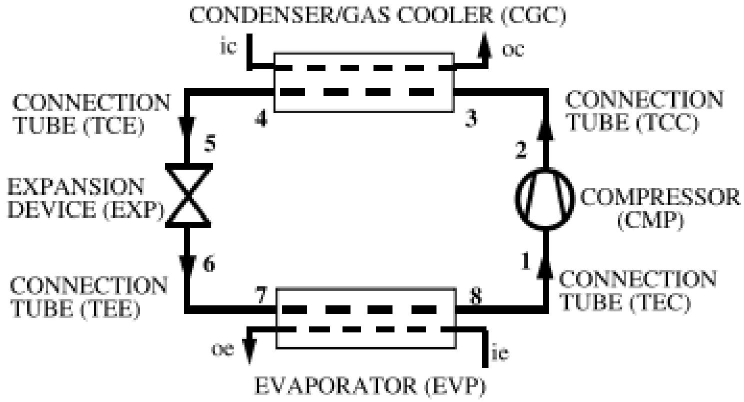


Figure 1.1 – Single stage vapor compression refrigeration cycle scheme[28]

1.2.2 Compressor driven metal hydride cooling system (*CDMHC*)

The schematic of *CDMHC* is shown in fig. 1.2. *CDMHC* is an environmental friendly technology based on the ability of certain metals to absorb and desorb hydrogen (H_2). H_2 is pumped by a compressor which creates a pressure difference between two metal reactors. The major disadvantage of this technology is the result of parasitic losses. The *COP* of *CDMHC* may approach or exceed that of a *VCC* by reducing parasitic losses [29]. In *CDMHC*, the problem of compressor overloading during start-up and during sudden load changes is less severe compared to *VCC* [30]. Park et al. [31] and Mazumdar et al. ([32] [33] [34]) investigated the operating characteristics leading to better performance and cooling capacity of *CDMHC*. They found that the properties of metal hydride have a great influence on system performance. Besides, a high metal conductance, a high compressor efficiency, a low compressor pressure ratio and a low mass ratio of slurry to hydrogen lead to a high *COP*.

1.2.3 Thermoelectric cooler (*TEC*)

The principle of *TEC* is based on the Peltier effect where the heat is transferred when an electric current passes across a junction between two materials. The schematic of *TEC* is shown in fig.1.3. *TEC* is a clean technology that could be powered directly by a *PV* cell. However, it is very expensive and consumes a large amount of electricity. The *COP* and the cooling capacity of *TEC* could be maximized by optimizing the thermal conductance allocation between the hot and the cold side ([35] and [36]) or using an appropriate current ([37] [38] [39]). Moreover, the *COP* is affected by the temperature difference between the two

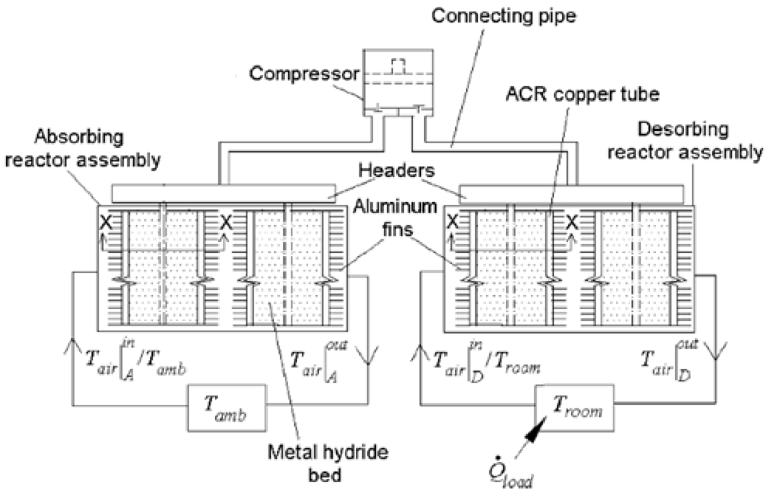


Figure 1.2 – Schematic Compressor driven metal hydride cooling system [34]

sides and a multi-stage *TEC* could be used when a large temperature difference is required.

There are different parameters that influence *TEC* performance such as the *TEC* module parameters, the thermal resistances of heat exchangers, the heat sink temperature, the allowable temperature of the cooled object and the applied electric current [35]. According to [37], *TEC* at transient cooling may reach a lower temperature than that reachable at the steady state, if a current much higher than the steady state optimum current is applied. The contribution of solar electricity in a *TEC* was also studied ([40] and [41]). In [40], the authors observed that the solar insolation has a great effect on the performance of solar driven *TEC*, and a temperature in the range $5 - 10^oC$ could be maintained in the refrigerator at optimum operating conditions. In [41], the authors tested a *TEC* connected to PV/T solar panel. A *COP* greater than 0.45 were achieved. However, the system couldn't achieve a temperature lower than 17^oC. The thermal energy gained both from the hot side of *TEC* and *PV/T* modules was able to rise the domestic water temperature by about 9^oC in $18.5L$ storage tank.

1.2.4 Vacuum Cooler

A vacuum cooler is a rapid cooling method which is 60 times quicker than conventional air cooling [42]. It is used to remove heat field and, thus, to extend shelf life and improve the quality for many types of horticultural and floricultural products. The difference between vacuum cooling and conventional refrigeration methods is that some water from product is directly evaporated in a vacuum chamber. Therefore, only products containing free water can be vacuum cooled [43]. A simple layout of vacuum cooler is shown in fig. 1.4.

Table 1.1 – Summary of studies conducted on *VCC*

	System description		**Obtained results**			
work (type of study)	**Type of Machine**	**Operating principle**	**COP and/or Efficiency (η)**	**Energy and/or Power**	**Size and/or Cost**	**Operating conditions**
[12] (E)	SC, SS	Three operating mode : -Cooling (CM) -Heating (HM) -Simultaneous (SM)	-For HM or SM : ($T_{ev,i}$ ↑) → (COP↑) -For CM or SM : ($T_{cd,i}$ ↑) → (COP↓) -COP_{SM} > COP_{HM} > COP_{CM}			
[15] (S)	SC, SS	SSLC using two VCC : -One VCC for sensible cooling ; -The other for latent cooling		ES (30%)	Compressor size ↓ (25%)	
[16] (S)	SC, CS	Using mechanical sub-cooling	Use of refrigerant R134a → (COP ↑)		Use of refrigerant R134a → Best relative compressor size[1]	
[17] (S)	SC, CS	Using mechanical sub-cooling	COP affected by fouling of the condenser	-Q_{ev} and $\dot{W}_{cp}$ affected by fouling of the evaporator ; -$\dot{W}_{cp,sc}$affected by fouling of condenser		
[18] (E)	SC, CS	Using mechanical sub-cooling	η ↑ compared to system without sub-cooling	-Q_{ev} ↑ -$\dot{W}_{cp}$ ↑		

[22] (S)	SC, SS	Solar electricity to drive the compressor of VCC	COP $\in$ (3.04-4.07)		-T_{ev}=10°C $\rightarrow$ $A_{PV} \approx$ 19 m^2required -T_{ev}= -10°C $\rightarrow$ $A_{PV} \approx$ 39 m^2 required	
[21] (S)	SC, SS	A combined cycle of ORC and VCC	Adding sub-cooling and cooling recuperation $\rightarrow$ COP $\uparrow$ (22%)			
[20] (E)	SC, SS	A combined cycle of ORC and VCC		About half of waste energy is converted into cooling		
[19] (S)	SC, SS	VCC combined with an ice storage tank		Cooling power$\downarrow$ (26%)	Cost $\downarrow$ (8%)	
[13] (E,S)	TC, SS	Working with semi hermetic compressor	COP $\downarrow$ (23%)	-$Q_{ev} \downarrow$ (20%) -$\dot{W}_{cp} \uparrow$ (5 %)		Great influence of P_{ev} and P_{ds}on SH
[14] (S)	TC, SS	Working with ideal compressor with isentropic efficiency equal to 1				(Δ($T_{gc,o}$,T_{ev}) > 45 ^{o}C) $\rightarrow$ > $T_{cp,dc}$ is obtained
[23] (S)	SC, SS	GCHP combined with NCR as heat rejection system		-In heat pump : ($\dot{m} \uparrow$), ($L_{GHE} \uparrow$) or ($A_{NCR} \uparrow$) $\rightarrow$ ($E_{cs} \downarrow$) -In heat rejection system : ($\dot{m} \uparrow$)$\rightarrow$ ($E_{cs} \uparrow$)	Cost$\downarrow$ (10.22%) during 10 years operation compared to VCC with GCHP only	In heat rejection system : if ($\dot{m} < \dot{m}_{min}$) $\rightarrow$ (T_{cd} > 40°C)
[24] (E)	SC, SS	GCHP combined with CT connected in parallel	(T_i at CT=28°C) and (T_i at GCHP=40°C) $\rightarrow$ COP $\uparrow$ (21 %) compared to GCHP standing alone	Compared to GCHP standing alone : -PS =5% -HR $\downarrow$ (42%)		
[1] : Size of main cycle compressor to sub-cooling cycle compressor						

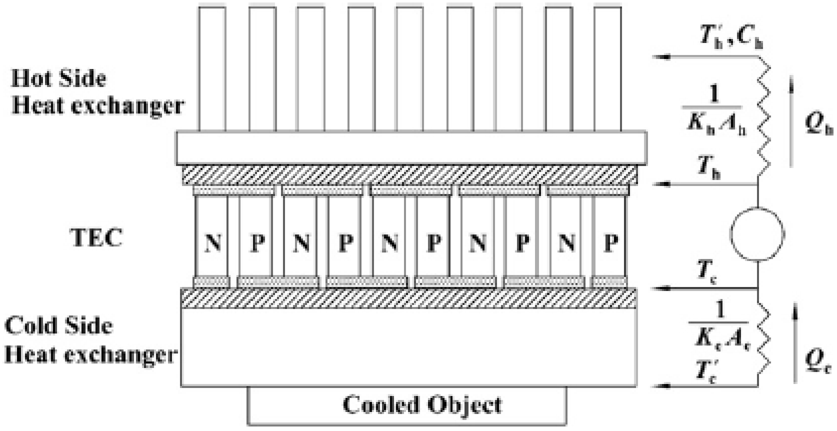


Figure 1.3 – Schematic of Thermoelectric cooler [35]

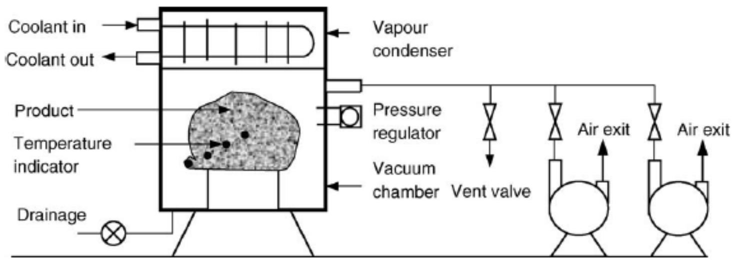


Figure 1.4 – Layout of vacuum cooler [43]

1.3 Thermally driven cooling system

Thermally driven cooling systems consist of sorption systems based on sorption process and ejector cooling systems. The sorption process is a physical or chemical process by which one substance becomes attached to another. The most common thermally driven sorption systems used for building *HVAC* are absorption and adsorption systems as well as desiccant dehumidifier. Also, heat-driven metal hydride cooling machine is a type of sorption systems.

1.3.1 Absorption cooling system (*ABSC*)

1.3.1.1 Principle

The *ABSC* principle is based on the ability of a liquid to absorb and desorb vapor, of another fluid, whose solubility varies with temperature and pressure. It consists mainly of four heat exchangers : a generator, an absorber, an evaporator and a condenser as shown in fig.1.5. The refrigerant vapor, resulting from the generator, is cooled down in the condenser at high pressure. The condensed refrigerant is evaporated again at low pressure in the evaporator, thereby extracting heat from the medium to be cooled. Meanwhile, the weak solution (poor in refrigerant) is then sprayed over the top of the absorber where the refrigerant vapor leaving the evaporator is absorbed. The condensing heat in the condenser and the mixing heat in the absorber are rejected to a cooling fluid. $LiBr/H_2O$ and H_2O/NH_3 are the most mixtures used. In a *ABSC* where volatile absorbent are used such as H_2O/NH_3, the cycle requires a rectifier to separate the water evaporated from NH_3 before entering in the condenser.

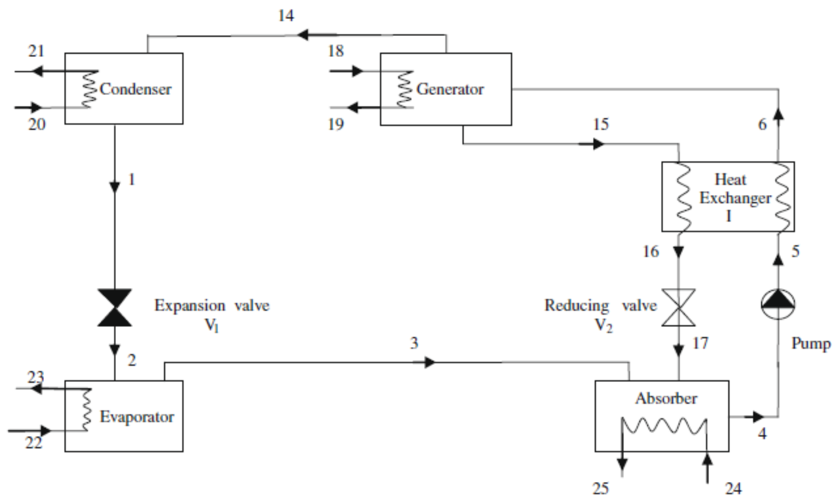


Figure 1.5 – Schematic of the absorption cooling machine [44]

1.3.1.2 Advantages

- May be driven by a low grade heat source such as solar energy, waste heat, natural gas, biomass, etc. and thus electricity consumption is reduced ;
- Environmentally friendly working fluid pairs ;
- Operates over a wide range of heat source temperatures from 70^oC to 230^oC ;

- Can be water cooled or air cooled for small machine size ;
- Possibility of multi-staging to improve *COP* for high-temperature heat sources ;
- Quiet operation since only one pump is used ;
- Reduction of electricity demand during peak cooling seasons by shifting to gas or oil [45] ;
- Possibility of no moving part for a small system [46] ;
- Can be direct fired or exhaust fired.

1.3.1.3 Disadvantages

- Low *COP* compared to *VCC* where the maximum *COP* reached doesn't exceed 1.7 [47] ;
- Requires high source temperature compared to adsorption machine ;
- Crystallization at high concentration in case of $LiBr/H_2O$ machine ;
- Rectification is required in case of H_2O/NH_3.

1.3.1.4 State of the art

There are different types of *ABSC* studied in the literature namely single effect, double effect, triple effect, diffusion-absorption, generator-absorber heat exchanger (*GAX*), etc. The single effect *ABSC* has the simplest design, however it has a poor *COP* compared to other types. It is found that the *COP* of single effect cycle is approximately equal to the half of double-effect *COP*, and slightly greater than the third of the *COP* of triple effect system [44]. The authors in [48] tested a new design of air-cooled *ABSC* where double effect and *GAX ABSC* are combined. Diffusion-absorption is an *ABSC* similar to a single-effect absorption cycle but with same total pressure in the entire cycle : the pressure difference is compensated by circulating an inert gas (H, He, NH_3, etc.) between the evaporator and the absorber. A diffusion-absorption machine consists mainly of an evaporator, a combined condenser/absorber, and a generator. In the condenser/absorber, the pressure equalizing vapor is absorbed by the absorbent ; thus the refrigerant wills condense due to the increase of its partial pressure. The Einstein refrigerator is a type of diffusion-absorption machine where butane is the refrigerant, and the water is the absorbent. Ammonia acts as the inert gas in the cooling cycle. Based on the literature review, it is found that complex *ABSC* achieves higher performance compared to single effect, but with much greater initial cost. Accordingly, the acceptable price of single effect *ABSC* make its used suitable for small application such as small residence.

Numerous works investigated the solar *ABSC* ([49] [50] etc.), a suitable solution to reduce the dependency on grid electrcity and to minimise the energy consumed by cooling systems. Evacuated tube collector and flat plate collector are the most solar collectors used with *ABSC*. An *ABSC* working with $LiCl/H_2O$ was studied and compared with $LiBr/H_2O$ *ABSC* in [51]. In a $LiCl/H_2O$*ABSC*, during the desorption process, when the solution reaches saturation point, solid crystals of $LiCl$ are formed and fall under gravity in the vessel, which leads to high energy density storage in the solid crystals and good heat and mass transfer. The $LiCl$ crystals do not move in the cycle ; only, the saturated solution is pumped over the heat exchanger in the evaporator/condenser where it absorbs the vapor evaporated and thus producing a cooling effect. The solution becomes unsaturated and returns into the reactor containing the $LiCl$ crystals, where some of the crystals are dissolved and the solution is saturated again [52]. The comparison results indicated that similar solar fraction and primary energy ratio were achieved in both systems with the difference of an additional surface required for external storage in $LiBr/H_2O$ system.

As in *VCC*, the ground cooling is used to eliminate and reduce the energy consumption of cooling towers. A solar *ABSC* installed in Almeria, Spain, where the cooling tower is replaced by a shallow geothermal system was tested in [53]. The authors in [54] compared three different heat rejection systems (dry, wet and geothermal) used in a solar *ABSC* located in Rimsting, Germany. On the other hand, geothermal energy provided by sedimentary basins could offer the heat energy required for heat-driven sorption cooling systems in some cases. For instance -according to [55] - a $3\ km$ deep or shallower geothermal sink can produce a hot water at a temperature between greater than 70^oC in the Great Artesian Basin, Australia.
The summary of some recent studies conducted on *ABSC* is presented in Table 1.2.

Table 1.2 – Summary of studies conducted on ABSC

Work (type of study)	System description			Obtained results		
	Type of Machine	**Working fluids**	**Operating principle**	**COP and/or SF**	**Operating conditions**	**Saving**
[56] (S)	DE	-Refrigerant : NH_3 -Absorbent : water	-Air cooled -Solar hot water to drive both HPG and MPG	(T_{gn}=85°C) → -(COP_{th} [1]=0.34) -(COP_{el} [2]=26)	-Q_c affected by : • T_{hw} • T_{ca} • T_{ev} -No influence of T_{ws} on Q_c	
[48] (E)	GAX	-Refrigerant : NH_3 -Absorbent : water	-Air cooled -Two GAX heat exchangers were incorporated to the conventional cycle : (1)LPGAX (2)HPGAX	(COP ↑)	(T_{ev}= -5°C) at (T_{sk}=35°C)	30-40% heat recovery of the total internal heat
[57] (S)	DA	-Refrigerant : NH_3 -Absorbents : $LiNO_3$, $NaSCN$ and Water -Inert gas : He and H_2	Conventional diffusion-absorption system	-at T_{ev}= -15°C : Best COP using NH_3-$LiNO_3$-He -at T_{ev}=7.5°C : Best COP using NH_3-H_2O-He -at T_{ev} < 7.5°C : Best COP using NH_3-$NaSCN$-He	-T_{hs}(NH_3-$LiNO_3$) < T_{hs} (NH_3-$NaSCN$) -(T_{rs} ↑) and (T_{gn} ↓) → preventing crystallization	
[58] (S)	ER	-Refrigerant : butane -Absorbent : water -Inert gas : ammonia	Add a rectifier to purify NH_3 vapor leaving the generator	Water cooled machine : very low COP (0.19)	In case of air cooled machine, T_{ev} limited due to : -the condensation of ammonia -the presence of a small quantity of water -in the evaporator the need of high P in the condenser/absorber	

[49] (E)	SE	-Refrigerant : water -Absorbent : LiBr	-Solar driven 30 KW absorption chiller -Hot and chilled water tanks			-($I_{set}\downarrow$) from 180 W/m^{2}to 250 W/m^2 $\rightarrow$ ($E_{cs}\downarrow$) -The fan of the cooling tower starts at ($T_{cw,set}$= 38^{o}C) and stops when ($T_{cw,set}$ < 33^{o}C) $\rightarrow$ ($E_{cs}\downarrow$) without affecting Q_c
[53] (E)	SE	-Refrigerant : water -Absorbent : LiBr	-Solar absorption chiller -Geothermal heat rejection system			Compared to cooling system that uses a cooling tower , the systems achieves the following savings during a cooling season : -(ES=31%) -CO_2 emissions $\downarrow$ (833kg) -water consumption $\downarrow$ (116m^3)
[54] (S)	SE	-Refrigerant : water -Absorbent : LiBr	-Solar driven 15 KW absorption chiller -Hot and chilled water storage tanks	-SF^3 $\in$ (70-88%) depending on the heat rejection system -Wet cooling tower : COP_{el} $\in$ (6 -11.5) -Dry cooling tower : COP_{el} $\in$ (4-8) -Geothermal heat rejection : COP_{el}=13		-Geothermal heat rejection : (ES=30%) -Replacing the auxiliary heating by auxiliary cooling (VCC), a (PER^4 $\uparrow$) especially if a geothermal heat rejection system is used.

[1] : the ratio between the capacity of the evaporator and the thermal energy consumed in the generator

[2] : the ratio between the capacity of the evaporator and the electric energy consumed by the fan and the pump

[3] : the ratio between the cooling energy provided by solar hot water driven absorption chiller and the total cooling load of the zone to be cooled

[4] :the provided cooling energy divided by the sum of consumed auxiliary electricity and auxiliary thermal energy (if auxiliary heating system is used such as gas burner, boiler, etc.) or auxiliary cooling demand (if auxiliary cooling system is used such as VCC) multiplied by a primary energy factor

1.3.2 Adsorption cooling system (*ADSC*)

1.3.2.1 Principle

ADSC (or also known as thermal compression cooling system) consists of four main components as shown in fig. 1.6 : an adsorber filled with adsorbent, a condenser, an evaporator and an expansion valve. The adsorption cooling cycle is intermittent and comprises mainly two phases : adsorbent heating-desorption-condensation at high pressure and adsorbent cooling-adsorption-evaporation at low pressure. In the first phase, the refrigerant in the bed is desorbed, then condensed in the condenser. During the adsorption phase, the refrigerant is evaporated in the evaporator, then adsorbed by the adsorbent bed. The heat of condensation and adsorption is rejected to a cooling medium. For applications such as air conditioning, two or more adsorption beds should be used to produce a continuous cooling effect.

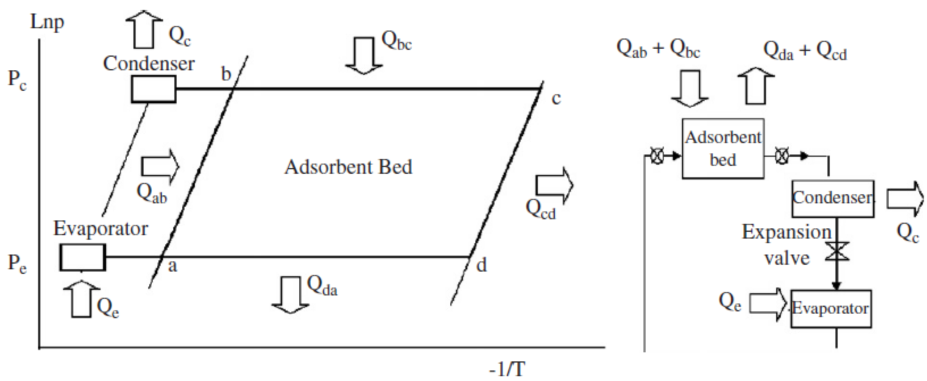


Figure 1.6 – Cycle of basic adsorption heat pump [59]

1.3.2.2 Advantages

- Possibility of being driven by low temperatures as low as 50^oC [60] heat sources such as solar, waste, geothermal, etc. ;
- Driven at a wide range of hot water temperature from 60^oC to 120^oC [47]) ;
- Environmental friendly refrigeration with a zero ozone depleting potential since it can use a natural working fluid as refrigerant ;
- Ability of thermal storage [59] ;
- Operate without noise and vibration [59] ;
- No frequent maintenance is required ;

- driven by lower driving temperature compared to *ABSC*.

1.3.2.3 Disadvantages

- Lower *COP* compared with *ABSC* and *VCC* ;
- Poor heat and mass transfer ;
- Only air cooled machine available in the market ;
- Low specific cooling power [61] ;
- High initial cost compared to *ABSC* and *VCC* ;
- Deterioration of adsorbent material caused by large adsorption/desorption pressure difference ;
- Intermittence of cooling cycle ;
- Requires high technology and special design to maintain high vacuum ;
- Larger volume and weight than *VCC*.

1.3.2.4 State of the art

The low performance and the high initial cost of *ADSC* make its use not compteteive with other cooling systems, namely *VCC* and *ABSC*. Therefore, many efforts were devoted in order to enhance the performance and to reduce the cost of this type of cooling system. To enhance the heat and mass transfer characteristics, the authors in [62] studied the possibility of using an inner vacuum tube filled with zeolite inserted into a larger tubular shell as adsorbent bed. They noted that the adsorption process took longer than the desorption process. On the other side, the possibility of using free energy to drive *ADSC*, such as solar energy and waste heat was also investigated. Solar ADSC was studied and different designs were proposed in [63], [64], [65], [66], [67] and [68]. Moreover, waste heat driven *ADSC* was studied in [69] and [70]. The authors in [69]investigated the performance of a modified *ADSC*- designed for engineering truck air-conditioning - driven by exhaust gas waste heat. They indicated that the number of assembled adsorption modules must be determined according to the required cooling load. The studied system is suitable for engineering vehicle, transportation vehicle, and fishing boat with high temperature waste gas. In [70], a waste heat driven dual evaporator adsorption cycle consisting of two evaporators, three adsorbent beds, and a condenser, where evaporators work at different pressure levels was proposed and modeled. Each adsorbent achieves six processes during a cycle, namely, the desorption process, the pre-cooling process, the lower pressure adsorption process, the higher pressure adsorption process, and the pre-heating process. The authors studied the effect of hot water inlet temperature, chilled water inlet and outlet temperatures, and cycle time on system performance and specific cooling effect. The summary of studies conducted on ADSC are represented in table 1.3.

1.3.3 Desiccant system (Open sorption system)

1.3.3.1 Principle

The desiccant system is mainly based on two processes : dehumidification and regeneration. The process air (hot and humid) enters in the desiccant dehumidifier which removes moisture (Latent load) thanks to the pressure difference between the air and the desiccant surface. In order to reuse continuously the desiccant material, a regeneration source is required to desorb the moisture attracted onto the surface of desiccant material. If the air temperature is still above the comfort level, an additional cooling system is applied to remove the remaining sensible load. The desiccant material can be solid such as silica-gel, activated alumina and zeolite or liquid such as lithium chloride, lithium bromide and calcium chloride. The schematics of solid and liquid desiccant systems are shown in fig.1.7.

1.3.3.2 Advantages

- Running cost lower than that of *VCC* ;
- Can be driven by low-grade heat of about 70^oC such as solar energy, industrial waste heat [74] ;
- Both air temperature and humidity can be simultaneously controlled [75].

Advantages of *LD*

- Possibility of separating absorption and regeneration processes and thus using sorption material as chemical storage ;
- Can achieve higher humidification than solid desiccant at the same regeneration temperature since it is easy to cool the sorption process in liquid desiccant ;
- Operating costs 40% lower than that for *SD* [76] ;
- Lower initial cost than that that for *SD* [76] ;
- Higher *COP* than *SD* ;
- Regenerated at relatively lower temperature [77] and causes lower air side pressure drops [78] compared to *SD* ;

Table 1.3 – Summary of studies conducted on ADSC

Work (type of study)	System description		Obtained results			
	Working fluids	**Operating principle**	**COP and/or Efficiency (η)**	**Operating conditions**	**Energy and/or Power**	**Application**
[62] (E)	-Adsorbent : Zeolite -Refrigerant : water	A shell and tube adsorbent bed, an evaporator, a condenser, heating and cooling baths	COP_{av}=0.25	-($A_{cd}\uparrow$)→ ($t_d\downarrow$)and non-condensable gases are eliminated at $<P$ -τ_{av}=395 min	$SCP^1{}_{av}$= 6.4 W/Kg	Suitable for engineering vehicle, transportation vehicle, and fishing boat with high temperature waste gas
[69] (E)	-Adsorbent : 13X Zeolite -Refrigerant : water	Driven by exhaust gas waste heat	COP influenced by : • T_a • T_{hs} • Vel_a • RH_a			
[63] (E,S)	-Adsorbent : silica-gel -Refrigerant : water	Solar driven adsorption chiller with hot water storage	-(Effect of I) > (effect of $T_{a)}$ on COP -During 8h of operation : COP_{av}=0.35 and $COP_{el,av}$=8.19 -(Height-to-diameter ratio of water tank $\uparrow$) → (COP $\uparrow$) and (η_{cl} $\uparrow$)		During 8h of operation : $Q_{c,av}$=15.3 KW	Suitable for ice making application
[64] (E)	-Adsorbent : silica-gel -Refrigerant : water	Solar driven adsorption chiller with hot water storage	Under Shinchu weather conditions : $\eta_{cl,av}$= 27.3% and COP_{av}=40.3%	Introducing a small buffer tank at the outlet of adsorption chiller→ (fluctuation of $T_{chw,out}\downarrow$)	-T_{hw}=80°C, T_{cw}= 30°, and T_{chw}=14°C→ Q_c=9 KW and SCP=72 W/kg -Under Shinchu weather conditions : $Q_{c,av}$= 7.79 kW	

[65] (E,S)	-Adsorbent : silica-gel -Refrigerant : water	-Solar driven adsorption cooling system -Study of the system without hot water storage tank	($COP_{el} \uparrow$) and ($\eta_{cl} \uparrow$) compared to system with heat water storage	Fluctuation of $T_{cl,out}$ and T_{chw} compared to system with heat water storage	-(ES $\uparrow$) compared to system with heat water storage -same Q_c as system with heat storage especially at $> A_{cl}$	suitable for areas with high solar energy
[70] (S)	-Adsorbent : silica-gel -Refrigerant : water	Two evaporators, three adsorbent beds, and a condenser	-COP$\uparrow$ (70%) compared to single stage ADSC -($T_{chw,i} \uparrow$) $\rightarrow$ (COP $\uparrow$) -(Δ ($T_{chw,i}$, $T_{chw,o}$) $\uparrow$) $\rightarrow$ (COP $\uparrow$)	($t_{ph} \uparrow$) and/or ($t_{pc} \uparrow$) $\rightarrow$ fluctuation of $T_{chw} \downarrow$	-SCP $\uparrow$ (50%) compared to single stage ADSC -($T_{chw,i} \uparrow$) $\rightarrow$ (SCP $\uparrow$)	
[71] (E,S)	-Adsorbent : granular activated carbon -Refrigerant : R134a	1.5 kW electric heater is used to heat water for the regeneration of the adsorbent	-At T_{hw}=373 K and T_{ev}=293K $\rightarrow$ $COP_{max,sim}$=0.35 -($T_{hw} \uparrow$) $\rightarrow$ (COP$\uparrow$) -($\tau \uparrow$) at constant T_{ev} $\rightarrow$ (COP$\uparrow$) -($\tau \uparrow$) at constant T_{hw} $\rightarrow$ (COP $\downarrow$)	τ=900s	-$SCE^2{}_{exp}$= 70 kJ/kg -SCE_{sim}= 83 kJ/kg -($T_{hw} \uparrow$) $\rightarrow$ (SCE$\uparrow$) -($\tau \uparrow$) at constant T_{ev} $\rightarrow$ (SCE$\uparrow$) -($\tau \uparrow$) at constant T_{hw} $\rightarrow$ (SCE$\downarrow$)	
[66] (S)	-Adsorbent : activated carbon -Refrigerant : methanol	Solar driven adsorption chiller	-During hot days : minimum COP (0.42) and $SCOP^3$ (0.12) obtained -During cold days : maximum COP (0.59) and SCOP(0.24)	5-13kg of ice per m^2 of solar collector under Dhahran climate conditions		Activated carbon is suitable for solar adsorption ice-maker

[67] (E)	-Adsorbent : activated carbon -Refrigerant : ethanol	Conventional adsorption chiller	(bed apparent density $\uparrow$)$\rightarrow$ (COP $\uparrow$)	- At the beginning of the adsorption process : (the mass of activated carbon $\uparrow$) $\rightarrow$ (the mass of refrigerant adsorbed per kg of adsorbent $\downarrow$) -At equilibrium state : no influence of bed apparent density		

[1] : defined as the produced cooling power per kg adsorbent

[2] : defined as the produced cooling energy per kg adsorbent

[3] : defined as the ratio of cooling energy to the total diurnal incident solar energy

- Reduces odors and cleans air from bacteria, particulates and microorganisms ;
- Reaches the control humidity point rapidly.

Advantages of *SD*

- Solid desiccants are compact, less subject to corrosion [78].

1.3.3.3 Disadvantages

- Complexity of cooling system.

Disadvantages of *LD*

- Requires more maintenance than *SD* ;
- The carryover of a corrosive desiccant out of the absorber and into the supply ductwork or occupied space is the most important potential problem [76] ;

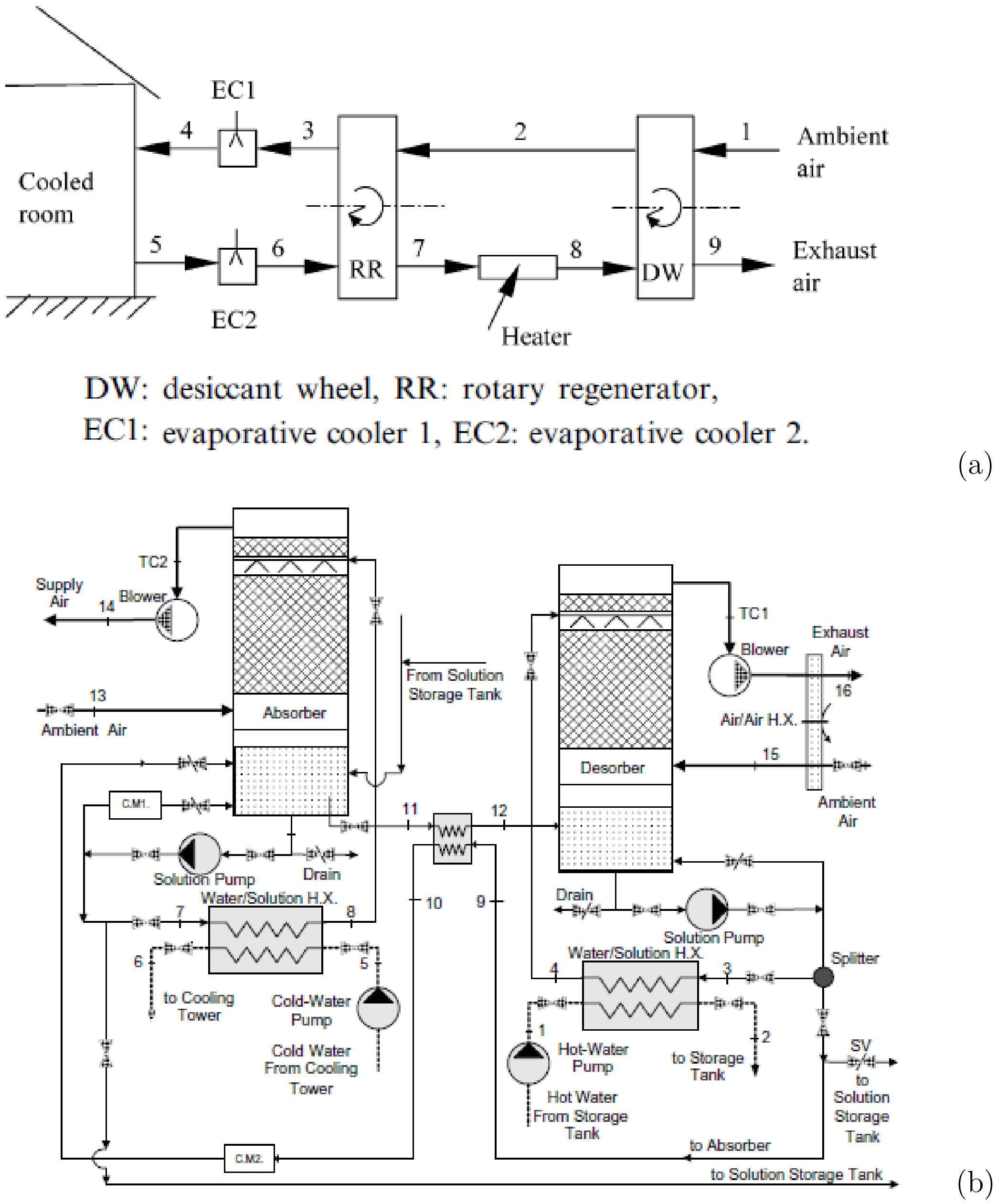


Figure 1.7 – Schematic of desiccant system : (a)Solid Desiccant (*SD*) [72]; (b)Liquid Desiccant (*LD*) [73]

- Requires a cooling tower or chiller to cool the liquid desiccant solution to the desired temperature [8].

Disadvantages of *SD*

- Reduced performance due to leakage between supply air and return air ;
- Process air gains energy during dehumidification due to adsorption heat [79].

1.3.3.4 State of the art

Desiccant systems are usually used with other cooling systems namely *VCC*, *ABSC*, *ADSC* and evaporative coolers. Therefore, few researchers have been interested in studying standing alone desiccant systems. Yin et al. [80] tested the dehumidifier and the regenerator of a liquid desiccant evaporation cooling air conditioning system. They studied the influence of operating parameters (e.g. heating source temperature, air temperature and humidity, desiccant solution temperature and desiccant solution concentration) on the rates of dehumidification and regeneration. Liu et al. [81] studied the effect of regeneration mode on the performance of *LD* using packed bed regenerator. Yin et al. [74] designed and tested an internally cooled/heated dehumidifier/regenerator of *LD*. They compared the internally cooled/heated dehumidification/regeneration with the adiabatic dehumidification and regeneration processes, respectively. Yin et al. [82] developed a mathematical model for internally cooled/heated dehumidifier/regenerator. The model was then validated using aqueous lithium chloride as desiccant and water as heat transfer fluid. Qi et al. [83] developed a simplified numerical model for internally cooled/heated dehumidifier/regenerator by defining three different "parameter" effectiveness namely enthalpy effectiveness, moisture effectiveness and temperature effectiveness ; which are defined as the ratio of the actual "parameter" change of air to the maximum possible one. The results were compared to experimental and simulation results obtained for other models.

Zhao et al. [84] proposed a new design for *SD*. They used two silica gel coated heat exchangers working in parallel to provide continuous dehumidification, and a vacuum tube solar collector to provide hot water to regenerate the silica gel. Dehumidification and regeneration were held simultaneously by dividing the process air into two parts for each heat exchangers. The system was tested under Shanghai summer conditions. Ge et al. [85] experimentally investigated the performance of a novel *TSDC*. A sensible heat exchanger is introduced after each wheel to cool the process air and preheat the regeneration air. In order to reduce the volume of *TSDC*, Ge et al. [86] designed an *OTSDC* by dividing the cross-section of one desiccant wheel into four parts. Ge et al. [87] designed a solar *SD* consisting of two desiccant coated heat exchangers working in parallel to produce a continuous cooling. The simulation results under Shanghai summer conditions showed that in August a switch time less than 2 minutes is required to provide satisfying supply air, due to the extremely high temperature and humidity ratio.

The summary of studies conducted on desiccant dehumidifier are shown in Table 1.4.

1.3.4 Heat-driven metal hydride cooling machine (*HDMH*)

The schematic of *HDMH* is shown in fig. 1.8. Similar to *CDMH*, *HDMH* are based on the absorption and desorption of H_2 with the difference of creating a pressure difference between hot and cold reactors. The *COP* of *HDMH* is quite low compared to other thermally driven systems. Moreover, it is highly influenced by the quantity of H_2 transferred between reactors [88]. The amount transferred could be enhanced by increasing the heat source temperature which increases the pressure difference between the reactors [89]. On the other hand, this amount decreases as the difference between ambient and cooling temperature increases [90].

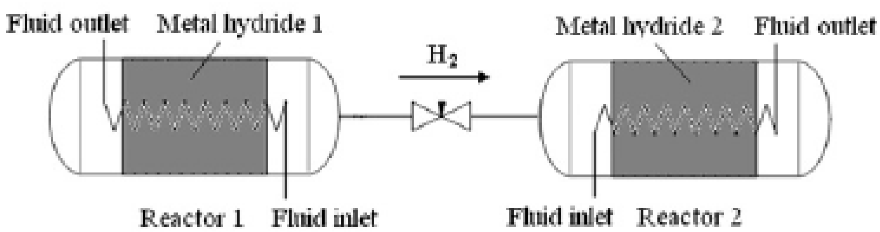


Figure 1.8 – Schematic of Heat-driven metal hydride cooling system [88]

1.3.5 Steam ejector cooling system or jet compression system (*EJC*)

1.3.5.1 Principle

The schematic of *EJC* as well as the cross section of ejector are shown in fig.1.9. A typical ejector consists of a motive nozzle or primary nozzle, a suction chamber, a mixing section, and a diffuser. The working principle of an ejector is based on converting the internal energy and pressure of the motive fluid into kinetic energy. The motive nozzle is a convergent-divergent nozzle which allows the

Table 1.4 – Summary of studies conducted on desiccant dehumidifier

Work (type of study)	**System description**		**Obtained results**	
	Type of desiccant	**Operating principle**	**COP and/or Efficiency (η)**	**Operating conditions**
[80] (E)	LD	Packed bed dehumidifier and regenerator	$\omega_{dh,i}$optimum→ η_{df}maximum	-$\alpha_{rg,av}$=4g/m^2.s -($T_{hs}\uparrow$) → ($\alpha_{rg}\uparrow$) -($T_{air,i}\uparrow$) from 29 to 34.1 °C → (DR ↓) from 0.104 to 0.073 g/s
[81] (S)	LD	Packed bed regenerator	-Hot air driven regenerator : parallel flow (> COP) and counter-flow (< COP) performance -Hot desiccant driven regenerator : counter-flow (> COP) and parallel-flow (< COP) - In packed bed, hot desiccant driven regenerator is preferred	
[74] (E)	LD	-Internally cooled/heated dehumidifier/regenerator -PFHE for dehumidifier and regenerator	Dehumidification : -($T_{cw}\uparrow$) → ($\eta_c\downarrow$) -($T_{cw}\downarrow$) or ($\dot{m}_{air}\uparrow$) → (COP ↑) -($T_{cw}\approx T_{ds,o}$) → (> COP_{PFHE}) Regeneration : -($\dot{m}_{air}\uparrow$)→ (COP ↑) -η_{rg} > than that in adiabatic regeneration	-Dehumidification : ($T_{cw}\uparrow$) → ($\alpha_{dh}\downarrow$) and ($T_{ds,o}\uparrow$) -Regeneration : ($T_{ds,i}\uparrow$)→ ($\Delta\omega\uparrow$) rapidly
[82] (E, S)	LD	-Internally cooled/heated dehumidifier/regenerator -PFHE for dehumidifier and regenerator	-η_{rg} and η_{dh} > than adiabatic ones -Compared to adiabatic ones : (COP ↑) in internally-cooled dehumidifier < (COP ↑) in internally-heated regenerator.	Internally-heated regenerator avoid the dehumidification possibility which may happen in adiabatic one
[83] (S)	LD	Internally cooled/heated dehumidifier/regenerator ;	-Great impact of $T_{ds,i}$on η_{air} -Great influence of $\Delta(T_{hw},T_{cw})$ on η_{ds}	

[84] (E)	SD	-Solar regenerated -Two silica gel coated heat exchangers	-($\tau \uparrow$)$\rightarrow$ ($COP_{th} \uparrow$) -($\omega_a \uparrow$) $\rightarrow$ ($COP_{th} \uparrow$)	-System affected by τ (dehumidification +regeneration) -τ_{op}=600s -under mild conditions : system uses 100% fresh air -at > ω_a : return air is necessary -$T_{hw} \in$ (50-80°C) is required -($\tau \uparrow$)$\rightarrow$ (D $\uparrow$) -($\omega_a \uparrow$) $\rightarrow$ (D $\uparrow$)
[85] (E)	SD	-Two desiccant wheels in series -Two parallel groups of regeneration air	-(> COP_{th}) compared to OSDC -($T_{rg,i} \downarrow$)and ($\omega_{rg,i} \downarrow$) $\rightarrow$ ($COP_{th} \uparrow$) -($T_{air,i} \uparrow$)$\rightarrow$ ($COP_{th} \uparrow$)	-($\omega_{air,i} \uparrow$)$\rightarrow$ (D $\uparrow$) -(> lower operating cost) and (>T_{rg}) compared with OSDC
[86] (E)	SD	One desiccant wheel divided into four parts : Two for regeneration and two for dehumidification	-N=N_{op} $\rightarrow$ COP_{th}=$COP_{th,max}$ -(> COP_{th}) compared with OSDC and TSDC	-(N=N_{op})$\rightarrow$ (D=D_{max}) -($T_{rg} \uparrow$) and ($e_{wheel} \downarrow$) $\rightarrow$ ($N_{op} \uparrow$) -(< T_{rg}) compared with OSDC and TSDC
[87] (S)	SD	Two desiccant heat exchangers working in parallel	-COP is not affected by V_{hw} stored, $\dot{m}_{hw}$ and A_{cl} -SCOP $\approx$ 0.24	($V_{hw} \downarrow$), ($\dot{m}_{hw} \uparrow$) and ($A_{cl} \uparrow$)$\rightarrow$ (Q $\uparrow$)

high-speed jet exiting the nozzle to become supersonic.

An *EJC* consists mainly of four components : a generator, an ejector, a condenser and an evaporator. The liquid refrigerant vaporized at high pressure in the generator enters the motive nozzle where it is accelerated and expanded. Consequently, the resulting vapor entrains the evaporation of the liquid in the evaporator. A low pressure region is created and the cooling effect is produced. The vapor from the generator is then mixed with the vapor from the evaporator in the mixing chamber (refer to fig. 1.9(b)). This mixture is then compressed in the diffuser to an intermediate pressure equal to that in the condenser (refer to fig. 1.9(b)).

1.3.5.2 Advantages

- Simple in design and reliable in operation ;

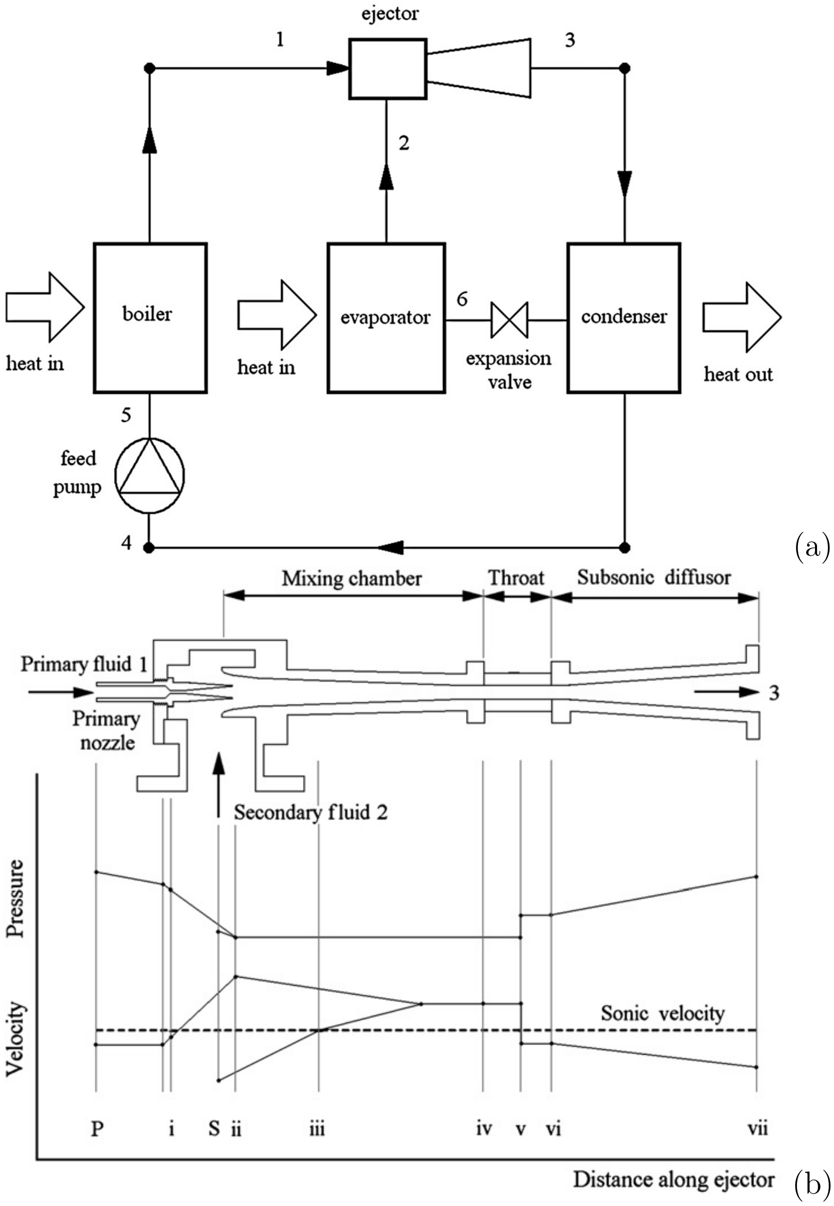


Figure 1.9 – (a) Ejector cooling cycle ; (b) Cross-section of ejector [91]

- Good performance when it functions at part load [a] [92] ;
- Water can be used as refrigerant and furthermore as only working fluid in the whole system ;
- Low initial cost ;
- Capacity to use low a temperature heat source such as flat-plate or va-

a. Fraction of operational load to the nominal cooling capacity of the ejector cuum tube solar collectors [93] or other low grade heat such as geothermal, waste, etc. ;

- No moving parts and low maintenance ;
- Low operational cost.

1.3.5.3 Disadvantages

- Low *COP*, generally around 0.5 [94] ;
- Dependence of the Performance on geometry dimensions of ejector ;
- Inability to generate cooling below 0^oC, thus applications are limited to air conditioning.

1.3.5.4 State of the art

Meyer et al. [91] investigated the possibility of driving a *EJC* by temperatures in the range from 95^oC to 105^oC. Pollerberg et al. [92] tested a solar *EJC* under the weather conditions of Oberhausen, Germany. The system was then simulated for different locations : Essen-Germany, Toulouse-France, Genova-Italy, Safi-Morocco and St.Katrine-Egypt. Varga et al. [93] developed a simplified model-based on energy, mass and momentum conservation equations for a solar *EJC* using water as operating fluid. They presented the simulation results for various climatic conditions of Mediterranean countries. Varga et al. [95] simulated a simple *EJC* using the Computational fluid dynamics. They studied the ejector efficiencies for the primary nozzle, suction, mixing and diffuser. The multifunction generator (*MFG*) combined with ejector cooling has the advantage of eliminating the mechanical pump of the system. (*MFG*) consists of a vapor generator and an evacuation chamber. The cooling jacket of the evacuation chamber depressurizes the generator by producing a cooling effect in order to entrain the liquid from the condenser. (*MFG*) can generate one half-cycle cooling effect ; therefore 2 (*MFG*) are required to produce continuous cooling. Wang et al. [96] tested an *EJC* using 2 *MFG* for thermal pumping. Dai et al. [97] proposed and modeled a combined power and refrigeration cycle. The system consists of an EJC for cooling production and a Rankine cycle for power generation.

The summary of studies conducted on *EJC* is presented in table 1.5.

Table 1.5 – Summary of studies conducted on *EJC*

Work (type of study)	System description	Obtained results			
		COP and/or Efficiency (η)	**Energy and/or Power**	**Size and/or Cost**	**Operating conditions**
[91] (E)	-EJC Driven by T < 100°C ; -The Generator powered by two 4 kW electric elements for auxiliary heating	(T_{gn}=95°C) and (T_{ev}=10°C)→ (COP≈ 0.25) could be reached			-($\varphi_{throat} \downarrow$) → ($T_{gn,min} \downarrow$) -Operation of the system influenced by : T_{gn}, T_{ev}, $P_{cd,cr}$, primary nozzle exit position and the primary and secondary nozzle throat ratio -No influence of ($P > P_{cd,cr}$) and SH of primary steam
[92] (E, S)	1KW EJC connected to a 10.5 m^2 parabolic trough collector	-Great influence of T_{cw} and T_{chw} on COP -($P_{cd} \downarrow$) → (COP ↑) -At part load : ($T_{chw} \uparrow$) → (COP ↑) -Simulation : ($COP_{av} > COP_n$) → ($\eta_{av,to} \uparrow$)		-The simulations were made for the locations Essen in Germany, Toulouse in France, Genova in Italy, Safi in Morocco and St. Katrine in Egypt. -< operation cost obtained in Egypt (0.15 €/kWh) -> operation cost in Germany (0.62 €/kWh)	
[93] (S)	-Solar driven 5KW EJC -Water as refrigerant	COP=0.1 at T_{ev} <10°C → Using water as refrigerant not suitable at T_{ev} <10°C	Solar energy is not sufficient and auxiliary heating required even at high solar radiation	At (T_{cd} >35°C) and (T_{ev} <10°C) → (A_{cl} >50m^2) is required	

[95] (S)	Simple EJC	-Nozzle : η affected by φ_{nozzle} and η independent of the operating conditions -Diffuser : η affected by condenser operating conditions -Suction : ($P_{cd} < P_{cd,cr}$) $\rightarrow$ ($\eta \downarrow$) -Mixing : ($P_{cd}=P_{cd,cr}$) $\rightarrow$ ($\eta = \eta_{max}$)			
[96] (E)	EJC using 2 MFG for thermal pumping	-(T_{gn}=90°C), (T_{cd}=37°C) and (T_{ev}=8.5°C) $\rightarrow$ (COP=0.225) -(COP using R365mfc) < (COP using R141b)	-(T_{gn}=90°C), (T_{cd}=37°C) and (T_{ev}=8.5°C) $\rightarrow$ (Q_c=0.75KW) -(Q_c using R365mfc) < (Q_c using R141b)		
[97] (S)	-Waste driven EJC -Vapor from boiler expanded through turbine to generate power -Ejector driven by exhaust flue gas from gas turbine	-COP is greatly improved if exergy losses are reduced -$\epsilon_{max} \approx$ 27 % obtained at T_{gn} $\approx$ 119°C	Exergy loss in the boiler and ejector are the higher		

1.4 Other cooling system

1.4.1 Evaporative Cooler (Swamp cooler)

1.4.1.1 Principle and types

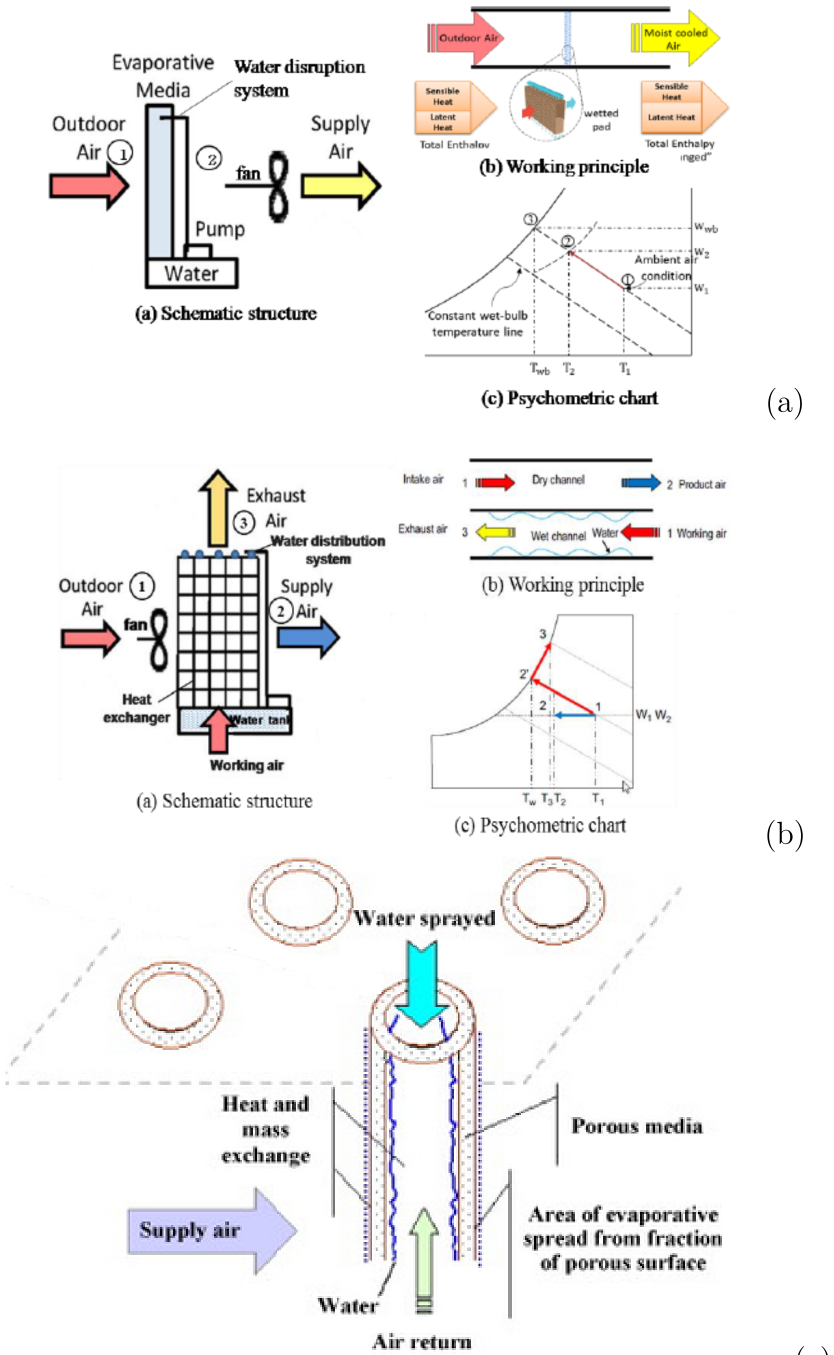
It is based on using heat in the process air for water evaporation. The temperature and vapor partial pressure differences are the main driving forces for heat and mass transfer between air and water. According to [98], there are three types of evaporative coolers (as shown in fig. 1.10) : (1) direct evaporative cooling (*DEC*) where the process air is in direct contact with water, which increases the relative humidity of the process air, (2) indirect evaporative cooling (*IEC*)where the secondary air directly cooled by water evaporation is used to cool process air through a heat exchanger, and (3) semi indirect evaporative cooling (*SIEC*) which is similar to *IEC* with the difference that a porous material plate is used to separate secondary and process air instead of a metal sheet, which allows heat and mass transfer between the two air streams.

1.4.1.2 Advantages

- High *COP* in the range 15-20 [101] ;
- Environmental friendly system since it uses water as cooling medium ;
- Reduced CO_2 and power plant emissions ;
- Reduced electrical energy consumption : it requires only a quarter of the electric power of *VCC* [102] ;
- Improved indoor air quality in dry and hot climate by adding moisture to air ;
- Simple system configuration and operation ;
- Have a lower initial cost compare to the *VCC* system and other air conditioning systems.

1.4.1.3 Disadvantages

- High dependence on the ambient air conditions since the driving force of both evaporative cooler is the temperature difference between the dry-bulb and wet-bulb temperatures of the process air ;
- Smaller temperature reduction potential ;
- High water consumption ;
- *DEC* pads need to be wetted continuously [103] ;



(a)

(b)

(c)

Figure 1.10 – Structure, working principle and psychometric chart of (a) *DEC*, (b)*IEC* [99] and (c) *SIEC* [100]

- Evaporative cooler pads are damaged by deposit of water mineral.

- Growth of bacteria such as legionnaires into the air stream supplied to the room [104] ;
- Only sensible load can be handled [105].

1.4.1.4 State of the art

An evaporative cooler is rarely used individually : it is mostly hybridized with desiccant dehumidification or used as auxiliary cooler combined with another cooling system such as *VCC* or *ABSC*. Heidarinejad et al. [105] designed, constructed and tested a two-stage evaporative cooler consisting of an *IEC* followed by a *DEC*. The experiments were carried out under different climatic conditions in Iran. Sheng and Agwu Nnanna [106] investigated experimentally the effect of system parameters on the cooling performance of *DEC*. Farahani et al. [107] modeled a two stage system consisting of a *NCR*, a cooling coil, and an *IEC*. The chilled water provided by *NCR* is stored in a tank during the night ; and is used to pre-cool the hot process air through the cooling coil during the next day. Then, the cooled air passes through the *IEC* to reach the desired temperature. The secondary air of *IEC* is provided from three sources : outdoor air, the air leaving from the cooling coil, and the air leaving the indirect stage. Khalajzadeh et al. [108] proposed a new cooling system where a *GCHP* is used to pre-cool the air before entering the *IEC*. The *GCHP* comports four vertical ground heat exchangers arrayed in a series configuration. Heidarinejad et al. [109] investigated the performance of a ground-assisted direct evaporative cooling air conditioner. The process air is pre-cooled in a *GCHP* before entering the *DEC*. Riangvilaikul and Kumar [110] investigated, experimentally, the outlet air conditions and the effectiveness of a novel counter flow dew point evaporative cooling system. The system was tested under various inlet conditions representing typical tropical climate conditions. The system presented a good performance at operating conditions, covering dry, moderate and humid climates. Velasco Gomez et al. [111] tested a *SIEC* using ceramic as porous material. They sprayed into the secondary air stream the water cooled in the cooling tower by return air from the conditioned zone. They recommended the use of this cooler as recovery system in climates with high temperature and humidity such as in tropical regions. Velasco Gomez et al. [104] compared the behavior of two *SIEC* where the first system consists of a bank of ceramic pipes arranged vertically and staggered and in the second one the ceramic pipes are replaced by hollow bricks. They found that the second system acts as *DEC*.

The summary of studies conducted on evaporative cooler is presented in table 1.6.

Table 1.6 – Summary of studies conducted on evaporative coolers

Work (type of study)	System description		Obtained results			
	Type of Evaporative Cooler	**Operating principle**	**COP and/or Efficiency (η)**	**Energy and/or Power**	**Size**	**Operating conditions**
[105] (E)	DEC, IEC	IEC followed by a DEC	($\eta \in$ (108-111%)) > ($\eta_{IEC} \in$ (55-61%))	($\dot{W}$ < 33%) compared with VCC	It requires 55% water more than DEC	
[106] (E)	DEC		-$T_w \approx T_{air} \rightarrow$ ($\eta \downarrow$) -(DBT of frontal air $\uparrow$) $\rightarrow$ ($\eta \uparrow$) -(Velocity of frontal air$\uparrow$) or ($T_w \uparrow$)$\rightarrow$ ($\eta \downarrow$)			
[107] (S)	IEC	NCR, cooling coil, and IEC	For weather conditions of Tehran (Use of NCR) $\rightarrow$ ($\eta \uparrow$)			
[108] (S)	IEC	GCHP combined with IEC	For weather conditions of Tehran : COP >1		($L_{GCHP} \downarrow$)	For weather conditions of Tehran : ($T_{air} \downarrow$) and (T_{air} < WBT_a)
[109] (S)	DEC	GCHP combined with DEC	(η >100%)			provide comfort level and replace the VCC
[110] (E)	IEC	Counter flow dew point evaporative cooler	(T_a=30°C) and ($Vel_{air,i}$ <2.5m/s) $\rightarrow$ (η >100%)	For hot and dry climate : If (T_a <45°C) and (ω < 11.2 g/kg) $\rightarrow$ system can operate alone and provide all E_c		

[111] (E)	SIEC	SIEC using ceramic as porous material				-Ceramic material acts as a filter and no mix between the two flows intervening in the exchange process $\rightarrow$ risk of outdoor air contamination is eliminated -(>RH$_{air}$) and (>T$_{air}$) $\rightarrow$ exchange process controlled by T$_{CT,w}$ and RH$_a$
[104] (E)	SIEC	Comparison of two SIEC : -In (a) the porous material is ceramic -In (b) the porous material is hollow bricks.	COP$_{(b)}$ > COP$_{(a)}$	-In (a), E associated to the return air from the chamber can be covered -In (b), Q$_c$ depends on the outdoor air conditions -(Q$_{c,(b)}$ > Q$_{c,(a)}$)		(b) acts as DEC

1.4.2 Magnetic Refrigerator

The principle of magnetic refrigerator is presented in fig. 1.11. It is a based on the magneto caloric effect in certain materials in which a reversible change in temperature is caused by exposing the material to a changing magnetic field. Compression and expansion in *VCC* are replaced in a magnetic refrigerator by adiabatic magnetization and adiabatic demagnetization, respectively. Most refrigerators use Gadolinium as refrigerant material at room temperature [112]. The efficiency of this refrigerator can reach $30-60\%$ of Carnot cycle [113] and running cost 20% less than conventional chillers. However, a high magnetic field $(5-7T)$ is required for remarkable effect [114]. Magnetic refrigerators can produce cooling power up to $600\ KW$ [113], which is suitable for household applications.

1.4.3 Thermoacoustic refrigerator

The schematic of thermoacoustic refrigerator is shown in fig. 1.12. It is based on the heat transfer between a noble gas (such as helium, argon or nitrogen) and a solid porous medium. The temperature change results from the compression and the expansion of the gas by the sound pressure and heat transfer between

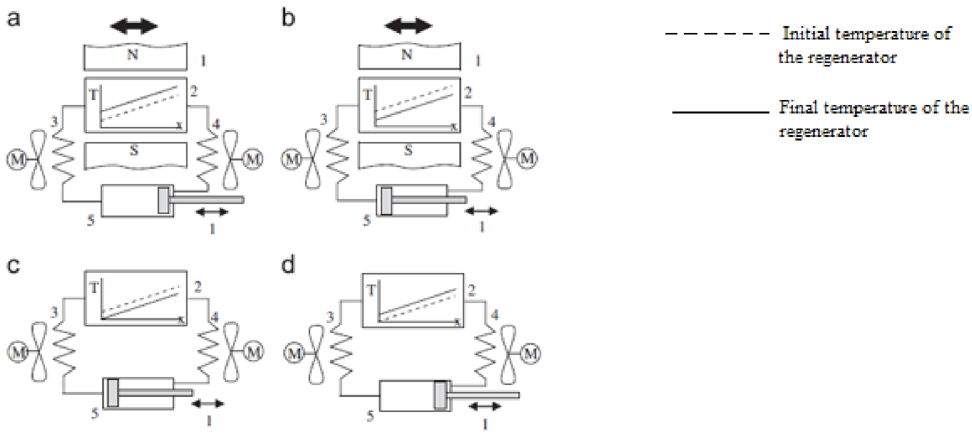


Figure 1.11 – The magnetic refrigeration cycle : (a) Magnetization process, (b) Cooling of the regenerator, (c) Demagnetization process and (d) Heating of the regenerator [112]

the gas and solid medium. The frequency of the acoustic wave has a great influence on the cooling power produced [115]. This refrigerator can be thermally driven using a thermoacoustic engine that converts heat into acoustic energy : The system could achieve a cooling power of $40\ W$ at a cooling temperature of 0^oC when the system operates with Helium as working fluid at a frequency of $234\ Hz$ and a heating power of $300\ W$ [116].

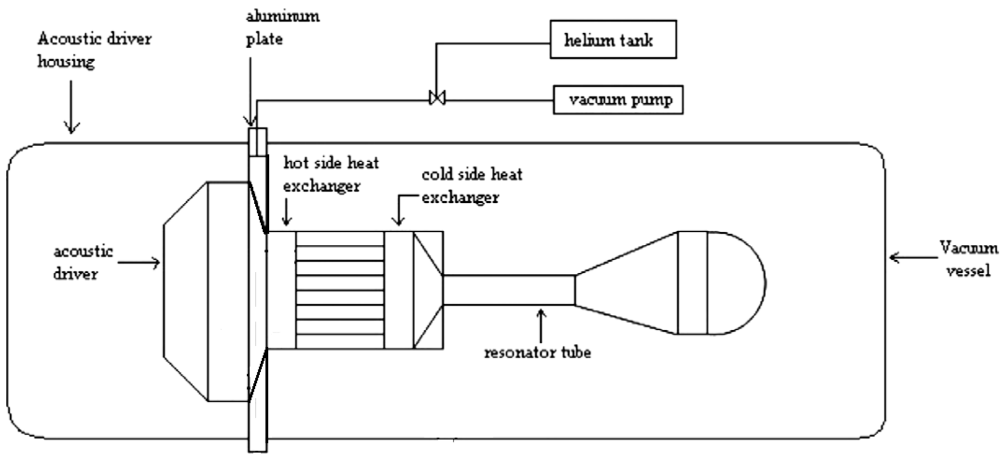


Figure 1.12 – Schematic of a Thermoacoustic refrigerator [117]

1.5 Conclusion

In this chapter, generalities about different types of cooling machine are presented. The most common systems used for building air -conditioning are described in details. An updated state of the art with the recent advances in cooling system for building use are reviewed. Based on the literature review, the most cooling machine used in building are vapor compression chiller, absorption chiller and adsorption chiller as well as desiccant system and evaporative cooler for humidity control. Vapor compression chiller is the most widely use. However, its use is matter of debate due to the negative impact of the refrigerant used on environment and the high electric energy consumption. Consequently, many efforts are devoted to replace *VCC* refrigerants by pollution free fluids, and to replace electric energy by other type of renewable energy. The application of absorption machine for cooling is found appealing due to the possibility of being driven by a low heat source temperature such as solar energy. Solar absorption cooling is commonly investigated in the literature, and the results of its application were found advantageous. Solar adsorption is also an option of environment friendly cooling technology. However, *ADSC* is still noncompetitive with *VCC* and *ABSC* due to its low *COP*.

In the next chapter, the effectiveness of a solar absorption cooling system in representative locations is studied. Thus, the part of *ABSC* consuming energy (generator) is modeled. The model is then, simulated and validated with the experimental results found in the literature. A factor to compare the effectiveness of the solar absorption chiller in different locations at different latitude is defined.

2.Effectiveness Factor of Solar Absorption Cooling System in Representative Locations

Nomenclature

Variables

α	*Solar collector absorbance*
λ	*Thermal conductivity [W/m.°C]*
ρ	*Density [Kg/m^3]*
τ	*Solar transmittance [-]*
ϕ	*Diameter [m]*
$\dot{m}$	*mass flow rate [Kg/s]*
h	*Coefficient of convection [W/m^2.°C]*
t	*Time [h]*
A	*Collector Area [m^2]*
C	*Specific heat [J/Kg. °C]*
$\mathcal{CDH}$	*Cooling Degree Hours [°C.h]*
Dx	*Solar collector division length [m]*
dx	*Generator division length [m]*
E	*Energy [KWh]*
$\mathcal{EF}$	*Effectiveness Factor [-]*
G	*Global solar radiation [W/m^2.°C]*
H	*Day integrated radiation[KWh.°C]*
L	*Solar collector lengh [m]*
P	*Perimeter [m]*
R	*Radius [m]*
S	*Section [m^2]*
T	*Temperature [°C]*

Superrcsipt and Subscript

a	*Ambient*
c	*Cooling*
e	*External*
g	*Generator*
h	*Hourly*
i	*Internal*

r	*Rich solution*
s	*Solar collector*
w	*Wall of copper tube*
HX	*Heat Exchanger*
in	*Inlet*
out	*outlet*
sat	*Saturation*
set	*Set-point*

2.1 Introduction

The demand for cooling and air conditioning has increased continuously throughout the last decades. Conventional energy will not be enough to meet this growing demand in the future, hence the development of cooling systems using renewable energy is accelerating promptly. An alternative solution is solar cooling due the near coincidence of peak cooling loads with the available solar radiation. The use of solar cooling hold promises for reducing the energy consumption and environmental impacts by more than 80% decrease in CO_2 emissions [118]. One of these solar cooling methods is solar absorption chiller.

Solar absorption technologies are not only regarded as environmentally friendly and noiseless alternatives to compression systems, but also as energy-efficient cooling technologies. Besides, using such a system may face different obstacles such that high initial cost, low system performance, low thermal efficiency, and larger cooling tower capacity than that of electric chillers. In addition, the solar energy usage by absorption cooling system is only for a short period during 1-day operation. Hence it is important to study the suitability of this system in different locations.

Given the importance of solar absorption cooling in the field of renewable energy, several works were devoted to this subject. It is found that the most suitable solar cooling technology for a certain location depends on the performance and the cost of the solar collectors [119]. Most of solar cooling systems are based on a $LiBr/H_2O$ single-effect absorption chiller since it has higher coefficient of performance (*COP*) (namely, to be inside the interval $(0.5 - 0.7)$ [47]) compared to other thermal chillers and can be driven by a flat-plate or evacuated tubular solar collector available in the market. Besides, in order to reduce the primary energy consumption, the auxiliary energy used in solar cooling systems is recommended to be other options of renewable or clean energy (e.g. ground source heat pump, biomass, free cooling) or high-efficiency heat pumps [119]. For instance, an improvement on the *COP* up to 40% was achieved when the dry cooling tower of a solar absorption cooling system was replaced by a geothermal sink [120]. Moreover, the *COP* of the cooling system will no longer depend on the outdoor temperature if a geothermal heat rejection system is used [121]. In [117], the authors studied a solar- biomass based single effect $LiBr/H_2O$ absorption. They noted that the solar- biomass absorption cooling system could achieve thermal comfort and great greenhouse gas emissions reduction for residential buildings in tropical locations.

Solar absorption cooling has achieved great energy saving and solar cooling fraction in very large cities such as subtropical Hong Kong [122], southern European and Mediterranean areas [123], Abu Dhabi [124], Zurich, Palermo [125], Madrid [126], Albuquerque-New Mexico (see [127] [128] [129]), Pittsburgh, PA [130] and Los Angeles[118]. Moreover, significant lower total costs (capital costs, operating costs for maintenance and system operation and the energy costs) were achieved at higher cooling energy demand [124]. In addition, the solar fraction could increase greatly according to the control strategy adopted [131]. However, solar absorption cooling failed to prove its effectiveness in all cases from an economic standpoint. For instance, the current energy price in Barcelona make the use of solar absorption cooling economically unappealing, even with reducing gases emissions contributing to global warming [132]. According to [6], the solar thermal cooling costs would be competitive with solar electric cooling costs consisting of a vapor compression chiller connected to photovoltaic panels (*PV*), if the *COP* of thermal refrigeration increases to be more than 1. Consequently, although it is an environmentally friendly technology, the solar absorption technology needs to prove its competence.

As it could be seen from previous works, the efficiency of solar absorption system and the solar fraction vary widely depending on cooling load, available solar energy, and ambient temperature. Hence the interest of making a comparative study of single effect solar absorption chillers in different cities has spread out in various regions of the world. In addition, it is established (see for instance [133]) that most losses in a solar absorption cooling system resides in the solar collector and then in the generator. Therefore, the energy variations on both solar collector and generator have to be considered. The energetic and the economic benefits achieved using solar absorption cooling vary widely according to the situation and the climatic conditions of each case study. More precisely, a simplified method for the comparison of solar absorption cooling suitability in different cities is presented. The effect of different variables related to the location such as temperature, solar radiation and cooling load are investigated. An effectiveness factor ($\mathcal{EF}$) is suggested in order to compare the behavior of solar absorption chiller based on simulation results and energy study for various locations characterized by different climatic conditions. In each city, two types of $\mathcal{EF}$ were calculated : (1) monthly $\mathcal{EF}$ estimated for the typical day of each month and (2) yearly $\mathcal{EF}$ combining the data of all typical days. This index serves as an assessment tool of solar absorption cooling contribution in different locations.

The chapter is organized as follow. In section 1 and 2, a solar absorption cooling system is defined and modelled. In section 3, the adopted mathematical model of solar collector and generator is presented. The formulation and discretization of heat balance equations are developed and then solved numerically, in section 4. The model is then validated in section 5 using the experimental results of [134] and [135]. Section 6 is dedicated to the description and the use of the effectiveness factor. Finally, the main conclusions are presented in the last section.

2.2 System description

Figure 2.1 depicts the schematic of a solar cooling system. It consists of a 35 KW water fired single-effect $LiBr/H_2O$ absorption chiller equipped with a solution heat exchanger and directly connected to the solar collectors. The operating conditions of the absorption chiller are obtained from [136].

The saturation temperature of water in the $LiBr/H_2O$ mixture at constant pressure varies according to the LiBr mass fraction. Therefore, the generator (see Figure 2.1-(d)) of the system covers a temperature glide in the range of $T_1^{sat} - T_2^{sat}$ where the evaporation of water starts at T_1^{sat}, namely the saturation temperature of water at generator pressure and LiBr mass fraction in rich solution. The water vapor, resulting from the generator, is cooled down in the condenser (see Figure 2.1-(a)). The condensed water is evaporated again at low pressure in the evaporator (see Figure 2.1-(b)) thereby extracting heat from the medium to be cooled. The water vapor is then absorbed by the $LiBr/H_2O$ mixture of the absorber (see Figure 2.1-(c)). Meanwhile, in the heat exchanger ($\mathcal{HX}$) the rich solution entering the generator is preheated to T_{HX}^{out} by the poor solution leaving the generator. The heat in the condenser and the mixing heat in the absorber are removed by cooling water. On the other side, Valve 2 (see Figure 2.1) is closed and Valve 1(see Figure 2.1) is opened when the solar thermal fluid energy is not sufficient to heat the rich solution in the generator to above its inlet temperature (i.e. T_{HX}^{out}). Else, Valve 2 is opened, and the thermal fluid enters the generator. An auxiliary heater provides the remaining thermal energy required to achieve the generation process.

Different types of solar collectors can be used in a solar absorption system such as flat-plate, parabolic or vacuum collectors. For a solar-assisted absorption chiller, an evacuated-tube collector is interesting and can provide good performance at high temperature (see [126] [137][138][139][140]). Therefore, in this study an evacuated tube solar collector with transmittance-absorbance factor of 0.85 is considered. The water used in the solar collectors should be mixed with chemical additives (i.e.antifreeze) in order to ensure frost protection inside the solar collector especially for desert climate where high temperature variations from above $40°$C during the day to below $0°$C during night-time can be noted. Therefore, 60% ethylene glycol- 40% water is considered as the heat transfer fluid circulating through the solar collector-heat exchanger system. In addition, the percentages of ethylene glycol-water in the heat transfer fluid are determined in order to increase the boiling temperature and avoid vapour presence in the solar collector.

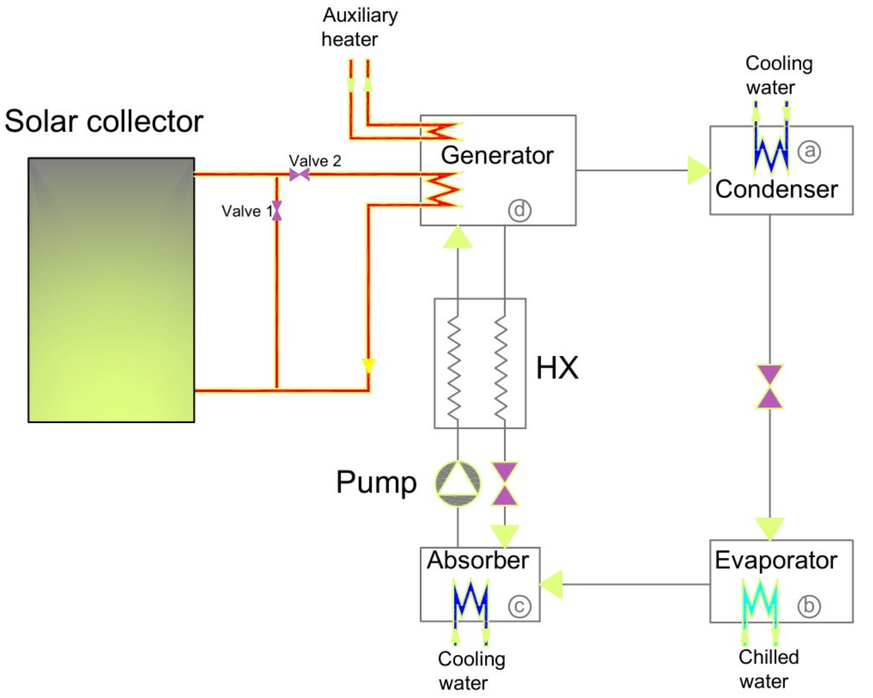


Figure 2.1 – Schematic diagram of solar absorption cooling system

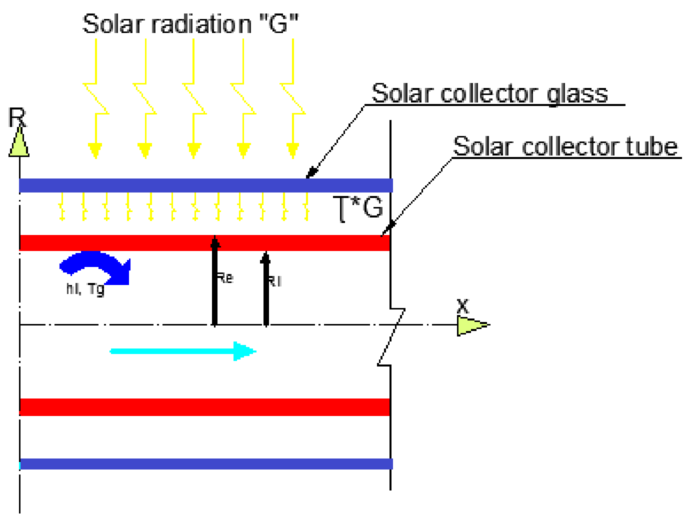


Figure 2.2 – Longitudinal sectional view of a solar collector evacuated tube.

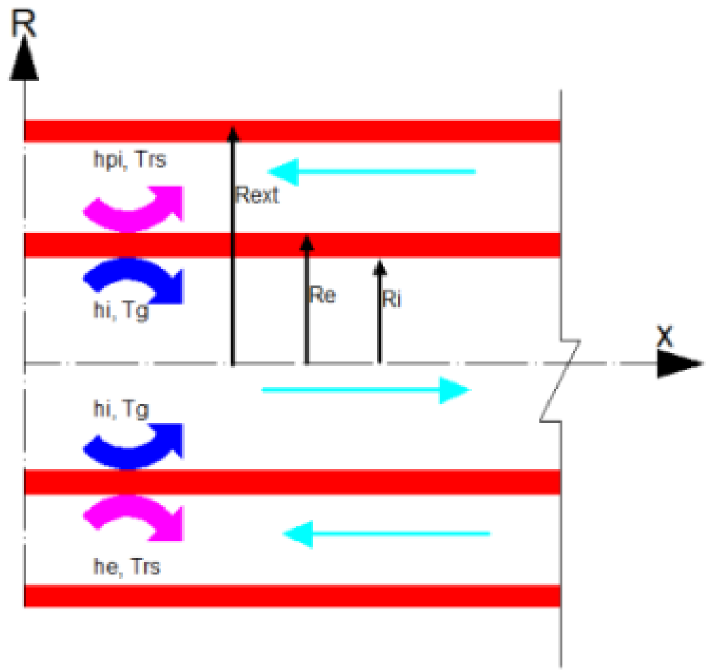


Figure 2.3 – Longitudinal sectional view of generator considered as counter flow heat exchanger.

2.3 Mathematical model

The model adopted is based on the simplified mathematical model. The solar collector and the generator sub-models are built on fundamental principles of heat transfer and heat balance equations. The dynamic behaviours of the solar collector and the generator are investigated. However, the remaining components of the system are considered to be in steady state conditions. The variables studied in the model are temperatures of fluids, EG/water mixture and rich solution, and tube wall in the solar collector and the generator. The simulation program consists of two parts : one for the solar collector with the EG/water mixture as the thermal fluid ; and the second for the generator, where the rich solution is heated by the EG/water mixture.

Solar collector and generator models as well as the numerical method are described in this section. The models are then validated using experimental results from previous works (see [134] [135]).

2.3.1 Solar collector model

An evacuated tube solar collector consisting of a serpentine copper tube is considered. The absorber, the outer surface of each tube, absorbs a part of the solar irradiation, and then transfers heat to the Ethylene-glycol/water mixture (see Figure 2.2). The performance of the solar collector is characterized by the transmittance-absorbance coefficient and the thermal losses depending on the overall heat transfer coefficient and temperature difference. The coefficient of losses used was determined from the results of tests done on an evacuated tube solar collector by [141].

— Equation for Heat balance on the tube wall

$$(\rho\, C\, S)_\omega \frac{d\, T_\omega}{dt} = (\tau\, \alpha) P_e\, G\, - P_i\, h_i (T_\omega - T_g) + (\lambda\, S)_\omega \frac{d^2\, T_\omega}{d^2\, x} - U\, P_e (T_\omega - T_a) \tag{2.1}$$

— EG/water mixture Heat balance equation

$$\rho\, C\, (T_g)\; S_g \frac{d\, T_g}{dt} + \dot{m}_g C\, (T_g)\, \frac{d\, T_g}{d\, x} = P_i\, h_i (T_\omega - T_g) \tag{2.2}$$

2.3.2 Generator model

In order to simplify the problem, the generator can be considered as a counter-flow heat exchanger where the rich solution is the fluid flowing through the outer tube (see Figure 2.3). The generator is considered to be well insulated and thus heat losses to the surroundings are neglected.

The lithium bromide mass fraction and rich solution temperature are obtained from the operating conditions of the single-effect $LiBr/H_2O$ chiller described by [136], whereas the thermophysical properties of the rich solution are calculated from equations defined by [142].

— EG/water mixture Heat balance equation

$$\rho\, C\, (T_g)\; S_g \frac{d\, T_g}{dt} + \dot{m}_g C\, (T_g)\, \frac{d\, T_g}{d\, x} = P_i\, h_i (T_\omega - T_g) \tag{2.3}$$

— Central tube wall heat balance equation

$$(\rho\, C\, S)_\omega \frac{d\, T_\omega}{dt} = (\lambda\, S)_\omega \frac{d^2\, T_\omega}{d^2\, x} - P_i\, h_i (T_\omega - T_g) - P_e\, h_e\, (T_\omega - T_{rs}) \tag{2.4}$$

— Heat balance equation for the rich solution

$$\rho\, C\, (T_{rs})\; S_{rs} \frac{d\, T_{rs}}{dt} + \dot{m}_{rs} C\, (T_{rs})\, \frac{d\, T_{rs}}{d\, x} = P_e\, h_e (T_\omega - T_{rs}) \tag{2.5}$$

2.3.3 Initial and boundary conditions

In this work, many assumptions were taken into consideration :

(A1) All losses are neglected except for the losses from solar collector wall tube to the surrounding.

(A2) The generator initially contains a mixture of $LiBr/H_2O$ (i.e. rich solution) at the temperature T_{hx}^{out}.

(A3) The initial heat transfer fluid and wall tube temperatures in the solar collector are the same as the ambient temperature.

(A4) The internal tube of the generator initially contains an EG/water mixture. Therefore, the internal wall tube and the EG/water mixture are in thermal equilibrium with the rich solution, and their initial temperatures are also set to T_{hx}^{out}.

(A5) When Valve 1 (see Figure 2.1) is opened, the EG/water mixture re-circulates in the solar collector and enters at the same temperature as that of the fluid just coming from the solar collector output. While in the other case, the EG/water mixture enters the solar collector at the same temperature as that of EG/water leaving the generator.

Based on previous assumptions, the rich solution will be cooled if the EG/water mixture enters the generator at a temperature lower than T_{hx}^{out}. Therefore, Valve 2 is held in closed position until the heat transfer fluid in the solar collector is heated to above T_{hx}^{out}.

2.3.4 Numerical method

The aim is to calculate temperature variation with time in the solar collector and the heat exchanger for one a day operation. To achieve this, equations (2.1)-(2.5) are integrated with respect to time, then solved using the finite volume method. Since the implicit method requires great computer core storage for large matrix inversion, the explicit finite volume method was used in the present work.

Initially, partial derivatives in heat balance equations are linearized using forward and central difference approximations :

$$\begin{cases} \frac{dT}{dt} \approx \frac{T_x^{t+1} - T_x^t}{\Delta t} \\ \frac{dT}{dx} \approx \frac{T_x^t - T_{x-1}^t}{\Delta x} \\ \frac{d^2 T}{d^2 x} \approx \frac{T_{x+1}^t - 2T_x^t + T_{x-1}^t}{(\Delta x)^2} \end{cases} \tag{2.6}$$

The heat exchanger and the solar collector are then each subdivided into n control volumes. Two operation modes are considered :

Mode I The EG/water mixture is only heated in the solar collector : the assistance of solar energy in the generator will not be needed in this operation mode (see equation (2.7)).

$$\begin{bmatrix} T^n_{\omega,s} \\ T^n_{g,s} \end{bmatrix}^{t+1} = \begin{bmatrix} T^n_{\omega,s} \\ T^n_{g,s} \end{bmatrix}^{t} + \Delta t \left(\begin{pmatrix} M_1 & M_2 \\ M_3 & M_4 \end{pmatrix} \begin{bmatrix} T^n_{\omega,s} \\ T^n_{g,s} \end{bmatrix}^{t} + A_1 \right) \tag{2.7}$$

where $T^n_{\omega,s}$,$T^n_{g,s}$ ϵ $\mathbb{R}^n$ and

$$M_1 = \frac{c}{Dx^2}(J_n + J_n^T) - (\frac{(a+b)}{f} + \frac{2c}{Dx^2})I_n,$$

$$M_2 = \frac{a}{f}I_n,$$

$$M_3 = dI_n \text{ and}$$

$$M_4 = \frac{e}{Dx}J_n - (d + \frac{e}{Dx})J_n^T$$

with

$$a = R_i h_i,\ b = R_e U,\ c = \frac{\lambda}{C_p \rho_p},\ d = \frac{2h_i}{R_i C_g \rho_g},\ e = \frac{\dot{m}_g}{\pi R_i^2 \rho_g} \text{ and } f = \frac{(R_e^2 - R_i^2)C_p \rho_p}{2}$$

I_n is the identity matrix of dimension n and J_n is the matrix defined as follows : for $i = 1 \ldots n-1$, the element situated at the i^{th} row and $(i+1)^{\text{th}}$ column is equal to one and zero elsewhere. ^{T} designates the transpose.

Mode II If the EG/water mixture circulates in the generator and supplies heat to the rich solution. Temperature variations in the solar collector and the generator are obtained from the following linear systems :

$$\begin{bmatrix} T^n_{\omega,s} \\ T^n_{g,s} \end{bmatrix}^{t+1} = \begin{bmatrix} T^n_{\omega,s} \\ T^n_{g,s} \end{bmatrix}^{t} + \Delta t \left(\begin{pmatrix} M_1 & M_2 \\ M_3 & M_4 \end{pmatrix} \begin{bmatrix} T^n_{\omega,s} \\ T^n_{g,s} \end{bmatrix}^{t} + A_2 \right) \tag{2.8}$$

and

$$\begin{bmatrix} T^n_{g,gen} \\ T^n_{\omega,gen} \\ T^n_{rs,gen} \end{bmatrix}^{t+1} = \begin{bmatrix} T^n_{g,gen} \\ T^n_{\omega,gen} \\ T^n_{rs,gen} \end{bmatrix}^{t} + \Delta t \left(\begin{pmatrix} N_1 & N_2 & N_3 \\ N_4 & N_5 & N_6 \\ N_7 & N_8 & N_9 \end{pmatrix} \begin{bmatrix} T^n_{g,gen} \\ T^n_{\omega,gen} \\ T^n_{rs,gen} \end{bmatrix}^{t} + A_3 \right) \tag{2.9}$$

where $T^n_{\omega,gen}$,$T^n_{g,gen}$, $T^n_{rs,gen}$ ϵ $\mathbb{R}^n$

$$N_1 = \frac{e}{dx}J_n^T - (d + \frac{e}{dx})I_n,$$

$$N_2 = dI_n,$$

$$N_4 = \frac{a}{f}I_n,$$

$$N_5 = \frac{c}{dx^2}(J_n + J_n^T) - (\frac{(a+g)}{f} + \frac{2c}{dx^2})I_n,$$

$$N_6 = \frac{g}{f}\bar{I}_n,$$

$$N_8 = \frac{2g}{k}\bar{I}_n,$$

$$N_9 = \frac{j}{k}J_n^T - (\frac{j+2\pi g}{k})I_n \text{ and}$$

$$N_3 = N_7 = 0_n$$

with

$$g = R_e h_{pi},\ j = \frac{\dot{m}_{rs}C_{rs}}{dx},\ and\ k = \pi(R_{ext}^2 - R_e^2)C_{rs}\rho_{rs}$$

where 0_n is the null matrix of dimension n and $\bar{I}_n$ is the matrix defined as follows : for $i = 1 \ldots n$, the element situated at the i^{th} row and $(n-i+1)^{\text{th}}$ column is equal to one and zero elsewhere.

The solution stability in an explicit finite volume scheme requires that the coefficients of present temperatures must be nonnegative. Therefore, a quite small time step size is adopted to minimize the possibility of error augmentation, to ensure that all the elements of the matrices multiplied by the elements of temperature vectors are positive, and thus lead to convergence.

2.4 Model validation

The developed model is validated, for $n = 100$, using the experimental data proposed in [134] and [135]. More precisely, the authors tested a solar absorption cooling system installed in the city of Madrid, Spain. It should be noted that the validations of the solar collector and the heat exchanger models were done separately.

2.4.1 Solar collector model validation

In [135], the authors tested a single-effect $LiBr/H_2O$ absorption chiller connected to vacuum flat-plate solar collectors and a hot storage tank. They registered the collector inlet and outlet temperatures during two clear days with maximum solar global radiation on the tilted surface of about $1000W/m^2$. They used a 30% EG/water mixture as a thermal fluid circulating through the solar collector. The collector length is calculated from the area of solar collector. The solar collector diameters, EG/water mass flow rate and the input data (i.e. solar global radiation, outdoor dry bulb and experimental collector inlet/outlet temperatures) were obtained from [135].

The developed model shown in section 3 was employed using the experimental EG/water inlet temperature to the solar collector. Simulation and experimental results obtained for collector outlet temperature were compared (Figure 2.4 and 2.5). Simulation results show quite good agreement with those obtained experimentally where the two graphs follow the same trajectory. The validation results show that the maximum absolute percentage error is less than 15% in both cases occurring only at two instants during the day, while the mean absolute error is about 5 and 3% for the first and the second case respectively. Hence, the above suggested model could be employed for further studies concerning the solar collector performance simulation. The parameter values used for validation are given in Table 2.1.

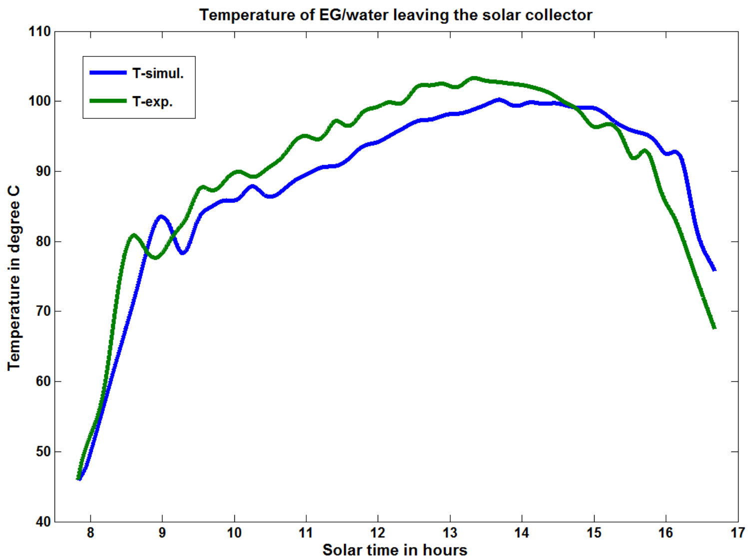


Figure 2.4 – Experimental and simulation results for solar collector output-25 August 2008

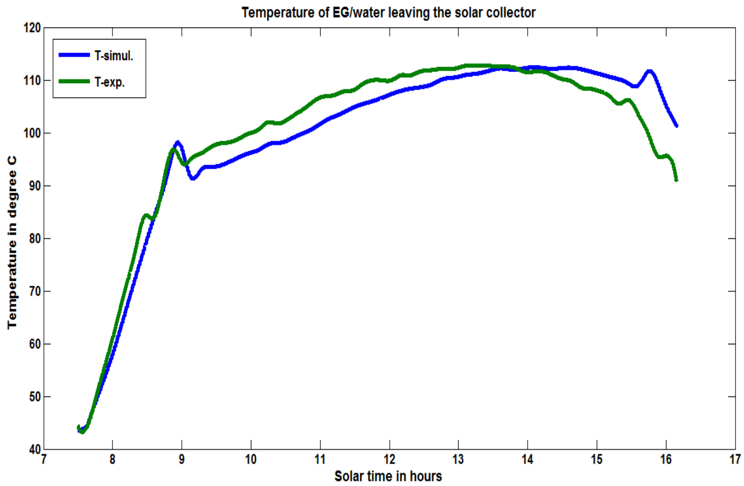


Figure 2.5 – Experimental and simulation results for solar collector output-28 August 2008

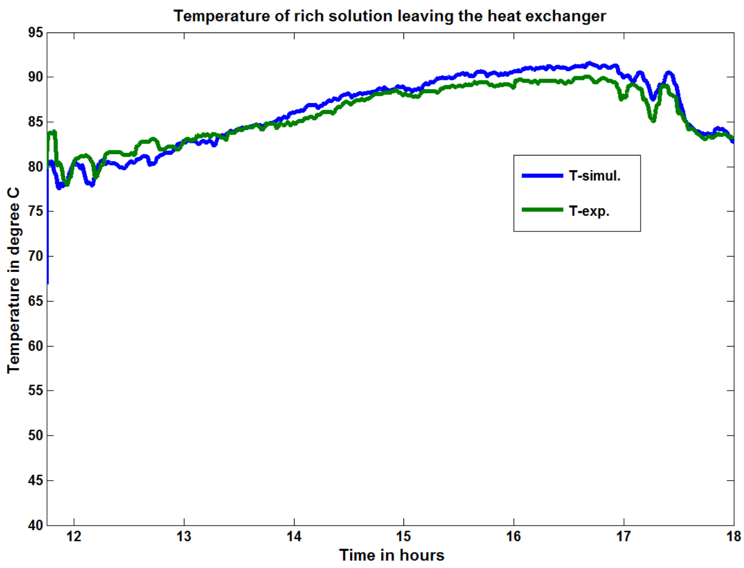


Figure 2.6 – Experimental and simulation results for the rich solution-5 August 2010

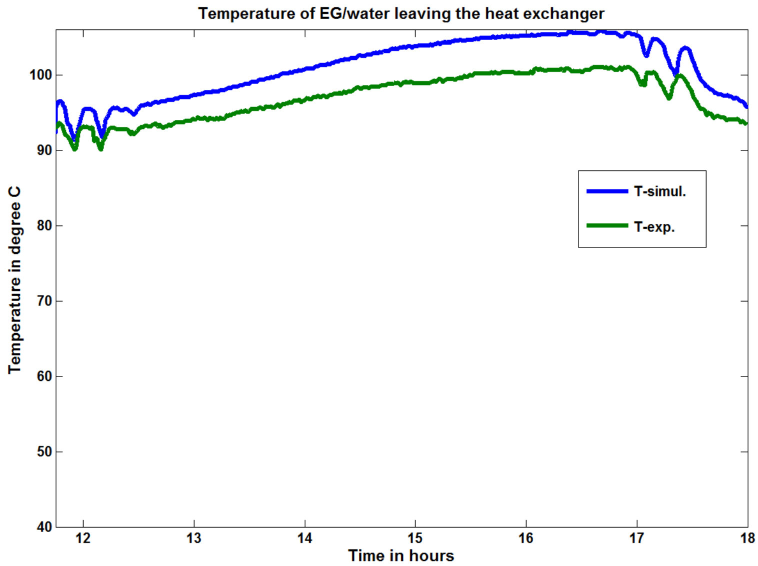


Figure 2.7 – Experimental and simulation results for EG/water mixture-5 August 2010.

2.4.2 Generator model validation

In [134], the authors tested the generator of a solar absorption cooling system. It consisted of evacuated flat-plate solar collectors, a stratified hot storage tank and a $4.5/7\ KW$ single-double effect $LiBr/H_2O$ absorption operating in a single mode. The generator of the absorption chiller was driven by a 10% EG/water of the hot water storage tank heated in the solar collector. The inlet/outlet temperatures of EG/water and rich solution flowing through the generator are extracted from graphs found in [134]. The parameter value used for validation are given in Table 2.1

Figure 2.6 and Figure 2.7 show the validation of generator model. The experimental inlet temperatures of EG/water and rich solution were used as inlet temperatures for the two fluids in the model, then the experimental and simulation outlet temperature were compared for rich solution and EG/water. The validation shows that simulation results follow closely the data registered especially for rich solution where the maximum absolute percentage error is less than 7% in both cases, while the mean absolute percentage error is 4% and 1% for EG/water and rich solution outlet temperatures respectively. Therefore, the generator of absorption machine may be treated as a counter-flow heat exchanger to predict the outlet temperature of rich solution, and then, the auxiliary energy needed can be estimated.

$Variable$	$Value$	$Unit$
Φ_i	0.016	m
Φ_e	0.02	m
Φ_{ext}	0.025	m
$\dot{m}_{g,s}$	0.47	Kg/s
$\dot{m}_{g,gen}$	1/3600	Kg/s
$\dot{m}_{rs}$	0.04	Kg/s
A_s	42.2	m^2
L_{gen}	0.45	m

Table 2.1 – Numerical values for validation

2.5 Results and discussion

The numerical model described above has been used to simulate the dynamic performance of the solar collector coupled to the generator of $35\ KW$ solar absorption chiller, using MATLAB. Here, the goal is to compare the suitability of a solar assisted absorption chiller in several cities with different geographical and meteorological conditions.

Table 2.2 presents latitude, meteorological yearly data and cooling-degree hours relative to a $22°C$ base temperature of each city using hourly Meteonorm outputs.

The numerical simulations (see Table 2.3 for parameter values) were carried out with the same mass flow rates for rich solution and EG/water mixture in all cities, since the same absorption chiller size is used. The mass flow rates, collector diameters, generator diameters and length used are the same for all cities. The collector area and length are determined in the following paragraph.

2.5.1 Collector area determination

The evaporation process, at constant pressure, that occurs in the generator of the absorption chiller is complex. During the process, the mass fraction of LiBr increases. Consequently, the saturation temperature changes from T_1^{sat} to T_2^{sat}. The saturation temperature and the mass fraction of LiBr could be determined by the temperature-concentration relationship of $LiBr/H_2O$ mixture. However, it is difficult to determine the actual mass fraction of LiBr in all finite control volumes of the previous model. Knowing that the thermophysical properties of the rich solution are not only related to the temperature but also to the LiBr mass fraction, therefore, the necessary collector area for each city is determined such that the rich solution temperature does not exceed T_1^{sat}, where the mass fraction

City	Latitude (degree)	Max T ($°C$)	Max G (W/m^2)	CDH [a] ($°C$. h)	Total H per year (KWh/m^2.year)
Canberra-Australia	-35.3	37	1190	4127.9	1733.3
Johannesburg-South Africa	-26.2	30.3	1374	3076.4	2082.4
Brasilia-Brazil	-15.9	34.1	1225	14877.8	1792.0
Jakarta-Indonesia	-6.1	35.1	1143	48537.3	1764.7
Kuala Lumpur-Malaysia	3.1	36.5	1137	48342.1	1655.1
Khartoum-Sudan	15.5	42.3	1066	72797.6	2394.0
Riyadh-Saudi Arabia	24.6	46.2	1115	59218.6	2192.5
Beirut- Lebanon	33.9	33.2	1040	16563.9	1746.0
Tunis-Tunisia	36.8	41.4	1006	16203.7	1808.5
Chicago-United States	41.8	35.4	1015	6263.8	1414.9
Paris-France	48.9	34.1	927	2185.1	1018.0
Oslo-Norway	59.9	26.5	937	267.3	902.8

Table 2.2 – Yearly meteorological data [b] and $\mathcal{CDH}$ for cities

of LiBr is still constant at this level. For this reason, the area of the solar collector is determined using simulation results in each city for the maximum global or integrated solar radiation day (non-zero $\mathcal{CDH}$). The auxiliary heater provides the additional heat required to achieve the total evaporation process. Noting that, in real applications, the solar system could produce the total heat required for desorption process. However, the method used here serves as comparative tool of collector area required in different locations.

In general, the required solar collector area for cities with high latitudes is the largest. For instance in the considered locations, the greatest solar collector area required is obtained in Oslo. The smallest solar collector area found is for Johannesburg where the extreme global radiation is close to the solar constant.

a. $\mathcal{CDH}$ on a base of T_c^{set} is calculated by the sum of $\Delta(T_h, T_c^{set}) * h$ for all hourly temperatures greater than T_c^{set} where cooling is required.

b. The hourly extremes meteorological data are obtained from Meteonorm.

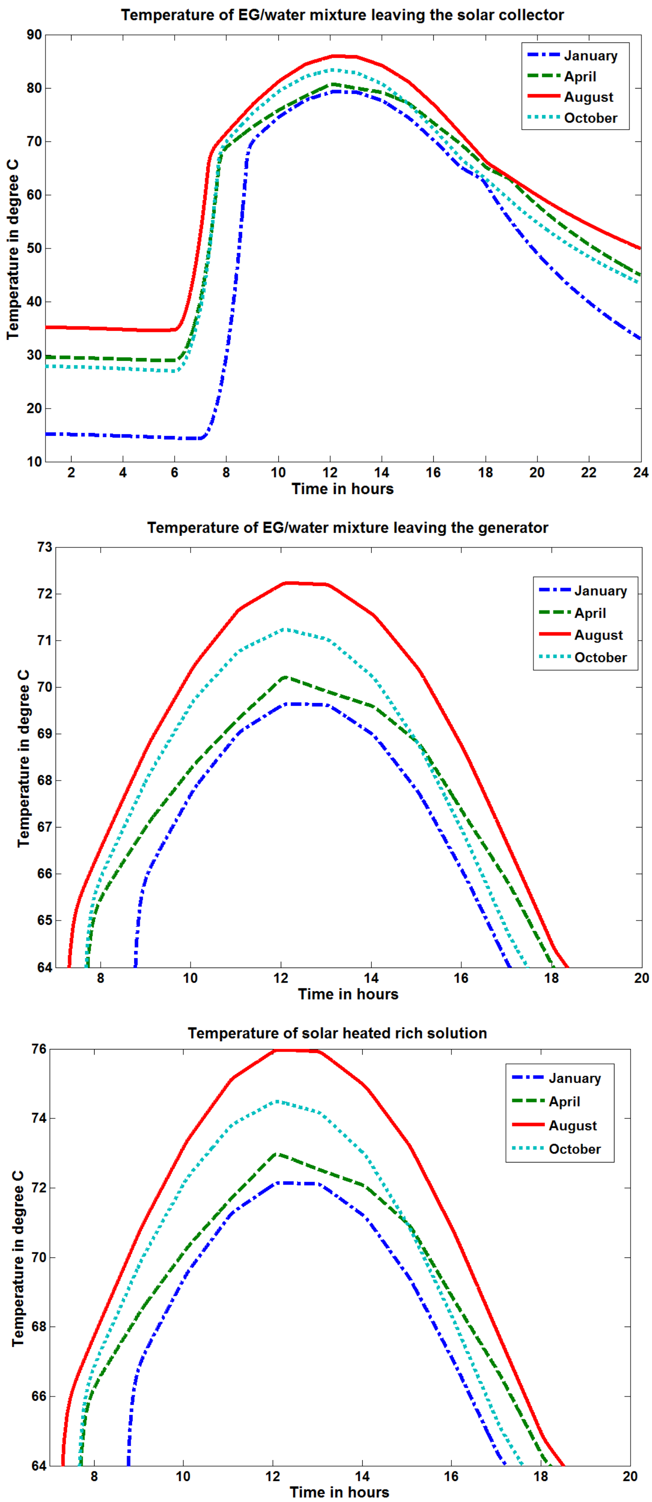


Figure 2.8 – Temperature variation of rich solution and EG/water mixture in Riyadh

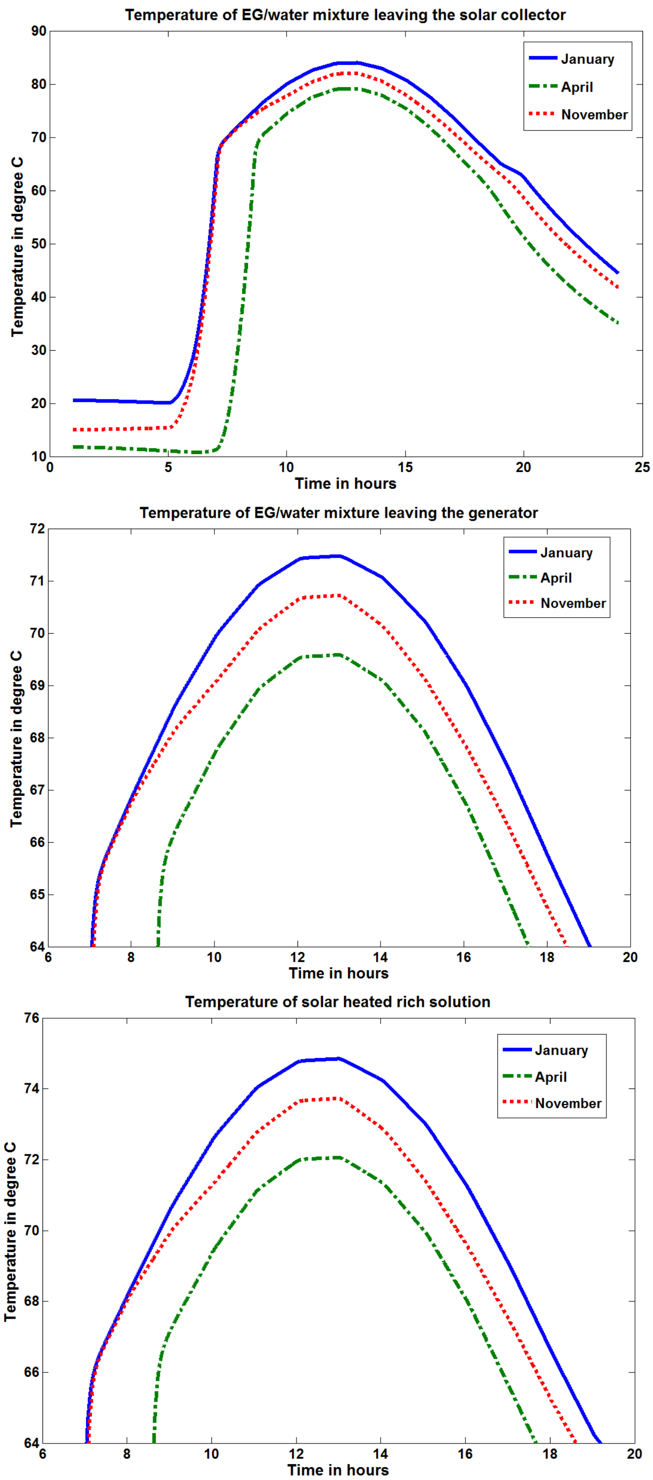


Figure 2.9 – Temperature variation of rich solution and EG/water mixture in Canberra

$Variable$	$Value$	$Unit$
T_{HX}^{out}	63.2	$^\circ C$
T_1^{sat}	76.8	$^\circ C$
T_2^{sat}	89.4	$^\circ C$
Φ_i	0.012	m
Φ_e	0.015	m
Φ_{ext}	0.025	m
$\dot{m}_g$	0.1	Kg/s
$\dot{m}_{rs}$	0.1642	Kg/s
L_{HX}	16	m
$\tau\alpha$	0.85	$-$

Table 2.3 – Numerical values for simulation

2.5.2 Simulation results

First, $\mathcal{CDH}$ was determined for all days of the year in the selected cities. Then, simulations are carried out for a typical day in each month. The typical day is defined as the one which needs the maximum cooling load during the month, and therefore the maximum $\mathcal{CDH}$ calculated. Noting that, the maximum $\mathcal{CDH}$ day fell on the maximum temperature day further away from the equator, while they do not necessarily coincide as we approach to this line. Using the solar collector area of Table 2.4 , rich solution and EG/water temperature variations along solar collector and generator were analysed during typical days.

Figures 2.8 and 2.9 present temperature variation for different typical days in Riyadh, situated in northern hemisphere, and Canberra situated in southern hemisphere). The figures show that for Riyadh, the maximum solar cooling load is covered in the typical day of August, whereas the minimum is obtained for the typical day of January. However, for Canberra the maximum energy obtained from solar radiation is in January, and no cooling load is required during August.

Earth's orbit and the tilt of axis cause cyclical variations in solar energy received and opposites seasons in southern and northern hemispheres. Thus, the great solar energy received in the northern hemisphere is during May, June and July ; whereas the southern hemisphere is exposed to high solar radiation during November, December and January.

It should be noted from results, that the difference between temperatures, denoted (ΔT), of EG/ water mixture entering and leaving the solar collector increases with solar collector output temperature. When EG/water mixture reaches higher temperature in the solar collector (e.g. Riyadh-August higher than Riyadh-October and Canberra-January), a lower temperature difference between EG/water

City	Latitude (degree)	Collector cumulative Length (m)	Area (m^2)
Canberra-Australia	-35.3	108.5	5.1
Johannesburg-South Africa	-26.2	99	4.7
Brasilia-Brazil	-15.9	105.5	5.0
Jakarta-Indonesia	-6.1	113.5	5.3
Kuala Lumpur-Malaysia	3.1	114	5.4
Khartoum-Sudan	15.5	120	5.7
Riyadh-Saudi Arabia	24.6	115	5.4
Beirut-Lebanon	33.9	126.5	6.0
Tunis-Tunisia	36.8	128.5	6.1
Chicago- United States	41.8	128.5	6.1
Paris-France	48.9	140.5	6.6
Oslo-Norway	59.9	165	7.8

Table 2.4 – Solar collector area for all cities

mixture entering and leaving the solar collector is obtained. These results are expected, because the heat transfer becomes more favorable when the temperature difference between hot and cold fluid increases. Since the inlet temperature of rich solution to the generator is fixed to T_{hx}^{out}, the EG/water mixture may cede more heat when its temperature increases.

2.6 Effectiveness Factor

Several parameters have to be considered when studying a solar-assisted absorption chiller such that the first cost, the functional cost, the solar availability and the required cooling demand. The majority of studies based on solar absorption cooling systems used one or more parameters (namely, *COP*, solar fraction, cooling capacity, required collector area, etc.). These parameters were analyzed separately for the comparison of solar absorption cooling machine working under different conditions or installed in different locations. However, in our knowledge, none of the previous works have examined the combined effect of all parameters cited above.

The aim of this work is to study the influence of city location on the behavior of solar absorption system. Therefore, the effect of factors closely related to the location ; the solar radiation and the ambient temperature are investigated. In a solar driven absorption chiller, the solar radiation affects the collector area required and the fraction of free heat provided ; while, the cooling load and the temperature in the solar collector are highly influenced by the ambient temperature. Consequently, an $\mathcal{EF}$ that combines the effect of these factors is defined in this section. The variables used for the definition of $\mathcal{EF}$ are : (1) the solar collector area related to the first cost, (2) the free energy provided illustrating the functional cost and (3) the cooling degree hours ($\mathcal{CDH}$) that represents the cooling demand in each city. The maximum values of these variables are taken as a reference for all cities.

Among all investigated locations, the minimum and the maximum collector area are obtained in Johannesburg and Oslo, respectively. The solar collector required for absorption chiller installed in Oslo is the more expensive. Consequently, it is considered to be the reference since the initial cost of solar collector in all cities is expected to be lower than that used in Oslo.

Concerning the functional cost, the best case is assumed to be where the solar system can provide the total of the energy required. Therefore, the renewable energy assured by the solar system during typical days were calculated from the temperature rise of the solar heated rich solution using the following expression :

$$E_{\text{solar}} = \dot{m}_{rs}\, C_{rs}\, \Delta t \sum_{i=1} \left(T^{i}_{rs,100} - T^{out}_{hx}\right) \tag{2.10}$$

The total energy required in the generator E_{Total} is the sum of the provided solar energy and the auxiliary heat :

$$E_{\text{Total}} = E_{\text{Solar}} + E_{\text{Auxiliary}} \tag{2.11}$$

In (2.11), the energy E_{Total} refers to the heat required to raise the temperature of rich solution to T_1^{sat} , and $E_{\text{Auxiliary}}$ the energy required to achieve the total evaporation process. The results showed that the maximum fraction of the assured renewable energy is reached in Oslo during a typical day of June. This is probably attributed to the important day length (around 19 hours) with a quite high day integrated solar radiation. The minimum solar fraction was found in Paris during a typical day of August with low integrated solar radiation. It was noted that the fraction depends on hourly global radiation and the day length from sunrise to sunset.

The cooling demand at a given city is considered to be directly proportional to the $\mathcal{CDH}$. It refers to an outside base temperature below which no cooling is needed. In general, $\mathcal{CDH}$ reaches higher values for cities located near equator. Moreover, $\mathcal{CDH}$ in these cities remains high all over the year.

2.6.1 Monthly effectiveness factor

In this section, the expression of the monthly $\mathcal{EF}$ - used for the comparison of monthly solar cooling contribution- is expressed by the following equation :

$$\mathcal{EF}_{\text{month}} = \frac{A_{\max}}{A}\left(\frac{E_{\text{solar}}}{E_{\text{Total}}}\right)_{\text{Typical-day,month}}\left(\frac{\mathcal{CDH}}{\mathcal{CDH}_{\max}}\right)_{\text{month}} \tag{2.12}$$

The first term $\frac{A}{A_{\max}}$ is related to the initial cost, which is the same for all months in a given city. Where A is the solar collector area needed in a city and $A_{\max}$ is the maximum solar collector area found among all cities, and thus represents the maximum initial cost. Since the great initial costs trivialize the use of solar absorption system in a city, the monthly $\mathcal{EF}$ is inversely related to this term.

The second term $\frac{E_{\text{solar}}}{E_{\text{Total}}}$ of the monthly $\mathcal{EF}$ represents the energy fraction assured by the solar system. Thus, increasing this ratio will make more favorable to employ the solar absorption cooling system in a city.

The third term $\left(\frac{\mathcal{CDH}}{\mathcal{CDH}_{\max}}\right)_{\text{month}}$ estimates the proportion of cooling load needed to the maximum found for all cities during a month, where $\mathcal{CDH}$ is the cooling-degree hours needed for a month in a city and $\mathcal{CDH}_{\max}$ is the maximum found in all cities for a given month. Note that, when $\mathcal{CDH}$ increases, the cooling requirements become more extensive and energy consumption rises. Thus, it should benefit more from renewable energy available in a location. Therefore, $\mathcal{EF}$ is directly related to this term.

Generally, monthly $\mathcal{EF}$ - for a given month - rises when we head from South to Riyadh expect for some cases. Then, this factor decreases when we move to the North. This path comprises many exceptions due to nearness to cities of water, geography, topographical features, city environment and other factors that influence climate and cooling need.

The monthly $\mathcal{EF}$ of cities is shown in Figures 2.10 and 2.11

Figures 2.10 and 2.11 show that all cities situated above Riyadh have their maximum monthly $\mathcal{EF}$ in July. For Riyadh, the maximum is reached in August where the $\mathcal{CDH}_{\max}$ is attained, and the monthly $\mathcal{EF}$ of July in Riyadh is close to that obtained in August. For Kuala Lumpur and Khartoum the maximum is reached in May. The obtained graphs show that in the southern hemisphere of the earth, there is no common maximum $\mathcal{EF}$ month. The month of maximum $\mathcal{EF}$ varies from one to another city. For Canberra and Jakarta, January is the maximum $\mathcal{EF}$ month where the maximum solar fraction for these two cities is provided with a sufficiently high $\mathcal{CDH}$. In Johannesburg, the maximum solar fraction is achieved in January too, but the $\mathcal{CDH}$ would not raise the $\mathcal{EF}$ enough. In addition, we remarked that away from tropics ($23°26'S$ and $23°26'N$) the number of zero monthly $\mathcal{EF}$ increases with latitude, probably because the solar energy received over the year changed significantly which produces great seasonal differences.

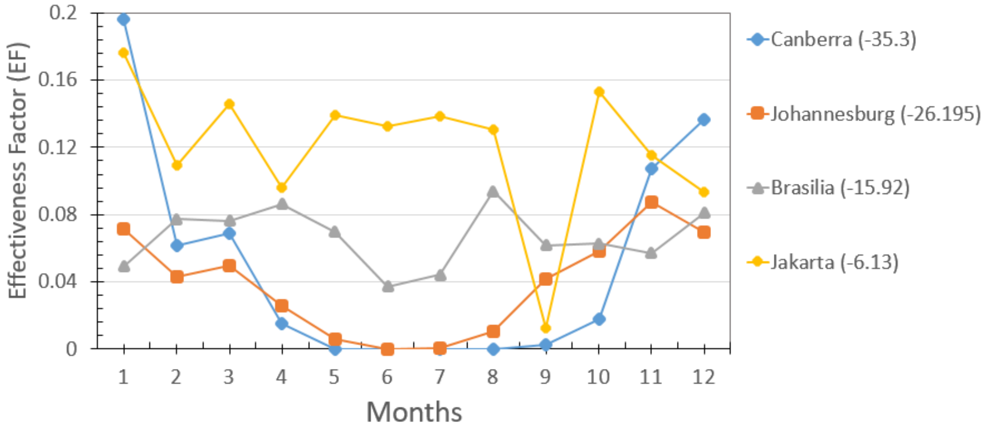


Figure 2.10 – Monthly effectiveness of cities situated in the southern hemisphere

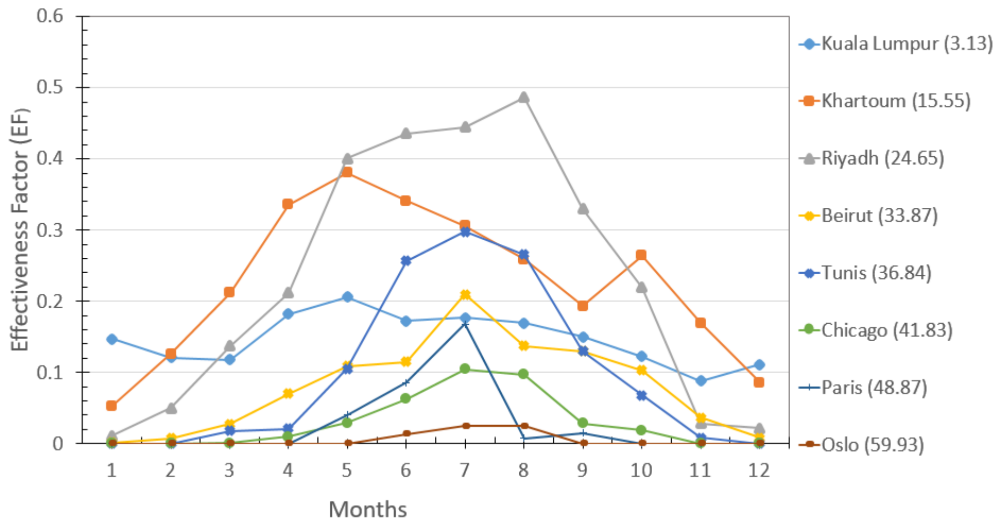


Figure 2.11 – Monthly effectiveness of cities situated in northern hemisphere

2.6.2 Yearly Effectiveness factor

In this part, a yearly $\mathcal{EF}$ is defined in order to compare the behavior of the solar absorption system for a whole year by using the following expression :

$$\mathcal{CF}_{\text{yearly}} = \frac{A_{\text{max}}}{A} \frac{\sum_{i=1}^{12} \left((E_{\text{solar}})_{\text{Typical-day}} \mathcal{CDH} \right)_i}{E_{\text{Total}} \mathcal{CDH}_{\text{max}}} \tag{2.13}$$

The solar energy is considered only for the typical day in each month. In (2.13), the variable i represents the month of the year (e.g. 1 for January, 2 for February, etc.).

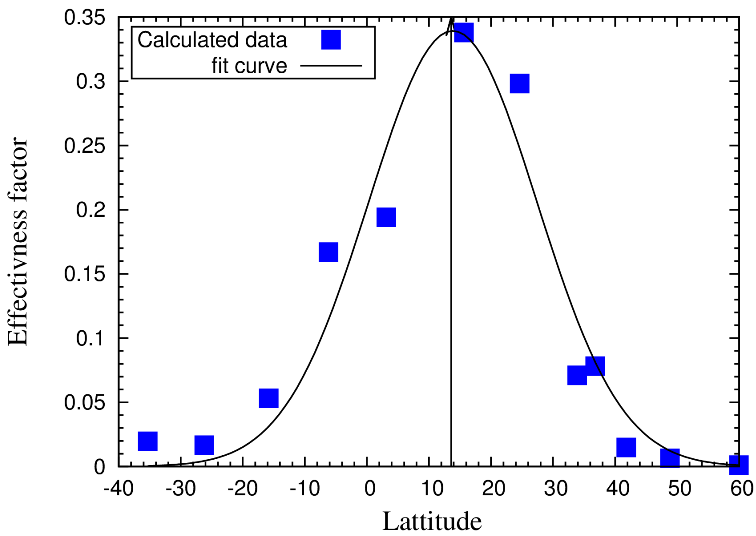


Figure 2.12 – Yearly EF in function of latitude.

Figure 2.12 gives the yearly $\mathcal{EF}$ plotted in function of latitude. This graph shows that the variation of yearly $\mathcal{EF}$ in function of latitude follows an exponential trajectory (guassian distribution) centered near Khartoum of latitude $15.5°N$. The equation of the fit curve is defined in 2.14. This result agrees with the solar distribution over the world found in [40] where it is noted that the most favorable belt of solar energy is situated between $15°N$ and $35°N$ with over than 3000 sunshine hours per year. The belt situated between equator and $15°N$ is less favorable. For southern cities, even though the belt situated between $15°S$ and $35°S$ is favorable for solar energy, the yearly $\mathcal{CDH}$ calculated are very low as shown in Table 2.2. In addition, the obtained yearly $\mathcal{EF}$ confirms that solar absorption cooling is recommended for high cooling demand applications [124] and for regions with high solar radiation [140]. Indeed, the region where yearly

$\mathcal{EF}$ reaches maximum values has the higher $\mathcal{CDH}$ and the higher yearly integrated solar radiation.

$$f(x) = \frac{1}{c} exp(-0.5(\frac{x-a}{b})^2) \tag{2.14}$$

where $a = 13.8$, $b = 13.6$ and $c = 3$

In Figure 2.12, the yearly $\mathcal{EF}$ of Canberra (-35.3) is higher than that for Johannesburg (-26.2) even though the yearly solar radiation is greater in the second. Indeed, Johannesburg is a high elevation city, and therefore temperatures are generally mild throughout the year. This is probably the cause for higher yearly $\mathcal{EF}$ in Canberra.

Regards to the occurrence of local minimum in the curve profile on Beirut latitude, the high temperature and solar radiation during June, July and August in Tunis (see Figure 2.11) make its $\mathcal{EF}$ more important than that of Beirut.

In conclusion, the use of solar absorption system is more feasible for cities situated between $10°N$ and $25°N$, then the suitability of the system decreases from both sides out of this latitude interval. Finally the yearly $\mathcal{EF}$ is an objective index that compares the behavior of a solar absorption system in function of latitude, but there are other factors that play significant roles in determining the climate such as topographical features, seasonal swings of weather, and distance from water source (seas, rivers, etc.). Therefore, the behavior of solar absorption chiller in different locations is nearly predicted using the factor defined. Thus, the choice of absorption cooling under certain climatic conditions could be led properly based on the simplified method presented in this work.

2.7 Conclusion

In this chapter, the influence of the location on the behavior of solar absorption chiller is studied. Therefore, the influence of factors related to the location on solar absorption system are investigated. The behavior of solar absorption chiller in different cities situated on different latitude lines is studied. Then, an effectiveness factor that combines the effect of different parameters as solar radiation and ambient temperature is defined. Also, the initial cost of solar collector, the free energy provided by solar energy and the $\mathcal{CDH}$, representing the cooling load, are considered in the definition of this factor.

In general, $\mathcal{EF}$ calculated for a given month increases when we move from latitudes that are to the south of Riyadh, then it decreases when moving to latitudes to the north of Riyadh. Then, a yearly $\mathcal{EF}$ was defined for the comparison of yearly operation in different locations. The yearly $\mathcal{EF}$ showed that there is a relation between latitude and effectiveness of the solar absorption system. The system studied has recorded high yearly $\mathcal{EF}$ in cities located between 10^oN and 25^oN. It is noted that a high year-integrated solar radiation is not necessarily accompanied by a high cooling load, especially in the earth's southern hemisphere, and thus does not imperatively lead to a high $\mathcal{EF}$.

The defined effectiveness factor in this chapter can serve as an objective index to predict the influence of different parameters related mainly to the location, except cost, on solar absorption chiller. This work may be a basis for the study of similar effectiveness factors on other thermal systems working under same conditions. For instance, the energy price which varies from one to other location is not included in this work. However, it has a great influence when a thermal system is analyzed from an economic standpoint. Therefore, more improvement needs to be made on $\mathcal{EF}$ so it combines the effect of all parameters related to the machine operation and cost. Moreover, this factor should be enhanced in such way that it serves in the comparison of two systems, and consequently, to determine the best system for a given location in the world.

In the next chapter, a detailed study of hybrid cooling systems and the benefits attained by combining different cooling technologies is presented. Based on this review, a selection scheme to find the best hybrid cooling system for minimum energy consumption and pollution emissions according to each climatic zone is suggested.

3.Hybrid cooling systems : a review and an optimized selection scheme

Nomenclature

Abbreviations

ABSC	*Absorption Cooling System*
ADSC	*Adsorption Cooling System*
ASC	*The evaporator of ABSC is considered as a sub-cooler in VCC*
CE	*Condensing ejector*
CEV	*Cascade ejector-vapor compression cooling system*
CPVT	*Concentrating Photovoltaic Thermal*
DCHE	*Desiccant Coated Heat Exchanger*
DEC	*Direct Evaporative Cooler*
DSC	*Desiccant System*
EC	*Evaporative Cooler*
EEA	*Cooling machine with an ejector located at the absorber inlet replacing the solution expansion valve in ABSC*
EEV	*Ejector as an expansion valve in a VCC*
EJC	*Ejector Cooling System*
ERC	*Cooling machine with an ejector between the rectifier and the condenser of ABSC*
GAX	*Generator Absorber Heat Exchanger*
HR	*The heat of condensation of VCC is rejected into the ABSC*
HSTC	*High Stage Thermal Compressor*
IC	*The evaporator of ABSC is considered as an intercooler in a two stage VCC*
IEC	*Indirect Evaporative Cooler*
LD	*Liquid Desiccant System*
LiBr	*Lithium Bromide*
LSTC	*Low Stage Thermal Compressor*
NH3	*Ammonia*
OTSDC	*One-Rotor Two-Stage Rotary Desiccant Cooling System*
PAG	*Parallel configuration of absorption cooling system with mechanical vapor compression between absorber and generator*
PEV	*Parallel ejector-vapor compression cooling system*
PPHE	*Polymer Plate Heat Exchanger*
PV	*Photovoltaic*
RA	*Vapor recompression-absorption cooling system*

RC	*Resorption-Compression machine*
REC	*Regenerative Evaporative Cooler*
SA	*Surface Available*
SD	*Solid Desiccant System*
SE	*Solar Energy*
SH	*Series configuration of absorption cooling system with integrated compressor at the high pressure side*
SIEC	*Semi-Indirect Evaporative cooler*
SL	*Series configuration of absorption cooling system with integrated compressor at the low pressure side*
TSDC	*Two-Stage Desiccant Cooling System*
VCC	*Vapor Compression Cooling System*

Variables

A	*Area* $[m^2]$
COP	*Coefficient of performance* $[-]$
c	*Compression ratio* $[-]$
D	*Moisture removal factor* $[g/kg]$
e	*Thickness* $[mm]$
E	*Energy* $[KJ]$
ES	*Energy saving* $[\%]$
f	*Frequency* $[Hz]$
I	*Solar radiation* $[W/m^2]$
L	*Length* $[m]$
$\dot{m}$	*Mass flow rate* $[kg/s]$
N	*Rotation speed* $[rpm,\ rps\ or\ rph]$
P	*Pressure* $[bar or Pa]$
PS	*Power saving* $[\%]$
Q	*Capacity* $[W\ or\ KW]$
RH	*Relative humidity* $[\%]$
RHXA	*Relative heat exchanger area* $[\%]$
SF	*Solar fraction* $[\%]$
T	*Temperature* $[^oC\ or\ K]$
t	*Time* $[s\ or\ min]$
V	*Volume* $[m^3\ or\ L]$
$\dot{W}$	*Power* $[W or KW]$
ϵ	*Exergy efficiency* $[\%]$
ϕ	*Diameter* $[mm]$
η	*Efficiency* $[\% or -]$
ν	*Entrainment ratio of the ejector* $[-]$
ω	*Air humidity ratio* $[g/kg]$

Subscripts

a	*Ambient*
ab	*Absorber*
ad	*Adsorber*
av	*Average*
c	*Cooling*
ca	*Cooling air*
cd	*Condenser*
cl	*Solar collector*
cp	*Compressor*
cr	*Critical*
cs	*Consumption*
d	*Desorption*
dc	*Discharge*
ds	*Desiccant solution*
el	*Electric*
ev	*Evaporator*
exp	*experiment*
gc	*Gas cooler*
gn	*Generator*
go	*Geothermal*
h	*Heating*
hs	*Heat source*
hw	*Hot water*
i	*Inlet*
l	*Latent*
max	*Maximum*
min	*Minimum*
o	*Outlet*
op	*Optimal*
rg	*Regenerator / Regeneration*
sc	*Sub-cooling / sub-cooler*
sim	*Simulation*
sp	*Supply*
th	*Thermal*
to	*Total*
w	*Water*
ws	*Weak solution*
wt	*Waste*

Symbols and abbreviations used in the table

Type of study

E	*Experiment*
S	*Simulation*

Obtained results

$\uparrow$	*Increasing or Increase*
$\downarrow$	*Decreasing or Decrease*
$\rightarrow$	*Implies*
$>$	*Greater*

3.1 Introduction

Hybrid cooling system was found energy efficient, and energy saving. It combines the advantages of different cooling processes, improves the coefficient of performance (*COP*) and reduces the energy consumption of cooling system. The authors in [61] noted that the development of hybrid systems is an efficient method that makes adsorption technology cost competitive compared with other cooling systems. The authors in [143] showed that the highest *COP* was reached in a hybrid adsorption-vapor compression cooling system where the adsorption bed used activated carbon/CO_2. Mohammad et al. [144] pointed out the importance of sizing and operating temperatures of vapor compression chiller in hybrid liquid desiccant combined with vapor compression machine. Compared to standing alone evaporative coolers, the hybrid liquid desiccant systems combined with direct or indirect evaporative achieved a *COP* increase greater than 20% [145].

The hybrid cooling system is widely investigated in the literature. However, to our knowledge, there are no review that exposed the different types of hybrid system, except that are concerned about one type ([123] [146] [145]). The aim of this chapter is to present a detailed study of hybrid cooling systems and the benefits attained by combining different cooling technologies. Therefore, the hybrid cooling systems are classified into five main categories according to the combination of cooling processes or machines : vapor compression base cooling, absorption base cooling, adsorption base cooling, desiccant-evaporative and multi-evaporator cooling system. Based on this review, a selection scheme to find the best hybrid cooling system for minimum energy consumption and pollution emissions according to each climatic zone is suggested. As a result, a hybrid cooling system could be chosen properly according to the meteorological data of the studied climatic zone. The data required are for temperature and relative humidity to estimate the sensible and the latent load, respectively. In addition, the availability of solar energy must be studied in order to investigate the possibility of using thermal driven cooling machine.

3.2 Hybrid systems based on a vapor compression cooling machine

The high power consumption is the major operating disadvantage of a vapor compression chiller(*VCC*). Cycle modification by incorporating thermal driven components or combining different cooling machines could be an effective method to reduce the electrical energy consumed and consequently mitigate the problems related to environment pollution. Moreover, a part of energy required could be provided by a renewable energy source especially if the *VCC* is combined with a thermally driven cooling process.

3.2.1 Cascaded absorption- vapor compression cooling systems

The absorption-compression based *VCC* cooling cycle has usually a cascaded configuration. There are three configurations of cascaded absorption-vapor compression cooling, as shown in fig. 3.1 (a) the evaporator of the absorption cycle (*ABSC*) is used to subcool the liquid leaving the condenser of *VCC* (*ASC*) ; (b)the evaporator of *ABSC* absorbs the heat rejected from the condenser of *VCC* (*HR*), and (c) the evaporator of *ABSC* is considered as an intercooler installed between the compressors of a two stage *VCC* (*IC*). The cascaded absorption-vapor compression cooling system combines the advantages of *VCC* and *ABSC*. Moreover, it has the benefit of primary energy consumption reduction, compressor size reduction, *COP* improvement and the possibility of using free renewable energy (namely waste, solar, biomass, geothermal, etc).

Garimella et al. [147] analyzed the performance of a cascaded absorption-vapor compression cooling system where the absorption cycle is driven by waste heat. They found that the system exhibits high *COP* and energy consumption reduction over a wide range of temperature. They also compared the cascaded cycle to a two-stage *VCC*. Deng et al. [148] studied a solar hybrid cascaded absorption-vapor compression cooling system. When the inlet temperature of the generator is less than 80^oC, the cooling system works as a conventional *VCC*. Kairouani and Nehdi [149] studied the possibility of using geothermal energy to drive an absorption cycle cascaded with a conventional *VCC*. Hwang [150] analyzed the performance of a waste heat driven absorption-vapor compression refrigeration system integrated with a micro-turbine. The cooling produced by *ABSC* is used either to sub-cool the liquid leaving the condenser of *VCC* or to pre-cool the condenser and the micro-turbine air. The cascaded system with pre-cooling of the intake air of the micro-turbine showed the best results in energy saving. Seyfouri and Ameri [151] studied a combined heat and power system. The *VCC* working at low stage is driven by the micro-turbine whereas the resulting waste heat is used to drive the generator of *ABSC* working at high temperature stage. They presented different configurations of cascaded absorption- vapor compression cooling systems. They found that the configurations with two compressors are the highest energy saving ones. The summary of the studies conducted on cascaded absorption- vapor compression cooling system are shown in table 3.1 .

Table 3.1 – Summary of studies conducted on cascaded absorption- vapor compression cooling system

		Obtained results			
Work(type of study)	**System description**	**COP and/or Efficiency (η)**	**Energy and/or Power**	**Operating conditions**	**Comparison with conventional VCC**
[147] (S)	-Type of machine :HR -Waste heat driven machine -Single-effect $LiBr/H_2O$ ABSC -Subcritical CO_2 VCC	-COP$_{to}$ $\approx$ 0.6 -COP$_{el}$ $\approx$ 5.7 -waste heat considered free $\rightarrow$ (COP$>$ 8)	-$\dot{W}_{wt}$= 200MW and $\dot{W}_{cp}$=23MW -(T$_{ev}$=5°C) $\rightarrow$ Q$_c$=82MW -(T$_{ev}$=-40 °C) $\rightarrow$ (Q$_c$=51MW)		(PS=31%) compared to two-stage VCC
[148] (E, S)	-Type of machine : ASC -Solar driven system -Single-effect $LiBr/H_2O$ ABSC -Trans-critical CO_2 VCC	(ΔT$_{SC}$=7.7°C) , (T$_{ev}$=7°C) and (T$_{gc,o}$=38°C) $\rightarrow$ (COP$_{vcc}$ =3.8)	(>50%) of Q$_c$ supplied by solar energy.		(ES=19.3%/year) compared to conventional trans-critical VCC
[149] (S)	-Type of machine : HR -Geothermal driven system -Single-effect H_2O/NH_3 ABSC -Subcritical VCC	-COP $\in$ (5.4–6.2) -COP$_{ABSC}$ = 0.8		T$_{go}$ $\in$ (343-349K) provide hot water for ABSC with (T$_{gn}$=335 K)	-(COP $\uparrow$) $\in$ (37-54%) -(E$_{cs}$ $\downarrow$)
[150] (S)	-Type of machine : ASC -Waste heat driven system -Combined heat and power system.	(Q$_{ABSC}$ $\uparrow$) $\rightarrow$ (η $\uparrow$)			-(ES=12%) if conventional cascaded system is used -(ES=3%) if the air entering the condenser is pre-cooled

[151] (S)	-Type of machine : ASC, HR, IC -Configurations — (1) : HR with an evaporator-condenser heat exchanger ; — (2) : SC — (3) : IC — (4) : IC and SC			-(T_a ↑) → (T_{gn} ↑) and (T_{ev} ↑) -(T_a < 30°C) and (T_{gn} ↓) → avoid crystallization	-(> ES) in (4) followed by (3) -(> (COP ↑) in (1) followed by (4) -(4) is the Best configuration

3.2.2 Adsorption-vapor compression cooling system

In addition to reducing the mechanical compression energy needs by using thermal compression, the hybridization of adsorption bed with vapor compression machine makes the operation with a heat source close to ambient temperature possible [153], avoids the use of heat exchanger and pumps [152], reduces the mechanical compressor work and therefore improves the *COP*. However, the disadvantages of an adsorption system (such as frequent adsorption bed replacement in case of silica gel, the difficulty in dealing with extremely fine micro-porous particles in activated carbon, etc.) are not eliminated by this sort of hybridization. There are two adsorption-compression cooling system configurations, as shown in fig. 3.2 [152] : (a)the thermal compressor (adsorption bed) -located between the mechanical compressor and the evaporator- provides the low stage compression (*LTSC*) and (b)the adsorption bed -located between the mechanical compressor and the condenser- provides the high compression stage(*HTSC*). In configuration (a), the amount of irreversible heating between the evaporator and the compressor will be larger and an intercooler between the thermal and mechanical compressors is required.

Banker et al. [152] investigated the efficacy of integrating a thermal compressor (adsorption) in a *VCC* . They compared the performance of hybrid systems with one-stage (1-stg) and two-stage (2-stg) thermal compression cooling systems. They found that the hybrid system is feasible even when a low grade heat source used. They also noted that the *COP* in low stage adsorption is marginally lower than that of a system with adsorption at high stage. According to Banker et al. [152], the system will deliver tangible benefits when cooling loads are in the range of a few kilowatts. Jribi et al. [154] studied the performance of a hybrid adsorption-compression cooling system. The thermal compressor providing the high stage is driven by waste heat. The results of the studies conducted on Adsorption-vapor compression cooling system are shown in Table 3.2.

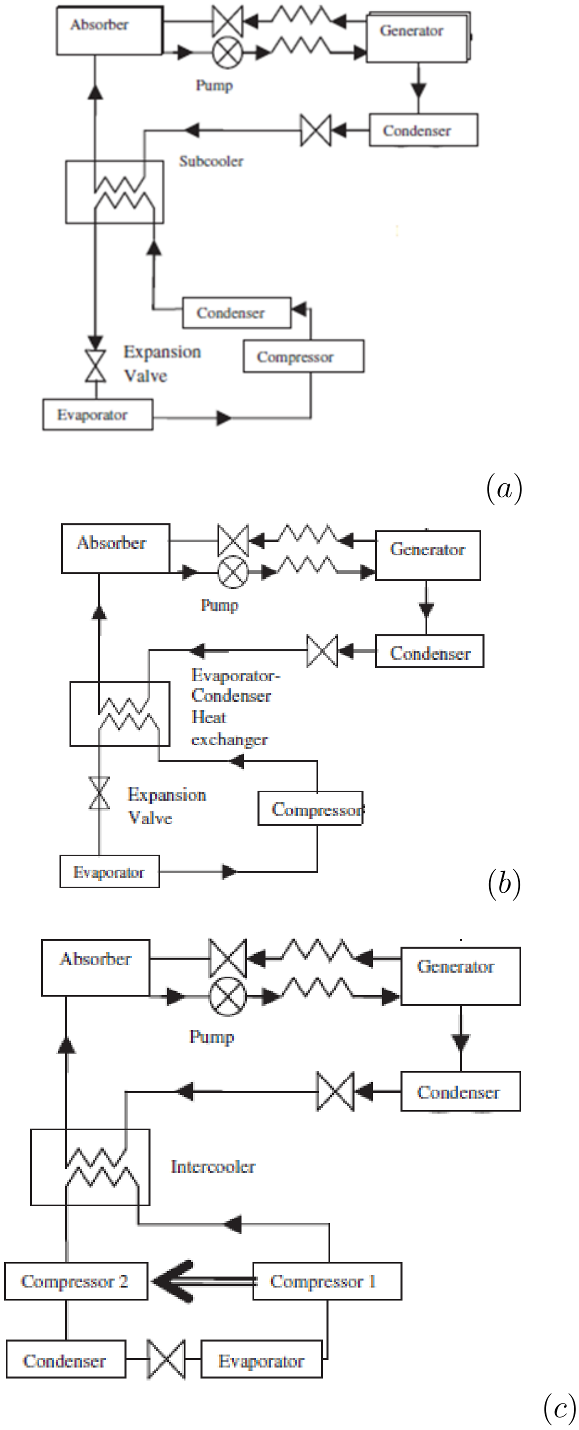


Figure 3.1 – Cascaded absorption-vapor compression cooling system : (a) *ASC* ; (b) *HR* ; (c) *IC* [152]

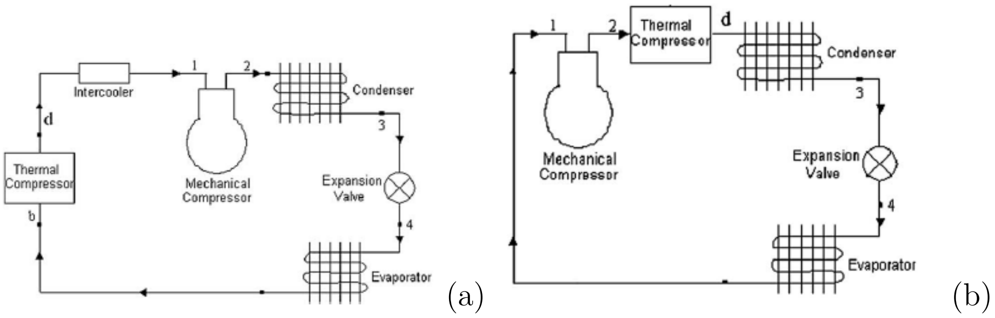


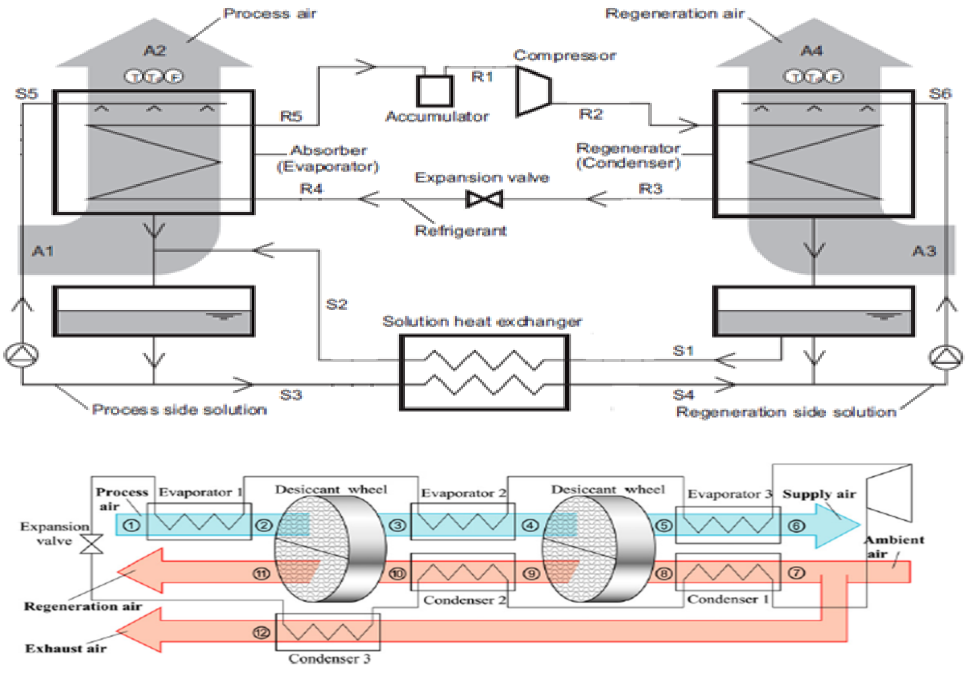
Figure 3.2 – Hybrid adsorption-compression cooling system : (a)LSTC ; (b)HSTC [155]

Table 3.2 – Summary of studies conducted on adsorption-vapor compression cooling system

		Obtained results		
Work (type of study)	**System description**	**COP and/or Efficiency** η	**Energy and/or Power**	**Comparison with conventional VCC**
[152] (S)	-Type of machine : LSTC, HSTC -VCC working with HFC 134a -Activated carbon adsorption bed	(T_d=90°C) , (T_{ad}=40°C), and (20°C<T_{ev} <15°C) → (COP > $COP_{ADSC,1stage}$ and $COP_{ADSC,2stage}$)		-Best results in ES when (T_{ev}-T_{cd})>40°C -(ES = 40%)
[154] (S)	-Type of machine : HSTC -Driven by waste heat from automobile -VCC working with CO_2 -4 activated carbon beds	($T_{gc,o}$=35°C) → (COP=9.5)	($T_{gc,o}$=35°C) → ($Q_{c,op}$=2.34 KW)	COP > (2-3)*COP_{VCC}

3.2.3 Combined desiccant-vapor compression cooling system

In hot and humid climates, the integration of a desiccant system with *VCC* has the advantage of removing latent and sensible load separately. It also improves the *COP* and reduces the energy consumption and the size of a *VCC*. Moreover, the comfort level could be reached by a higher supply chilled water temperature where the chilled water temperature can be raised from $7^{o}C$ to $13^{o}C$ [155]). As a result, the efficiency of the system is improved. The combination of a desiccant system with a *VCC* will also reduce the desiccant regeneration temperature from (70–80)^{o}C to (50–60)^{o}C [155] . Therefore, the heat rejected by the *VCC* condenser can contribute to the regeneration of the desiccant in the dehumidifier. Different configurations of desiccant-vapor compression cooling systems studied in the literature are shown in fig. 3.3. The summaries of studies conducted on combined desiccant-vapor compression cooling system are presented in Table 3.3.



(a)

(b)

Figure 3.3 – Hybrid desiccant-vapor compression cooling system : (a)*LD* with integrated absorber/evaporator and regenerator/ condenser [156] ; (b)two stage *SD* [157].

The use of a liquid desiccant-vapor compression hybrid system is encouraged for low humidity applications [169] . Niu et al. [158] proposed a double-condenser cooling system consisting of *LD* and a *VCC*. The *VCC* is composed by two condensers in series, the first being cooled by the liquid desiccant solution and the second being an air-cooled condenser. They studied the effect of solution flow rate, compressor rotation speed and air flow rate in the air-cooled condenser. They found that the 3 variables must be regulated simultaneously to optimize the system. Zhang et al. [159] studied a hybrid desiccant-vapor compression heat pump for cooling and heating applications. Yamaguchi et al. [156] simulated and tested a hybrid liquid desiccant-vapor compression heat pump where the absorber and the regenerator are integrated with the evaporator and the condenser, respectively. Bergero and Chiari [157] proposed a hybrid liquid desiccant-vapor compression cooling system where an air-solution membrane contactor is used for both the absorber and the regenerator. Bassuoni [160] investigated the performance of a *LD* integrated in a *VCC*. The process air is cooled and dehumidified in the evaporator box where a strong solution is sprayed over the top of the coil containing the refrigerant. Then, the dilute solution passes through the condenser box where it is preheated, thus the regeneration energy required is reduced. Zhao et al. [161] studied the performance of a system consisted of a *LD* combined with a *VCC* located in Shenzhen, China. The desiccant solution is cooled and pre-heated in the evaporator and the condenser, respectively. The chilled water and the process air are supplied separately into the zone to be cooled.

Fong et al. [162] studied a cooling system located in Hong Kong consisting of a solar *SD* combined with a *VCC*. They noted that the system performance was sensitive to the humidity in the zone. Angrisani et al. [163] tested a combined heat and power system consisting of heat pump, *SD* and small size cogenerator. They found that the system can guarantee a primary energy saving for outdoor air with a humidity ratio lower than 11.5 g/kg and a temperature in the range 25-36^oC. La et al. [164] tested a solar cooling system consisting of 10KW *TSDC* and a 20KW air-source *VCC* located in Shanghai. Then, a model of the cooling system was developed and studied for three different climates : Shanghai (humid), Beijing (temperate) and Hong Kong (extreme humid). They noted that the hybrid system studied is beneficial to power saving and not to energy saving, because the *COP* of the hybrid system is lower than that of a conventional *VCC*. Mandegari and Pahlavanzadeh [166] tested a desiccant wheel integrated with a *VCC* under various temperatures and humidity of the inlet air. They noted that this method of hybridization is more effective at high latent loads. Based on an economic study, they found that the system is more effective in countries where the price of energy is high. Wrobel et al. [166] developed a physical model for a hybrid air-conditioning system consisting of solar *SD* and a heat pump with a geothermal heat sink. In moderate climates, the model presented was able to predict the temperature and the relative humidity at the outlet of the desiccant

Table 3.3 – Summary of studies conducted on hybrid desiccant -vapor compression cooling system

			Obtained results		
Work (type of study)	**Type of DSC**	**Climate**	**COP, Efficiency (η) and/or Solar fraction (SF)**	**Operating Conditions**	**Comparison with conventional VCC**
[158] (S)	LD	Extreme humid		(($T_{cd,o}$ and N_{cp}) using Single condenser) > (($T_{cd,o}$ and N_{cp}) using Double condenser)	
[159] (S)	LD	Hot and humid	-Great influence of RH on COP_{LD} -(RH ↑) from 40% to 80% at (T_a=35°C) → COP ↑ (50%) -COP ∈ (4.3–5.7)		COP ↑ (20%)
[156] (E,S)	LD	Hot and humid	-COP=2.7 -COP_{VCC}=3.8 -(η_{cp} ↑) and (η in the solution heat exchanger ↑) → (COP_{to} ↑) and (COP_{VCC} ↑)	-(ω ↓) from 14 to 8.1 g/kg -(T_{air} ↓) from 30 to about 22°C	
[157] (S)	LD			(Q_l ↑) and (RH_a ↑) → the ratio of mass flow rate of solution in regenerator / mass flow rate of solution in dehumidifier should be increased	-(RH_a ↑) → (PS ↓) -(PS > 60%) especially at (>Q_l)
[160] (E)	LD		-($\dot{m}_{air}$ ↑) and ($\dot{m}_{ds}$ ↑) → (COP ↑) -($T_{ds,ev}$ ↑)→ (COP ↑) -(T_{rg} ↑)→ (COP ↓)	-($T_{ds,ev}$ ↑)→ (D↓) -(T_{rg} ↑) → (D↑)	-COP ↑ (54%) -The size of VCC ↓ from 9.13 kW to 5.27 kW -ES ∈ (33–46%)
[161] (E)	LD	Hot and humid	-COP_{to}= 4 -COP_{VCC} ∈ (3.7–4.1) -COP_{LD}=4.1		-(E_{cs} ↓) by more than 30%

[162] (S)	SD	Extreme humid	Study of the system in Chinese restaurant and in Wet market. -In Chinese restaurant : (SF=0.295) -In Wet market : (SF=0.286)		-In Chinese restaurant with high latent load : COP ↑ (5%) -In Wet market with lower latent load : COP↓ (0.3%)
[163] (E)	SD				-ES_{av}= 18% -(T_a ↑) and (RH↓) → (ES↑) -(η of electric grid↑) → (ES ↓) -CO_2 emissions ↓ (43%)
[164] (E, S)	SD	humid	-COP_{th}=1.24 -COP_{el}=11.48	Contribution of TSDC =35.7% of Q_c	-PS_{exp}= 25.5% -PS_{sim}= 34%
[164] (S)	SD	Extreme humid	-COP_{th}=0.87 -SF=27.7%		PS=22%
	SD	Temperate	-COP_{th}= 0.95 -SF=33.3%		PS=31% compared to VCC
[165] (S)	SD	Hot and dry	Using two –stage desiccant wheel , evaporators as cooler and condenser as heater -($T_{air,i}$ ↓) or ($\omega_{air,i}$ ↓) → (COP_{th} ↑) -COP_{th} ↑ to 6.3 by adding IEC	(e_{wheel} ↑) → ($N_{opt,wheel}$ ↓)	
[166] (E)	SD	Hot-dry			COP_{av} ↓ (36%) and $COP_{el,av}$ ↓ (5%)
		Hot-humid			COP_{av} ↓ (28%) and $COP_{el,av}$ ↑ (20%)
[167] (S)	SD		SF= 60% in term of regeneration heat		ES ∈ (81- 89%)
[168] (S)	SD		the evaporator of VCC removes a part of latent→ (COP ↑)		COP ↑ (94%)

wheel with an error less than 5% and 10%, respectively. Calise et al. [167] presented an energy and environmental study of a desiccant-based air handling unit coupled with Concentrating Photovoltaic Thermal (*CPVT*) collectors. An electric chiller is used to provide cooling after the dehumidification of the process air. The electricity required for the chiller and auxiliary need of the desiccant wheel are provided by the electricity produced in the *CPVT* collectors. Moreover, the excess of heat is used for the production of domestic hot water.

In a hybrid desiccant-vapor compression cooling system, the heat of condensation could be used for the regeneration of the desiccant material. A desiccant wheel normally requires a regeneration heat source temperature of about 80^oC [170] . However, if a multi-stage adsorption and regeneration processes are carried out, the regeneration temperature of the desiccant wheel will be reduced to 40–50^oC [168] which is close to the heat rejection temperature in the *VCC* condenser. Jeong et al. [168] proposed a hybrid cooling system with a four-partition desiccant wheel. The desiccant wheel is divided into four parts : two parts for the adsorption process and the remaining for the regeneration process. The *VCC* comprises three evaporators and three condensers in series. The process air passes through the evaporators and dehumidification parts of the desiccant wheel, whereas for regeneration of the desiccant wheel a stream of ambient air is introduced in the condensers, where it is heated before entering the regeneration parts. Tu et al. [165] analyzed the performance of a heat pump-driven *TSDC* under Beijing summer conditions. The evaporators and the condensers are considered as coolers and heaters, respectively, as in case of Jeong et al. [168] . They investigated the effect of the compressor power input, the ratio of the heat exchange areas of the evaporator and condenser to the total heat exchanger area, the wheel's rotation speed, and the inlet parameters of the processed air. They proposed the integration of Indirect Evaporative Cooler (*IEC*) between the return air and the processed air in order to recover the cooling capacity from the indoor exhaust air. Thus, an improvement of thermal *COP* by about 15% was registered.

3.2.4 Combined ejector-vapor compression cooling system

The ejector in a *VCC* usually replaces the throttling valve. Therefore, it reduces the throttling losses or the expansion irreversibility in the refrigeration cycle. Moreover, this integration offers several advantages such as raising the suction pressure in the compressor leading to smaller compressor size, *COP* improvement, work recovery and reduction of evaporator size by avoiding flash gas. The cooling system with an ejector as an expansion valve in a *VCC* (*EEV*) is shown in fig. 3.4(a). Based on previous work, Sarkar[146] observed that energetic or exergetic performance improvement by using ejector expansion depends mainly on the cycle operating conditions, working fluids and ejector geometries. Fong et al. [171] presented a dynamic simulation of a solar assisted ejector-vapor compression chiller under Hong Kong climate conditions. They compared the behavior of the system working with three different refrigerants. They observed that adding an ejector in a *VCC* working with R410A has no influence. Sarkar [172] studied and compared the performance of a hybrid cooling system - consisting of a *VCC* using an ejector as an expansion device- working with 3 different natural refrigerants. They also studied the influence of an internal heat exchanger in the cooling cycle. Liu et al. [173] presented a performance study of an enhanced trans-critical *VCC* with a controllable ejector. They studied the effects of ejector geometry on a cooling machine of a capacity of 10.3 KW. Yari [174] studied an ejector-expansion two-stage trans-critical *VCC*. He added an intercooler to cool the vapor resulting from the first compressor at constant pressure. The liquid leaving the gas cooler is sub-cooled in the internal heat exchanger located between the exit of the condenser and that of the evaporator. Shuxue and Guoyuan [175] presented an exergy analysis of a cooling system consisting of an ejector incorporated in a quasi-two-stage compression heat pump system. They found that the exergy of the heat pump is increased in the range $3 - 5\%$ by adding an ejector.

Bergander [176] and Bergander et al. ([177] and [178]) proposed a new design for an ejector-vapor compression cooling system. They studied the possibility of using a condensing ejector as a second step compression in a *VCC* (*EC*) as shown in fig. 3.4(b). According to Bergander et al. [177], this design allows the system to work with less mechanical work and an efficiency 35% higher than that of a *VCC*. In the new cycle, 2/3 of the compression work occurs in the compressor ; whereas the additional compression is provided by the condensing ejector : the vapor compressed in the compressor is sucked in the ejector where it will be mixed by liquid refrigerant ; therefore, a liquid stream with a pressure higher than both inlet fluids is obtained. A mathematical model based on the Computational Fluid Dynamic method has been developed and validated.

The cascaded ejector-vapor compression system (*CEV*) (fig. 3.4(c)) allows the cooling system to work with high capacity and efficiency since a *VCC* works with limited temperature difference. Moreover, a higher evaporator temperature is required in the ejector sub-cycle compared to individual *EJC* ; therefore, a higher efficiency is achieved, and a lower collector area is required in case of a solar driven ejector. Besides, the ejector sub-cycle could be eliminated if necessary and the system works as *VCC*. In order to investigate the contribution of a solar *EJC* in a *CEV*, Chesi et al.[179] compared 3 different system layouts : (1) conventional *VCC* operating alone, (2) solar ejector with a back-up *VCC* and (3) *CEV*. They studied the yearly power consumption of these systems in 4 cities with different climates. Based on simulation results, the *CEV* performed better than the

EJC alone expect for mild climates where a cooling load is required for a short period of the year. They pointed out that the ratio between the mass flow rates in both cycles is an important parameter in *CEV*. Later, Chesi et al. [180] studied the behavior of a 1 KW *CEV* -consisting of a *VCC* and a solar powered *EJC*- located at Palermo, Italy. The ejector cycle was used to cool the *VCC* condenser, when solar energy is available, whereas an air-cooled condenser completed the thermodynamic cycle of *VCC* if thermal energy is unavailable. The authors observed that the generator temperature in the *EJC* has the strongest effect on the system performance : the optimum generator temperature for the designed system was 65^oC, which is somewhat low. Vidal and Colle [181] modeled a *CEV* located in Florianopolis, Brazil. Yan et al. [182], experimentally investigated the influence of the evaporating, generating and condensing temperatures in the *EJC* on a *CEV*. Both *EJC* and *VCC* used an environmentally friendly refrigerant, R134a. Petrenko et al. [183] proposed a micro-trigeneration system (combined heating, cooling and power generation) composed of a cogeneration system and a 10 KW *CEV*, where *VCC* is a trans-critical machine type driven by the electricity generated. The *EJC* -driven by waste heat- used butane as working fluid. Huang et al. [184] tested, under different operating conditions, a hybrid solar-assisted cooling/heating system. The ejector driven by solar energy is used to cool the *VCC* condenser in a *CEV*. Moreover, the system could produce hot water by the heat pump in addition to solar collector supply.

Zhu and Jiang [94] designed a new hybrid system composed by an *EJC* and a *VCC* connected in parallel (*PEV*) (fig. 3.4(d)). The *EJC* and *VCC* shared a common evaporator. The ejector cooling cycle is driven by the waste heat from the condenser of the vapor compression unit. The study of the system working with 3 different fluids showed that an average *COP* increase of about 0.7% only was obtained for R134a, due to its low compressor discharge temperature in the range 70-90^oC. Moreover, the *COP* of both the *EJC* and the hybrid system is greatly influenced by the primary flow inlet pressure, area ratio and secondary flow inlet pressure of the ejector system. The summaries of studies conducted on combined ejector-vapor compression cooling system are presented in Table 3.4.

3.2.5 Combined evaporative-vapor compression cooling system

The performance of a conventional *VCC* decreases by about $2-4\%$ when the condenser temperature increases by one degree C [186]. Therefore, using an *EC* is a simple and cheap method that pre-cools ambient air before it passes over the condenser coil in very hot climate where ambient temperature is higher than 50^oC, leading to performance improvement and energy consumption reduction. The evaporatively cooled condenser has many advantages over an

Table 3.4 – Summary of studies conducted on combined ejector-vapor compression cooling system

		Obtained results			
Work (type of study)	**System description**	**COP and/or Efficiency (η)**	**Size and/or Cost**	**Operating conditions**	**Comparison with conventional VCC**
[171] (S)	-Type of machine :EEV -Solar electricity (Driven by PV panels)	Importance of ejector design for COP improvement			-using R22 : COP $\uparrow$ (2.85%), SF $\uparrow$ (2.61%), PS=4.13% -Using R134a : COP $\uparrow$ (4.18%), SF $\uparrow$ (3.49%), PS=7.05%
[172] (S)	-Type of machine : EEV -constant pressure mixing ejector			-($T_{ev} \uparrow$) or ($T_{cd} \downarrow$)$\rightarrow$ (optimum area ratio in the ejector $\uparrow$)	-Max (COP $\uparrow$) using propane -Min (COP $\uparrow$) using ammonia
[173] (E)	-Type of machine : EEV -Controllable ejector in trans-critical cycle	-($T_a \uparrow$)$\rightarrow$ (COP $\uparrow$) by up to 36% -($f_{cp} \downarrow$) $\rightarrow$ (COP $\uparrow$) by up to 147% -($\phi_{throat} \downarrow$) $\rightarrow$ (COP $\uparrow$) by up to 60%			
[174] (S)	-Type of machine : EEV -Ejector as an expansion in two stage trans-critical VCC -An intercooler and internal heat exchanger are added to the conventional cycle	-($T_{ev} \uparrow$) and ($T_{gc} \downarrow$) $\rightarrow$ (COP $\uparrow$) -($T_{ev,o} \uparrow$) and ($T_{gc,o} \uparrow$) $\rightarrow$ ($\eta \downarrow$) -COP_{max} and η_{max} are (12.5–21%) higher than cycle without heat exchanger and intercooler		($T_{ev} \uparrow$) $\rightarrow$ ($\nu \uparrow$)	
[180] (S)	-Type of machine : CEV -Solar Cascade solar EJC-VCC	SF=0.76			COP $\uparrow$ (15%)

[179] (S)	-Type of machine : CEV -Solar Cascade solar EJC-VCC				(ES > 20%) and (CO_2 emissions ↓) by more than 240 Kg/year
[181] (S)	-Type of machine : CEV -Cascade solar EJC-VCC	-COP=0.89 -SF=82%	Q_c=10.5KW → $A_{cl,opt}$=105m^2		
[182] (E)	-Type of machine : CEV -Air-cooled condenser in both cycles			$T_{ev,opt}$ ∈ (13-16°C) with a $T_{gn,opt}$ > 78°C	(COP ↑) ∈ (15.9-21%)
[176] (E, S)	-Type of machine : CE				-η ↑ (38%) -ES=16%
[178] (E)	-Type of machine : CE	-(c↓) and (ν ↑) → (COP ↑) -(ν ↑) → (η ↑)		Stability of ejector operation	
[183] (S)	-Type of machine : CEV -cascade EJC-trans-critical VCC combined with Cogeneration	(T_{ev} ↑) from -40°C to 0°C → (COP_{VCC} ↑) from 1.3 to 6.4			
[94] (S)	-Type of machine : PEV -EJC driven by waste heat	-($T_{cp,dc}$ > 100°C) → (COP↑) -(T_{ev} ↑)and (T_{cnd} ↓) → (COP↑)			COP↑ (8.6%) in case of R22 and (5. 5%) in case of R152a compared to VCC
[184] (E)	-Type of machine : CEV -Solar driven EJC -For cooling and water heating	COP ∈ (2.50-5.11)			-($T_{cd,VCC}$ ↓) ∈ (7.3–12.6°C) -PS ∈ (34.5-81.2%)

air-cooled one : minimal air pressure drop, lower fan power, lower condensing and compressor discharge temperatures, cheaper material and lighter weight. On the other hand, a *VCC* with an evaporatively cooled condenser could not be used in heating mode, due to the possibility of water freezing in the outside heat exchanger. Moreover, treatment of water pools is required in an evaporatively

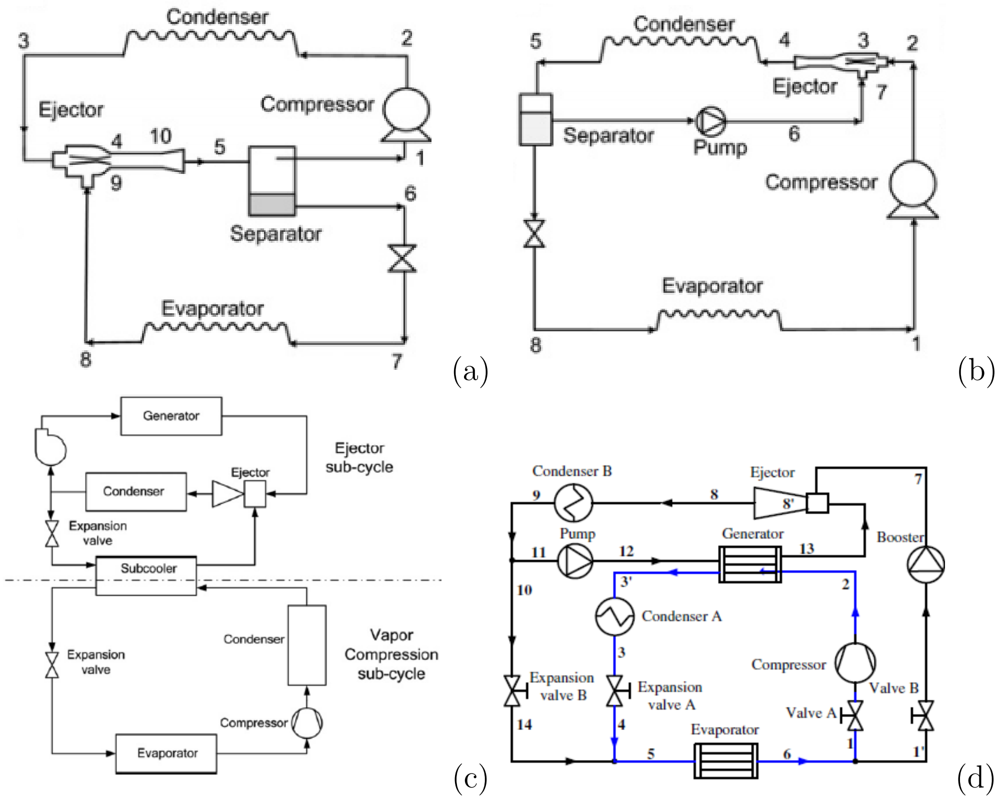


Figure 3.4 – Combined ejector-vapor compression cooling system : (a)*EEV* [185], (b)*CE* [185], (c)*CEV* [186], (d)*PEV* [99]

cooled condenser to avoid the growth of bacteria.

Generally, the evaporative cooling of the condenser is provided by two methods as shown in fig 3.5(a) and 3.5(b) : wetted cooling pads or water spraying (mist flows). Compared to cooling pads, the pulverization of water in a mist flows requires a high pressure pump with higher energy consumption. Moreover, the condenser in a spray water system leads to risks of fouling ; therefore, the water should be demineralized and treated before being sprayed. However, it requires lower maintenance and no pressure drop upstream of the condenser is created, as in the case of cooling pads.

Delfani et al. [187] experimentally investigated the performance of *IEC* used to pre-cool the air in the condenser of a *VCC*. They studied the system for different climates in Iran. Hao et al. [188] developed a mathematical model in order to evaluate the energy saving potential of a Direct Evaporative Cooler (*DEC*) combined with *VCC*. They studied the influence of different parameters on the energy saving potential. Hajidavalloo [185] tested a novel design for employing *IEC* in a 1.5 ton window-air conditioner. He installed 2 evaporative media pads on both sides of the condenser in order to increase the area available for cooling without increasing the total volume of the air conditioner. He recommended the use of an indirect evaporative condenser in very hot weather conditions instead of a conventional air condenser to save electric power and increase cooling capacity. Hajidavalloo and Eghtedari [189] experimentally investigated the performance of an air-cooled split-air-conditioner where the condenser is coupled with a *DEC*. Vrachopoulos et al. [190] studied a *VCC* with an incorporated evaporative condenser where the condenser acts as a cooling tower. In order to minimize water consumption in the system, they added a system of drop collector to the system. They found that the new design maximizes the life duration of the cooling machine. Nasr and Hassan [191] tested and simulated an evaporatively condenser incorporated in a residential refrigerator. The condenser consisted of a copper serpentine where thin sheets of wetted cloth are wrapped over each tube. They found that the condensing temperature in an evaporatively cooled condenser could be lower than the ambient temperature. Thu and Sato [192] tested an air conditioning system using an evapo-transpiration condenser under the summer weather condition of Tokyo, Japan. The copper tube of the condenser was covered with porous ceramics heat-exchanger, and two water tanks (top and bottom tanks from both sides of condenser) provided continuous wetting of the porous material. Hu and Hwang [193] used cellulose pads to provide evaporative cooling for a split air conditioning condenser. Hwang et al. [194] experimentally compared the performance of an evaporatively cooled condenser with that of a conventional air-cooled condenser. The evaporatively cooled condenser tubes are immersed in a water bath where air is blown over disks - partially submerged in the water bath and rotated by a direct-drive motor- carrying a thin film of water. The new system showed greater seasonal and steady state performance. Youbi-Idrissi et al. [195] developed a numerical model based on mass and energy conservation laws of a water sprayed condenser. Tissot et al. [196] carried out a numerical and experimental study of a refrigerating machine using a water spray upstream to pre-cool the air blown on the condenser. They investigated the possible performance improvements on a conventional heat pump in cooling mode. Yang et al. [197] conducted an experimental study to investigate the performance of air-cooled chillers with a water mist pre-cooling system under subtropical climate. The summary of the studies conducted on combined evaporative- vapor compression cooling systems are shown in Table 3.5.

Table 3.5 – Summary of studies conducted on evaporative-vapor compression cooling system

Work (type of study)	**Type of EC**	**Climate**	**Comparison with conventional VCC**
[187] (E)	IEC	Hot and arid	ES > 55%
[188] (S)	DEC	-hot and dry -hot and humid	-ES related to $V_{air,i}$, e_{pad} and climatic conditions -(ES in hot and dry climate) greater than (ES in humid and hot climate) -($V_{air,i} \uparrow$)$\rightarrow$ (ES $\downarrow$) -ES $\in$ (2.4-14%)
[185] (E)	IEC	Very hot	-PS=16% -COP $\uparrow$ (55%)
[189] (E)	SIEC	Hot	-($T_a \uparrow$) $\rightarrow$ greater (COP $\uparrow$) -PS >20% -COP $\uparrow$ (> 50%) -($\eta_{SIEC} \uparrow$) $\rightarrow$ (COP improvement $\uparrow$)
[190] (S)			COP $\uparrow$, $\dot{W}_{cp} \downarrow$, $V_{cd} \downarrow$, $\eta_{cp} \uparrow$, ES >58%, Installation cost $\downarrow$ (10%) and life duration $\uparrow$
[191] (E, S)	SIEC		-$T_{cd,VCC} \downarrow$ (20°C) -(> heat rejected by the condenser)
[192] (E)	SIEC	Warm and humid	-(T_a < 31°C) $\rightarrow$ (ES > 30%) -$T_{cod} \downarrow$ $\in$ (5–10 °C)
[193] (E)	SIEC		-COP $\uparrow$ (from 2.96 to 3.45) and Only extra 98W is needed -Payback period =10months
[194] (E)	DEC		-($Q_c \uparrow$) $\in$ (1.8- 8.1%) -(COP $\uparrow$) $\in$ (11.1-21.6%) -($\dot{W} \downarrow$)due to ($T_{cd} \downarrow$)
[195] (S)			COP $\uparrow$ (55%) and $Q_c \uparrow$ (13%)
[196] (E, S)	DEC	Hot and dry	At (RH≈ 20%) and (T_a)=308K : COP $\uparrow$ (28.9%), $Q_c \uparrow$ (7.2%) and ES= 16.7%

[197] (E)	DEC	Subtropical climate	-T_{cd} ↓ (7.2K) -COP ↑ (18.6%)

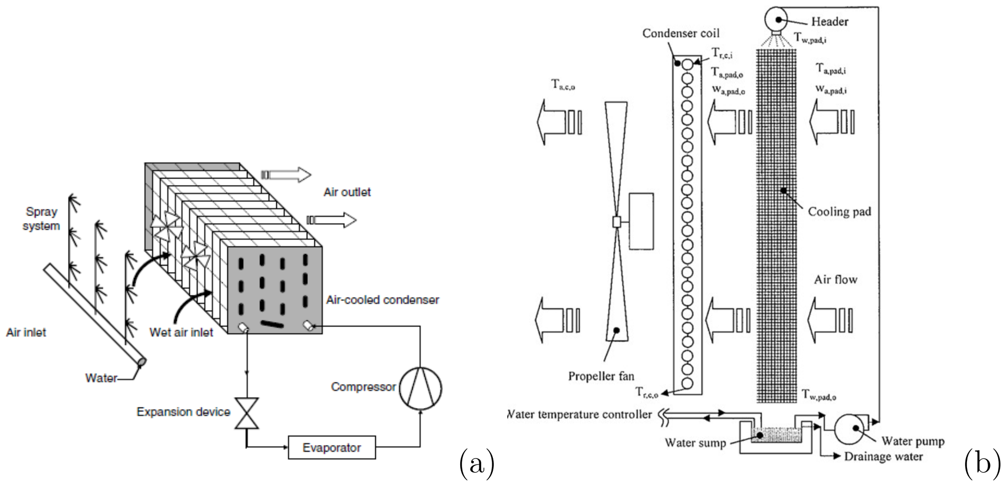


Figure 3.5 – Schematic of *VCC* (a)with Evaporatively cooled condenser using mist flow [198] ; (b)Evaporatively cooled condenser using cooling pad [199]

3.2.6 Combined *VCC* with desiccant and evaporative systems

Baniyounes et al. [198] tested and simulated, under TRNSYS, a hybrid solar cooling system located in Queensland, Australia. The system consisted of a $10m^2$ solar collector, $0.4m^3$ hot water storage tanks, a desiccant system (*DSC*), a direct evaporative cooler(*DEC*) and a *VCC*. The treated air enters the desiccant wheel where it is heated and dried by the rotating desiccant wheel. Then, the dried air is cooled in the heat exchanger to near ambient level. The resulting air passes through the *EC*, then through the *VCC* installed in the building. Beccali et al. [200] tested a hybrid cooling system installed in Palermo, Italy. The system consisted of a solar desiccant evaporative cooling system with two auxiliary cooling coils fed by *VCC*. The first coil is used to cool the air before it is dehumidified, and the second one controls the air temperature if the *IEC* could not achieve the comfort level. They noted that improving the controlling strategy and reducing the thermal losses in the storage allows better system performance. Jia et al. [201] studied a hybrid *DSC* combined with an air-cooled *VCC*. An *EC* is integrated in the system to humidify the process air. Dai et al. [199] studied a hybrid cooling system consisting of a *LD*, an *EC* and a *VCC*. The *DSC* removed the latent load from the process air blown on the evaporator, whereas the *EC* was used to cool the regenerated desiccant solution and the dehumidified process air. She et al. [202] presented a new energy-efficient *VCC* sub-cooled by a *LD-EC*. An *IEC* -fed by a very dry air from *DSC*- sub-cooled the refrigerant leaving the condenser. The liquid solution was pre-heated (before entering the regenerator) in a heat exchanger introduced between the condenser and the compressor of the refrigeration cycle. Moreover, the heat of condensation could be used for the regeneration of the *LD*. They studied the influence of different parameters on the performance of the cooling system. The summary of the studies conducted on *VCC* with *DSC* and *EC* are shown in Table 3.6.

3.3 Hybrid systems based on absorption cooling machine

The possibility of being driven with free energy namely solar, waste, biomass, geothermal, etc. makes *ABSC* economically appealing. However, it requires a higher source temperature compared to other thermally driven cooling system (adsorption, desiccant, etc.). Therefore, the hybridization of an *ABSC* with other cooling processes is an efficient method for cascaded heat utilization and reduction of the required heat source temperature.

3.3.1 Absorption cooling system with integrated compressor

An *ABSC* with an integrated compressor has the advantages of lower generation temperature and higher absorption temperature and higher solutions concentration difference compared to a typical *ABSC* [203]. This system can be driven by lower mechanical work than *VCC*, and can produce a lower evaporation temperature than that achieved in *ABSC*[204]. Compared to *VCC*, this sort of hybridization makes use of an environmentally friendly working fluid [205] and have better capacity control and lower pressure levels in the system due to the use of mixtures [206]. There are two configurations of hybrid absorption-compression cooling systems : in series and in parallel.

According to Zheng and Meng [204], there are two arrangements for series configuration as shown in fig. 3.6. In the configuration (a) (refer to 3.6(a)), the compressor is placed at low pressure side between the absorber and the evaporator (*SL*) leading to lower heat input and generator temperature with less power consumption. However, a larger compressor size is required due to the large specific volume of refrigerant vapor in the low-pressure side. In the configuration (b)(refer to fig. 3.6(b)), the compressor is integrated at high pressure

Table 3.6 – Summary of studies conducted on combined VCC with desiccant and evaporative systems

				Obtained results			
Work (type of study)	**Type of DSC**	**Type of EC**	**Climate**	**COP, Efficiency (η) or Solar fraction (SF)**	**Size and/or Cost**	**Operating conditions**	**Comparison with conventional VCC**
[198] (E, S)	SD	DEC	Subtropical	-SF=0.25 -(D $\uparrow$), ($T_{rg}\downarrow$), ($T_a\uparrow$) and/ or (RH $\uparrow$) $\rightarrow$ ($\eta_{SD}\downarrow$) -η_{max}=48%			ES= 18%
[200] (E)	SD	IEC	Hot and humid	-COP_{el}=2.4 -COP_{th}=1	$A_{cl}\downarrow$ (30%) if the heat rejected by VCC was recovered		-Qc ensured by VCC $\downarrow$: 53% of Q_c ensured by desiccant-evaporative system -ES $\approx$ 49%
[201] (E, S)	SD	DEC	T_a=30°C and RH=55%				-PS $\approx$ 37%
[199] (E, S)	LD	IEC	T_a=35°C and RH=40%				-($Q_c\uparrow$) $\in$ (20-30%) -COP $\uparrow$ -VCC size $\downarrow$
[202] (S)	LD	IEC		COP affected by desiccant concentration			-(COP $\uparrow$) by Using the heat of condensation for regeneration -System more beneficial at > T_{cd}

side between the generator and the condenser of the *ABSC* (*SH*). In this design, a smaller compressor size is required with a lower system performance. Zheng and Meng [204] proposed two fundamental concepts for the hybrid absorption-compression refrigeration systems in series : the ultimate refrigerating temperature and the behavior turning. The ultimate temperature is defined as the lowest refrigeration temperature for a given absorption cycle. Whereas, the behavior turning is the turning during the process from the state at the ultimate refrigerating temperature to a general cycle state. They found that the cycle performance varies from the dominance of the *ABSC* to the dominance of the *VCC* with the change in compressor outlet pressure. The optimal cycle performance was obtained at the behavior turning point. Ventas et al. [207] presented a numerical model of a hybrid *SL* working with an ammonia–lithium nitrate solution. They simulated the pressure ratios in the compressor for a wide range of hot water inputs in the generator. Later, Ventas et al. [208] tested the thermochemical compressor (absorber and generator) as part of a *SL* hybrid system. A simplified model describing the behavior of the cooling system has been developed. The simulation results compared with experimental data showed a good agreement. Meng et al. [209] studied a *SL* absorption-compression hybrid refrigeration cycle. A triple valve was installed before the compressor so that the hybrid can function as either *ABSC* or *VCC*. They investigated the low-grade heat utilization performance in the hybrid refrigeration cycle on the basis of two criteria : electricity saving rate and heat powered coefficient of performance which describes the benefit of replacing mechanical work with low-grade heat. Ramesh Kumar and Udayakumar [210] investigated the influence of different parameters on the performance of a Generator Absorber Heat Exchanger (*GAX*) absorption-compression cooler. The compressor is integrated at the low pressure side of the GAX chiller. They noted that the hybrid system can be driven successfully by low temperature energy sources. Kang et al. [72] modeled both arrangement of compressor (at high-pressure and low-pressure side) for a GAX cycle. They studied four configurations of hybrid *GAX* cycle : (A) *SL* (absorber pressure higher than that of the evaporator which is the same as that of standard *GAX* cycle) for performance improvement purposes, (B) *SL* (evaporator pressure lower than absorber pressure which is the same as that of *GAX* cycle) to provide a low evaporation temperature, (C) *SH* (generator pressure lower than condenser pressure which is the same as in *GAX* cycle) for reduction of desorption temperature, and finally, (D) *SH* (condenser pressure higher than generator pressure which is the same as in *GAX* cycle) to provide a hot water temperature (absorbent) higher than 100^oC, which is useful for heating application.

In the parallel configuration, the mechanical compressor is installed between the low- and high-pressure levels (absorber and generator) of the absorption chiller. There are two typical arrangements of parallel configuration (fig. 3.7) : (a) with a compressor between the low- and high-pressure levels of the absorption chiller (*PAG*), (b) which consists mainly of a mechanical compressor, a desorber and a resorber. In the second configuration, the desorber and the resorber replace the evaporator and the condenser, respectively. That is why the system is called resorption-compression (*RC*). In both configurations, the compressor pressure ratio is high in the range 3-6 [207] . Pratihar et al. [205] studied a *RC* hybrid

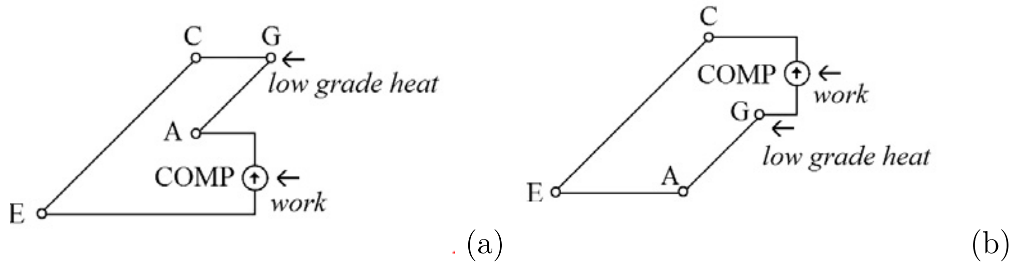


Figure 3.6 – Series configuration of hybrid absorption-compression cooling system : (a)*SL* ; (b)*SH* [207] . E : Evaporator, C :Condenser, G :Generator, A : Absorber, COMP : Compressor

system. The system consisted of an absorber, a desorber, a compressor, a solution heat exchanger, a separator (to facilitate the separation of the weak solution and vapors after desorption) and a mixer (to bring the hot vapors and the weak solution into thermal equilibrium). The machine uses an ammonia-water mixture as working fluid and produces cooling at the desorber. Pratihar et al. [205] developed a detailed model of a 400 kW refrigeration machine. They studied the effect of the relative solution heat exchanger area and mass flow rate of the weak solution on *COP*, cooling capacity and absorber heat load. Ayala et al. ([211] and [212]) studied a parallel configuration *PAG* of hybrid absorption-compression cooling system. They noted that the system is suitable for air conditioning systems at evaporator temperatures higher than 5^oC. The summary of the studies conducted on absorption cooling system with integrated compressor are shown in Table 3.7.

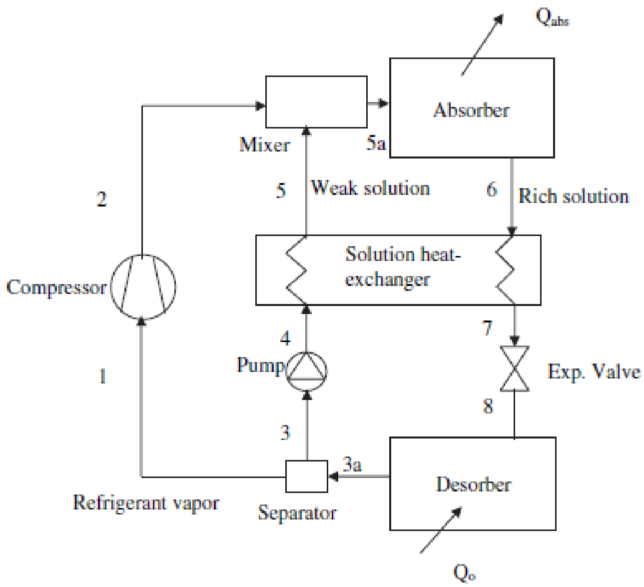
3.3.2 Desiccant-absorption cooling system

The integration of desiccant technology with *ABSC* is suitable for cascaded utilization of energy, since absorption systems usually require a higher heat source temperature (higher than 80^oC) than that of desiccant systems (50–80^oC [214]) and the exhaust heat from absorption systems can be reused by desiccant systems [215] . Moreover, the system can operate with a small electric power since both systems are thermally driven, and the performance of *ABSC* is improved. Fong et al. [213] proposed a solar hybrid system consisted of *ABSC* which removes the sensible cooling load and desiccant dehumidifier which removes the latent load from the cooling air. The chilled water was used to pre-cool the supply air and to feed the radiant cooling ceiling. The solar system was used to drive the *ABSC* generator and regenerate the *SD*. They also compared the cooling system using two types of chilled ceilings, the passive chilled beams and active chilled

Table 3.7 – Summary of absorption cooling system with integrated compressor

		Obtained results		
Work (type of study)	**System description**	**COP and/or Efficiency (η)**	**Operating conditions**	**Comparison with conventional ABSC and VCC**
[207] (S)	Type of machine : SL		-(c $\downarrow$) $\rightarrow$ ($COP_{el} \uparrow$) -(c $\uparrow$) $\rightarrow$ ($T_{hw,gn} \downarrow$) -(c $\uparrow$) $\rightarrow$ ($Q_c \uparrow$)	-$T_{hw,gn} \downarrow$ (> 24 °C) compared to ABSC -$COP_{el} \in$ (4-33) > COP_{VCC} -PS=10% compared to VCC
[209] (S)	-Type of machine : SL -Solar driven generator	-COP_{th} =0.322		-ES > 5% compared to VCC -$T_{ev} \downarrow$ compared to ABSC
[208] (E, S)	-Type of machine : SL -System based on an adiabatic absorber			-($T_{hw,gn} \downarrow$) $\in$ (30-47°C) compared to ABSC -COP $\uparrow$ at (> $T_{hw,gn}$) compared to ABSC
[210] (S)	-Type of machine : SL -GAX absorption-compression cooling system	-Optimum degassing ratio=0.4 $\rightarrow$ Max COP -($T_{gn} \uparrow$)and/or ($P_{ab} \uparrow$) $\rightarrow$ (COP $\uparrow$)	-($T_{gn} \uparrow$) $\rightarrow$ (cr $\uparrow$) -($P_{as} \uparrow$) has no effect on cr	-COP $\uparrow$ (30%) compared to conventional GAX
[72] (S)	-Type of machine : SL, SH -GAX absorption-compression cooling system. -Four systems were studied : — SL ($P_{ab} > P_{ev}$ and $P_{ev} = P_{ev,GAX}$) — SL ($P_{ab} > P_{ev}$ and $P_{ab} = P_{ab,GAX}$) — SH ($P_{gn} < P_{cd}$ and $P_{cd} = P_{cd,GAX}$) — (D) SH ($P_{gn} < P_{cd}$ and $P_{gn} = P_{gn,GAX}$)	Type (B) : -($P_{ev} \downarrow$)$\rightarrow$ (COP $\downarrow$)	Type (B) : -($P_{ev} \downarrow$) $\rightarrow$ ($T_{ev} \downarrow$) -$T_{ev} \in$ ((-80)-(-50)°C) -COP $\in$ (0.3-0.58) Type (D) : -the system is suitable for heating application	-Type (A) : COP $\uparrow$ (24%) compared to GAX ABSC -Type (B) : $T_{ev} \downarrow$ compared to GAX ABSC -Type (C) : COP $\uparrow$ (>19%) and ($T_{dc} \downarrow$) (from 190–200 °C in standard GAX cycle to 168°C) $\rightarrow$ no corrosion problem

[205] (S)	-Type of machine : RC -Cooling effect produced at the desorber.	-($\dot{m}_{ws}\downarrow$) and/or (RHXA $\uparrow$) $\rightarrow$ (COP $\uparrow$) -(RHXA$_{op}$) =45% $\rightarrow$ COP= COP$_{max}$ -(RHXA< 30%) : (RHXA $\uparrow$) $\rightarrow$ (COP $\uparrow$) and (Q $\downarrow$)		(COP < COP$_{VCC}$) due to low temperature glides obtained in the absorber and desorber
[211] (S)	-Type of study : PAG -Cooling produced at the evaporator			$Q_c\uparrow$ (10%) without any additional energy input
[212] (E)	-Type of machine : PAG	90% compression and 10% absorption $\rightarrow$ Maximum COP obtained		



(b)

Figure 3.7 – Parallel configuration of hybrid absorption-compression where cooling is produced at the desorber : (a)*PAG* [213] ; (b)*RC* [208]

beams. They found that the passive chilled beams is the more energy-efficient mode used in the hybrid system studied. Fong et al. [216] studied different designs of solar hybrid desiccant cooling system. They noted that the hybridization of desiccant with *ABSC* is attractive since it uses evacuated tubes instead of Photovoltaic which is more expensive. Khalid Ahmed et al. [217] studied a hybrid open-cycle consisted of a *ABSC* and *LD* system. The condenser is eliminated in the open-cycle *ABSC* and the weak solution (from absorption chiller and dehumidifier) is regenerated by solar energy. The air is dehumidified by the *LD*, and then it is cooled by chilled water from the *ABSC*. The summary of the studies conducted on desiccant-absorption cooling system are shown in Table 3.8.

Table 3.8 – Summary of studies conducted on desiccant-absorption cooling system

			Obtained results		
Work (type of study)	**Type of DSC**	**Climate**	**COP, Efficiency (η) and/or Solar fraction (SF)**	**Operating Conditions**	**Comparison with conventional systems**
[213] (S)	SD	Hot and humid	(SF_{av} and $COP_{av,ABSC}$ using passive mode) > (SF_{av} and $COP_{av,ABSC}$ using active mode)		(ES >36.5%) per year compared to VCC
[216] (S)	SD	Hot and humid	-SF=0.759 -COP_{ABSC}=0.784 -COP_{SD}=0.835		-(ES >35.2%) per year compared to air-cooled VCC -(ES > 33.6%) compared to water cooled VCC -(ES >57.9%) compared to solar DSC
[217] (S)	LD		COP_{max}=1.25	-($T_w \downarrow$)and/or ($\omega \downarrow$)$\rightarrow$ (COP $\uparrow$)	-COP $\uparrow$ (>50%) compared to ABSC $\rightarrow$ Suitable for hot and humid climate

3.3.3 Ejector-absorption cooling system

The integration of an ejector in an *ABSC* allows the system to work with three pressure levels instead of two pressure levels. Therefore, the hybrid ejector-absorption cooling system offers different advantages : it decreases the circulation ratio, the evaporation temperature and the generation temperature or it raises the cooling-water temperature in the absorber and the condenser. Moreover, the integration of an ejector increases the cooling capacity of the *ABSC* without increasing the capacity of the absorber. Therefore, the performance of the system could be improved without increasing its complexity. According to Srikhirin et al. [218], an increase greater than 20% in the *COP* compared to *ABSC* was achieved with the combined ejector-absorption cycle. There are three different approaches to the use of ejector with an absorption chiller (fig. 3.8) : (a)ejector located at the absorber inlet replacing the solution expansion valve (*EEA*), (b) ejector at the inlet of the condenser (*ERC*) and (c) vapor recompression–absorption cooling system (*RA*).

The ejector integrated at the inlet of the absorber is used to maintain the absorber pressure at a level higher than that of the evaporator by reducing the solution circulation. In this approach, the high pressure solution from the generator is the ejector motive fluid. Levy et al. [219] developed a steady state mathematical model based on mass and heat transfers between the liquid and gas phases in a *EEA* system. Sozen and Ozalp [220] presented an energy analysis of a *EEA*. They noted that the system can be used as a heat pump due to the increased absorber temperature. Vereda et al. [221] investigated numerically the feasibility of a *EEA* using an ejector whose nozzle area is adjustable while the rest of the ejector dimensions are fixed.

When the ejector is integrated at the inlet of the condenser, the generator operates at a pressure higher than that of the condenser. Thus, the ejector entrains more vapor from the evaporator which led to higher cooling effect produced. Wang et al. [222] proposed a combined power and cooling system consisting of Rankine cycle and an *ERC* hybrid system. The chiller used ammonia –water as working fluid. Sirwan et al. [223] proposed the integration of a flash tank to the *ERC*. The flash tank is used to improve the quality of the refrigerant that enters the evaporator by removing the flash gas created during the expansion process. It also enhances the ejector efficiency, and allows the system to work at higher condenser temperatures. The study showed that adding the flash tank to the *ERC* can be considered a novel enhancement. In the hybrid *RA*, the heat of condensation is recovered by a steam jet heat pump and supplied to the generator of the absorption chiller. Part of the vapor desorbed is entrained and compressed by the flow coming from the steam generator of the ejector heat pump. The combined flow is then fed in the generator where it condenses and entrains more vapor to be generated. Eames and Wu [224] simulated a *RA*. They studied the influence of the operating conditions on the *COP* of the system. Eames and Wu [225] experimentally investigated the performance of a *RA*. They noted that the *COP* is less than that predicted theoretically [226] ; therefore, the operation of the steam ejector should be improved to raise the *COP* of the cooling machine. The summary of the studies conducted on ejector-absorption cooling systems are shown in Table 3.9.

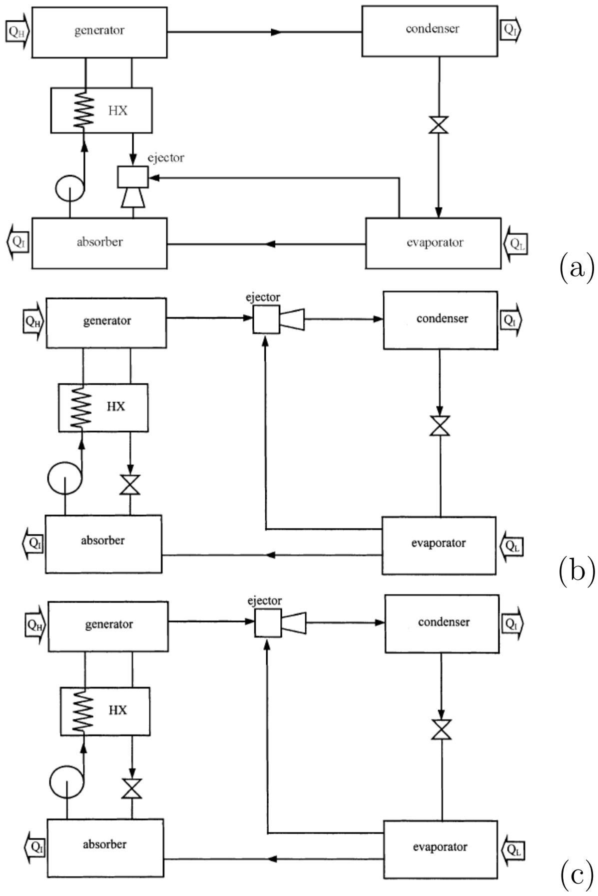


Figure 3.8 – Ejector-absorption cooling system : (a)*EEA* ; (b)*ERC* ; (c)*RA* [221]

3.4 Hybrid systems based on adsorption cooling machine

ADSC could be driven by a lower heat source temperature compared to *ABSC*. However the low coefficient of performance and the intermittence of the cooling cycle hindered the application of *ADSC*. The hybridization of adsorption with other process could enhance the performance of the cooling machine and make the use of low temperature source such as waste possible for cooling production. Moreover, it is an efficient method to mitigate the intermittence of the cycle.

Table 3.9 – Summary of studies conducted on ejector-absorption cooling systems

		Obtained results		
Work (type of study)	**System description**	**COP and/or Efficiency (η)**	**Operating conditions**	**Comparison with conventional systems**
[219] (S)	Type of machine : EEA	-(Droplet size $\downarrow$) $\rightarrow$ (diffuser performance $\uparrow$)	-(diffuser angle $\downarrow$) $\rightarrow$ (pressure recovery $\uparrow$)	
[220] (S)	Type of machine : EEA			Compared to ABSC : -COP $\uparrow$ (> 49%) -(system dimensions $\downarrow$) $\rightarrow$ (first cost $\downarrow$) -T_{ev} $\downarrow$ (>5°C) -T_{gn} $\downarrow$ (105°C)
[221] (S)	Type of machine : EEA	Great influence of ϕ_{mixing} on COP	Hybridization of EJC and ABSC is convenient for $T_{gn} \in$ (81-92°C)	-T_{gn} $\downarrow$ (9°C) with a minor loss on COP_{max}compared to ABSC
[222] (S)	Type of machine : ERC		Influence of T_{gn}, T_{cd}, T_{ev}, and basic solution ammonia concentration on Q_c	Q_c $\uparrow$ from about 224 KW to 246 KW compared to ABSC
[223] (S)	-Type of machine : ERC -A flash tank vessel is added between the condenser and the evaporator	Adding a flash tank $\rightarrow$ (COP $\uparrow$) from about 0.46 to 0.84	Max. exergy loss in the evaporator	
[224] (S)	Type of machine : RA	-Solution concentration optimum $\rightarrow$ COP_{max} -(T_{hs} $\uparrow$) $\rightarrow$ (COP $\uparrow$) -COP influenced by temperature difference between the steam and the solution		
[225] (E)	Type of machine : RA	-(COP $\uparrow$) depend on the performance of the steam ejector pump -COP directly proportional to ν of the ejector		-(COP $\uparrow$) (from 0.7 to 1.03) compared to ABSC

3.4.1 Compressor driven adsorption chiller

The integration of a compressor yields tangible benefits for an *ADSC*. It makes it possible to use refrigerants with a high pressure level, which is not possible in *VCC*. Also, it uses the desorption heat which is higher than the vaporization heat for some fluids ; therefore, a higher *COP* is achieved compared to *ADSC*. Sward and LeVan [227] proposed and simulated a compressor driven adsorption cooling system (fig. 3.9). The system consisted of two adsorbent beds connected by a compressor that drew fluid from the low temperature desorbing bed, and passed it into the high temperature adsorbing bed. The simulation results showed that the shape and the slope of the isotherm have a great impact on the performance of the system : The adsorbents with large adsorption capacities and steeper isotherms provide greater *COP*. Moreover, increasing the heat of adsorption improves the *COP* of the cooling system and increases the pressure ratio required to achieve a given level of performance. The authors simulated the system using different adsorbate/adsorbent pairs. They found that NH3/silica gel has the highest *COP*, whereas the *COP* of the hybrid system is still lower than *VCC*, and enhancing the performance requires a pressure ratio that makes the equipment cost prohibitive.

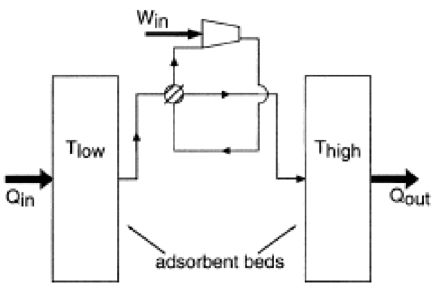


Figure 3.9 – Schematic of compressor driven adsorption cooling system [228]

3.4.2 Evaporatively cooled adsorption machine combined with desiccant dehumidifier

Dai et al. [229] studied a solar cooling system that combines a *SD* and an *ADSC*. The system, used for grain storage in China, is an environmental friendly technology that offers great energy saving and low operation costs. Moreover, the solar collector placed on the roof of the depot reduces the heating load of the grain storage depots, and the desiccant dehumidification lowers the cooling load of the chiller. The *ADSC* works intermittently. During the day, the adsorption bed is heated by solar radiation. Then the desorption process occurs when the bed pressure is higher than that of the evaporatively cooled condenser. The refrigerant is turned into liquid state and stored in the rectifier. At night, the adsorption process starts when the bed is cooled by natural convection and irradiation and the pressure in the adsorption bed becomes lower than that in the evaporator. The desiccant system removes the moisture produced by the grain. For a $20m^2$ solar collector and a adsorption bed maximum temperature about 100^oC, the simulation results showed that the system achieved a *COP* higher than 0.4 and an outlet temperature lower than 20^oC. The authors studied the effect of different parameters on the system performance. They found that the *COP* increases with increasing ambient temperature, humidity (linear relationship) and effectiveness of the heat exchanger (parabolic relationship). They observed that increasing the mass flow rate of refrigerant improves the *COP* but it is not helpful in decreasing the outlet temperature of the air. Therefore, the mass air-flow rate should be controlled to achieve best results for *COP* and temperature of the cooling system.

3.4.3 Ejector-adsorption cooling system

The ejector-absorption cooling system combined the advantages of *EJC* and *ADSC*. Moreover, it improves the *COP* and the cooling capacities and overcomes the intermittence of both cycles. Zhang and Wang [230] studied a solar ejector-adsorption system used for water heating and refrigerating. The heating system consisted of a solar adsorber (zeolite) and a water tank, while the cooling cycle comprises an ejector, a condenser and an evaporator. During day, the solar adsorber is heated by solar radiation, and the water is desorbed from zeolite. When the temperature and pressure reach certain values in the adsorber, the ejector refrigeration starts while the adsorber is connected to the ejector and disconnected from the evaporator. Li et al. [228] proposed a novel hybrid ejector-adsorption cooling system where the ejector provides cooling in the daytime and adsorption at night. They noted that the higher desorbing temperature can be obtained when a vacuumed tube adsorber or a Compound Parabolic Collector adsorber is used. The summary of the studies conducted on ejector-adsorption cooling systems are shown in Table 3.10.

3.4.4 Thermoelectric-adsorption cooling system

Gordon et al. [231] proposed a chiller that combines adsorption and thermoelectric cooling devices. The heat rejected to the environment by the thermoelectric device is recovered to drive refrigerant desorption in the *ADSC*. On the other hand, the heat rejected by the adsorber is regenerated by the thermoelectric device at its cold junction. The system has the benefit of large cooling densities at

Table 3.10 – Summary of studies conducted on ejector-adsorption cooling system

		Obtained results	
Work (type of study)	**Operating principle**	**COP and/or Efficiency (η)**	**Comparison with conventional systems**
[230] (S)	System for water heating and refrigeration	-Thermodynamic performance properties affected by the ejector -($\nu\uparrow$) and ($T_{d,s}\downarrow$) $\rightarrow$ (COP $\uparrow$) -In day time : COP=0.1 at Q_c = 0.15 MJ per kg zeolite -In evening : COP=0.23 at Q_c =0.34 MJ per kg zeolite	-COP$\uparrow$ (10%) compared to ADSC
[228] (S)	Refrigeration by EJC in the daytime and by ADSC at night	-(T $\uparrow$) and (P $\downarrow$) in the adsorber bed $\rightarrow$ ($COP_{EJC}\uparrow$) -If more adsorbent is used $\rightarrow$ ($COP_{EJC}\uparrow$) -COP_{to}=0.4	

high efficiency, no moving parts and no harmless materials. The results showed that the hybrid system reached a *COP* of about 1.8, which is higher than that of the *ADSC*. They noted that the system can offer lower temperatures at elevated *COP* compared to existing cooling systems used for micro-electronic cooling applications.

3.4.5 Desiccant-evaporative hybrid cooling system

Since the desiccant dehumidification process converts latent energy to sensible energy, auxiliary coolers like evaporative coolers or other chillers must be incorporated to remove the sensible heat, thereby achieving a cooling effect [232] and improving the efficiency of energy use in desiccant dehumidifiers [233] . On the other hand, the performance of *EC* at high wet bulb ambient temperature is limited [234] . Therefore, the integration of a *DSC* and an *EC* cooler in hot and humid climates could enhance the evaporative cooling effect and improve the performance of the cooling system.

Numerous works investigated the contribution of a regenerative evaporative cooler (*REC*) in a hybrid desiccant-evaporative system (fig. 3.10). In such system, the dehumidified air is divided into two parts : one part for producing chilled water in the EC and the other one for air conditioning. La et al. [235] designed and modeled a cooling system combining the technologies of *OTSDC* and *REC*. The system has the advantage of separate temperature and humidity control without increasing electrical load. The simulation was carried out under three typical conditions. Later, La et al. [236] compared the system defined by La et al. [235] with an *OTSDC*. They proposed a novel rotary desiccant cooling system consisted of isothermal dehumidification (the air flows alternatively over infinite desiccant wheels and intercoolers [215]) and the *REC*. The return air mixed with ambient air was used for regeneration of desiccant wheels. In Ge et al. [237], a simpler desiccant system with *REC* has been proposed. A desiccant coated heat exchanger (*DCHE*) where the desiccant material is coated to the surface of fin tube heat exchanger was used. The dehumidification process was self-cooled using the chilled water resulting from *EC*. The feasibility and the performance of this system under ARI summer conditions were investigated. El Hourani et al. [238] designed and modeled a *DSC* with two stage *EC*. The system installed in Beirut-Lebanon uses 100% fresh air. The outdoor air was firstly dehumidified using a desiccant wheel regenerated by solar energy (parabolic concentrator). A fraction of the dehumidified air was cooled by passing through an evaporative cooling pad, and then mixed with the bypass air. Finally, the resulting air was cooled locally in the personalized *EC* which allowed the occupants to individually control their thermal comfort level. Uçkan et al. [239] designed and tested a desiccant-evaporative cooling system. Before experiments, they carried out a theoretical analysis in order to maximize the performance of the system. Contrary to other studies where return air is reused for desiccant regeneration, they found that it is better to use outdoor air : it increases the regeneration heat and the capacity of desiccant wheel. Moreover, they introduced a rotary regenerative type heat exchanger to the conventional desiccant-evaporative cooling system. The outdoor air passes through a rotary regenerative type heat exchanger. Then, it was heated before passing through the desiccant wheel. Finally, the regeneration air was sent to the rotary regenerative heat exchanger before being discharged to the atmosphere. Baniyounes et al. [240] studied the effect of a hot water storage tank and solar collector area on the performance of a solar regenerated desiccant-evaporative cooling system. The system was simulated under TRNSYS 16 software using the climatic conditions of Rockhampton-Queensland, Australia.

Kim et al. [241] studied the contribution of integrating *LD* on *IEC* and *DEC* operations. In order to maintain the absorber temperature of *LD*, a cooling tower has been added. The system used 100% outdoor air for *LD*, while the *IEC* process air was a mixture of three air flows : return, outdoor and dehumidified air. A solar thermal system provided the regeneration heat. Yan et al. [242] proposed a new ice producing system consisted of *LD* and evaporative super-cooled water. The dehumidification could reduce the water vapor partial pressure of air below the pressure corresponding to the triple point of water. Therefore, the ice slurry producing requirements are achieved. In the proposed system, the dehumidified air was pre-cooled by water pumped from the *EC* rather than mechanical regeneration cycle. Two super-cooled *EC* were used in the system : pre-cooling and ice producing *EC*. Alizadeh [243] tested solar *LD* installed in Brisbane-Queensland, Australia. A cross flow polymer plate heat exchanger (*PPHE*) for dehumidification/*IEC* and a cooling pad for humidification (as *DEC*) were used. In a *PPHE*, the primary air –in direct contact with the desiccant solution-is cooled and dehumidified. On the other side, the secondary air coming from conditioned space -which is dryer than outside air- is in direct contact with water. This heat exchanger has the advantage of absorbing the heat of dehumidification process by the water. Therefore, a better performance due to nearly isothermal dehumidification is achieved. Jason and Eric [244] modeled and tested a *LD* combined with *EC*. A new design of *DSC* was used : The process air channel is separated from the exhaust air channel by a thin plastic plate. The exhaust channel surface is wetted by water film, therefore cooling the dehumidification process. The validation of experiment and simulation results showed a good agreement, with an error of $\pm 10\%$. The summary of the studies conducted on desiccant-evaporative cooling systems are shown in Table 3.11.

3.5 Multi- evaporator cooling systems

In different studies, the cooling effect was produced at different evaporator temperature levels where different cooling machine (*VCC*, *ABSC* or *ADSC*) were combined. In this sort of hybridization, the electrical energy used to produce cooling could be totally replaced (during a period of the day especially at peak load) when free energy is available. However, the first cost of the system is very high.

Liu et al [246] presented a hybrid system that covers power and thermal demands of a building located in Beijing, China. The system consisted of cogenerators, a *LD*, a *VCC*, an *ABSC* and a gas boiler. During the cooling season, the exhaust heat (waste) from the cogenerator was firstly used to regenerate the desiccant, and then to drive the double-effect *ABSC*. If the waste heat was not enough to drive the *ABSC*, the sensible load was removed using the *VCC*. Ma et al. [247] proposed a hybrid cooling system (total load 60KW) for the green building demonstration project in Shanghai. It consisted of two *ADSC*, a *VCC* and a *LD*. The *ADSC* were driven by a solar system. The condensation heat rejected by

Table 3.11 – Summary of studies conducted on desiccant - evaporative cooling system

			Obtained results		
Work (type of study)	**System description**	**Climate**	**COP and/or Efficiency (η)**	**Operating conditions**	**Comparison with conventional systems**
[235] (S)	-Type of DSC : SD -Type of EC : DEC -REC : chilled water used to pre-cool supply air -Dehumidification by OTSDC	(a) extreme hot and humid (b) ARI summer (T_a=35°C and RH=40%) (c) ARI humid (T_a=30°C and RH=60%)	T_{rg}=80°C $\rightarrow$ (COP_{th} >1) and (COP_{el}=8)	(a) $T_{rg} \in$ (80-90°C) $\rightarrow$ $T_{chw} \in$ (18-20°C) (b) $T_{rg} \in$ (80-90°C) $\rightarrow$ (T_{chw} <18°C) (c) $T_{rg} \in$ (80-90°C) $\rightarrow$ (T_{chw} <18°C)	
[236] (S)	-Type of DSC : SD -Type of EC : DEC -REC : chilled water used to pre-cool supply air -Dehumidification by OTSDC	(a) extreme hot and humid (b) ARI summer (T_a=35°C and RH=40%) (c) ARI humid (T_a=30°C and RH=60%)		(a)-($T_{sp,min} \downarrow$) from 18.0°C to 13.0°C -(Trg,min $\downarrow$) from 75°C to 65 °C (b)-($T_{sp,min} \downarrow$) from 13.5°C to 7.9°C (c)-($T_{sp,min} \downarrow$) from 14.2°C to 9.2°C	Compared to conventional desiccant-evaporative system, ES $\uparrow$ (> 10%)
[245] (S)	-Type of DSC : SD -Type of EC : DEC -Two stage isothermal dehumidification -Cross flow heat exchanger after each stage -REC	extreme hot and humid, ARI summer and ARI humid			Compared to conventional rotary desiccant system : -($T_{rg} \downarrow$) from 80°C to 60°C -$Q_c \uparrow$ (40%) -($\epsilon \uparrow$) from 18% to 29.1% -COP $\uparrow$ due to cascade energy utilization by 2-stages dehumidification
[237] (S)	-Type of DSC : SD -Type of EC : DEC -DCHE and REC	ARI summer conditions		-$T_{rg} \in$ (50-80°C) -high feasibility under high RH -Low influence of T_a	-$T_{rg} \downarrow$ compared to conventional SD -$Q_c \uparrow$ (30%) compared to system without REC

[238] (S)	-Type of DSC : SD -Type of EC : DEC -Two stage DEC -Solar energy for regeneration	hot and humid			Compared to one stage EC : -T_{rg} ↓ -$\dot{m}_{air,sp}$ required ↓ -$T_{air,sp}$ tolerated ↑ -ES=16.15% -Water consumption ↓ (26.93%) -$RH_{air,sp}$ ↓ (> 4.33%)
[240] (S)	-Type of DSC : SD -Type of EC : IEC Solar regenerated SD	subtropical climate	-COP > 07 -SF >22%		CO_2 emissions ↓ (4.4 tons per year) compared to VCC
[239] (E)	-Type of DSC : SD -Type of EC : DEC -Rotary regenerative type heat exchanger is added -Outdoor air used for regeneration	Hot and humid	-$COP_{to} \in$ (0.64-0.76) -COP_{th}=0.8	-T_a ↓ from 35°C to 14°C -34% heat recovery for E_h -75% heat recovery for E_c -25% of Q_c produced by DEC	
[241] (S)	-Type of DSC : LD -Type of EC : DEC, IEC -Regenerator connected to solar system, and absorber connected to cooling tower -Dehumidified air passes through IEC then DEC	Hot and humid			-E_{cs} ↓ (51%) compared to conventional VCC -At A_{cl}=70m^2, $E_{cs}$is the same as that consumed in conventional VCC
[242] (E, S)	-Type of DSC :LD -Type of EC : DEC -Ice producing system using super-cooled evaporative cooler				compared with the traditional super-cooled water method : -Ice producing rate ↑ (33.8%) -COP ↑ -Ice blocking is resolved

[243] (E)	-Type of DSC : LD -Type of EC : DEC, IEC -Solar regenerated -PPHE for dehumidification	tropical climate	-$\dot{m}_{air,op}$=1000 L/s and $\dot{m}_{dsc,op}$= 3 L/min $\rightarrow$ best COP -η=82% -COP$_{el}$ $\approx$ 6	-IEC $\rightarrow$ possibility of producing potable water	

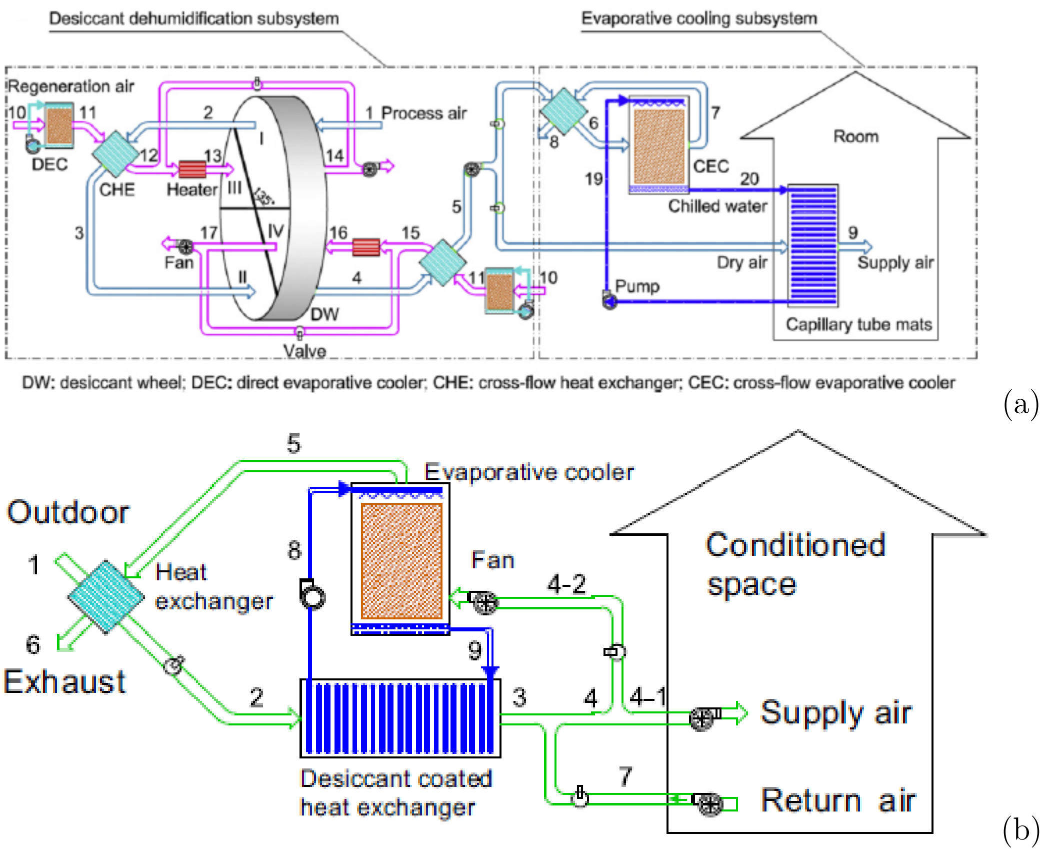


Figure 3.10 – Solid desiccant dehumidification with *REC* : (a)*OTSDC* [238] ; (b)*DCHE* [240]

the *VCC* was used for desiccant regeneration. The chilled water produced by the *ADSC* and the *VCC* was collected and is then distributed into two flows : (1) to indoor fan coils to deal with sensible load, (2) to cool the concentrated solution and cool the air in the desiccant system. The return chilled water is collected and then delivered to the *ADSC* and evaporator of the *VCC*. A factor called the "optimal match factor" that describes the optimal proportion of the rated refrigerating power of *ADSC* in total load was defined to study the contribution of adsorption cooling. Yari et al. [248] studied a dual evaporator cooling system consisting of an *ABSC* and ejector-expansion trans-critical cascade CO_2 cycle. They investigated the behavior of the system using three types of *ABSC* : the single-effect or double-effect series-flow or double-effect parallel-flow cycle. In the double-effect series-flow, the solution flow is directly pumped from the absorber to the high-pressure generator and then to the low-pressure generator. However in the parallel flow machine, the solution flow is distributed between the high pressure and low pressure generators. A fraction of the total heat required for the *ABSC* is provided by the compressed CO_2 in the ejector-expansion trans-critical cascade cycle, and the remaining part is received from solar energy. The simulation results showed the best performance is obtained when the double-effect parallel *ABSC* is combined with an ejector-expansion system. However, the use of a single-effect *ABSC* is more attractive due to its lower complexity. Agrawal and Karimi [249] presented a thermodynamic analysis of waste heat based triple effect refrigeration cycle. It consists of a Rankine cycle, *EJC*, *ABSC* and transcritical *VCC*. The *VCC* is driven by the power output of the Rankine cycle, while the *ABSC* generator and the heat recovery generator of Rankine cycle are run by industrial waste heat flue gas. Yari et al. [250] proposed a novel combined heat and power system. It consisted of a dual-evaporator system with dual-source (solar and electrical energies) combined with the Organic Rankin Cycle and supercritical CO_2 power cycle. The cooling effect is produced at two temperature levels : negative temperature by ejector-expansion trans-critical CO_2 refrigeration (-45 to -25^oC) and positive temperature by *GAX* chiller (5-10^oC). The saturated vapor was compressed in the trans-critical cycle and then cooled in the desorber of *GAX* cycle. This vapor is then cooled in the evaporator of the Rankine cycle and in the gas cooler of supercritical the CO_2 power cycle. The summary of the studies conducted on multi-evaporator cooling systems are shown in Table 3.12.

Table 3.12 – Summary of the studies conducted on multi-evaporator cooling systems

		Obtained results		
Work (type of study)	**System description**	**COP, Efficiency (η) and/or Exergy efficiency (ϵ)**	**Operating conditions**	**Comparison with conventional system**
[246] (S)	System consisting of one VCC (360KW), one ABSC (60KW), one LD (280 KW) and Cogenerators	- η of cogeneration > 80% due to use of both the power and exhaust heat in the hybrid cooling system	Thermal and desiccant storage tanks $\rightarrow$ operating hours of cogeneration system extended	-CO_2 emission $\downarrow$ (40%) compared to VCC -payback of 2 years compared to VCC -Size of VCC $\downarrow$ from 820KW to 360 KW -Size of LD $\downarrow$ from 600KW to 280 KW
[247] (S)	System consisting of one VCC , two ADSC (10 KW) and one LD		-(Q_l/Q_c) =0.56$\rightarrow$ The hybrid system can operate without ADSC	-(Q_l= 30%) $\rightarrow$ (COP $\uparrow$) (> 44%) compared to VCC -(Q_l= 42%) $\rightarrow$ (COP $\uparrow$) (> 70%) compared to VCC
[248] (S)	-Ejector-expansion trans-critical cascade CO_2 cycle with an ABSC -Three types of ABSC : (1)single-effect or (2)double-effect series-flow or (3) double-effect parallel-flow cycle.	- Higher ϵ using ABSC(1) -Using ABSC(1) : (Q_{VCC}/Q_{ABSC}) $\uparrow$ from 1 to 6 $\rightarrow$ (COP $\uparrow$) (36.32%) -$T_{gn,op}$ about 148°C : (COP Using ABSC(3)) > (COP Using ABSC(2))	EJC the most irreversibility source	
[249] (S)	-Combined heat and power system : Rankine cycle combined to cooling system -Cooling system consisted of VCC, ABSC and EJC	-Low ϵ due to low T_{ev} in VCC -$\eta about 20\%$ -($T_{ev,EJC}$ $\uparrow$) and/or ($P_{cp,o}$ $\uparrow$) $\rightarrow$ (η $\uparrow$) and (ϵ $\uparrow$) -(T_{wt} $\uparrow$) $\rightarrow$ (η $\downarrow$)and (ϵ $\downarrow$)		

[250] (S)	- Combined heat and power ; - Cooling by ejector-expansion trans-critical VCC and GAX ABSC	- (Q_{VCC}/Q_{ABSC}) $\uparrow$ $\rightarrow$ (COP$\uparrow$) and ($\eta\uparrow$) - ($T_{ev,GAX}\uparrow$)or ($T_{ev,VCC}$) $\rightarrow$ ($COP_{max}\uparrow$) - Ejector and compressor have greater defect efficiency	- Convenient temperature range for transcritical CO_2 evaporator : ((-25 °C)-(40°C)) - Convenient temperature range for GAX evaporator : (5-10 °C)	

3.6 Discussions

Individual cooling systems are widely used. However, each machine has its own advantages and disadvantages. For instance, the *VCC* has the greatest *COP* ; however, the high energy required procures high operation costs and CO_2 emissions. The *ABSC* has a greater coefficient of performance compared to other thermally driven cooling systems, but it requires a high heat source temperature. In order to mitigate the drawbacks encountered by individual cooling machines, different hybrid cooling systems have been previously proposed and investigated. Based on the study presented in this work, main observations on hybrid cooling systems are drawn as follows :

- Concerning hybrid systems based on *VCC*.
  - Generally, hybrid systems based on *VCC* offer a great energy saving, and *COP* improvement over *VCC* standing alone. Moreover, hybrid systems allow the use of renewable energy when a *VCC* is hybridized with another thermal cooling process/machine.
  - In cascaded absorption-vapor compression cooling system, the *ABSC* is used either to sub-cool the liquid leaving the condenser, or to intercool the fluid between two compressors in a two-stage *VCC*. The greatest energy saving is achieved in the system with intercooler.
  - A cascaded absorption-vapor compression system using a solar-assisted absorption cycle is not beneficial at low solar radiation due to the high generator temperature required.
  - The adsorption bed integrated in a *VCC* produces either high or low compression level. The best energy saving is achieved at high temperature difference between evaporator and condenser, with a similar *COP* in both configurations.

- In a hybrid desiccant-vapor compression cooling system, the thermal comfort is achieved by smaller *VCC*. This system is beneficial for power saving rather than energy saving. From an economic standpoint, the hybrid desiccant-vapor compression cooling system is suitable for high humidity climates and countries with high energy price. The use of a desiccant system with multi-stage adsorption and regeneration process reduces the regeneration temperature to 40-50^oC. Thus, the desiccant material could be regenerated by the condenser rejected heat in the *VCC*. Therefore, the needed auxiliary thermal energy is reduced.
- In a combined ejector-vapor compression cooling system, the performance of the system is highly dependent on the ejector geometry and working fluids. The ejector replacing the throttling valve in a *VCC* reduces the compressor size and improves the *COP* of the system especially at high ambient temperature.
- The ejector sub-cycle of cascaded ejector-vapor compression cooling systems could be driven by solar energy. The hybrid system works with higher capacity than *VCC* alone, higher efficiency and lower collector area than *EJC*. The required collector area in this system is somewhat high. The *EJC* standing alone performed well than the hybrid system in regions where cooling is required for a short time of the year.
- The *EC* used to pre-cool the ambient air before it passes through the condenser achieves a great energy saving compared to an air cooled *VCC*, especially in very hot climates where ambient temperature exceeds 50^oC. A higher *COP* and cooling capacity are also obtained due to lower condenser temperatures. This hybrid system is more beneficial at low relative humidity.

- Concerning hybrid systems based on *ABSC* :
  - The *ABSC* could produce a lower evaporator temperature at low generator temperatures with an integrated compressor in series at low pressure side. A generator temperature decreased by more than 20^oC is achieved compared to conventional *ABSC*. The parallel configuration of an absorption cooling system with an integrated compressor is not advantageous since the increase in cooling capacity obtained is not encouraging in accordance with the mechanical compression work required.
  - The desiccant-absorption cooling system is suitable for cascade utilization of thermal energy in humid climate with high solar energy.
  - The ejector-absorption cooling system achieves *COP* improvement of 20%-49% [218] with an increased cooling capacity, lower generator temperature and lower evaporator temperature compared to *ABSC*.

The generator temperature is decreased by more than 9^oC.

- Concerning hybrid systems based on adsorption cooling machine :
  - The adsorption cooling process is rarely considered as the main process in a hybrid cooling system due to the poor performance and the intermittence of the *ADSC*.
  - The use of a compressor to draw the fluid between adsorbent beds improves the performance of *ADSC*. However, the *COP* is still lower than *VCC*.
  - An ejector-adsorption cooling system improves the *COP* and overcomes the intermittence of both cycles.
  - It is found that hybrid systems based on *ADSC* are only beneficial for low cooling loads and for cases where intermittent cooling is acceptable, such as food storage, automobile air conditioning, etc.

- Concerning the desiccant-evaporative cooling system :
  - The use of *REC* could produce chilled water temperature in the range 18-20^oC [235] .
  - The regeneration temperature in the *DSC* is decreased using multi-stage dehumidification.
  - The desiccant-evaporative cooling system is recommended for hot and humid climate where relative humidity is greater than 60$\%$ and ambient temperature does not exceed 35^oC.

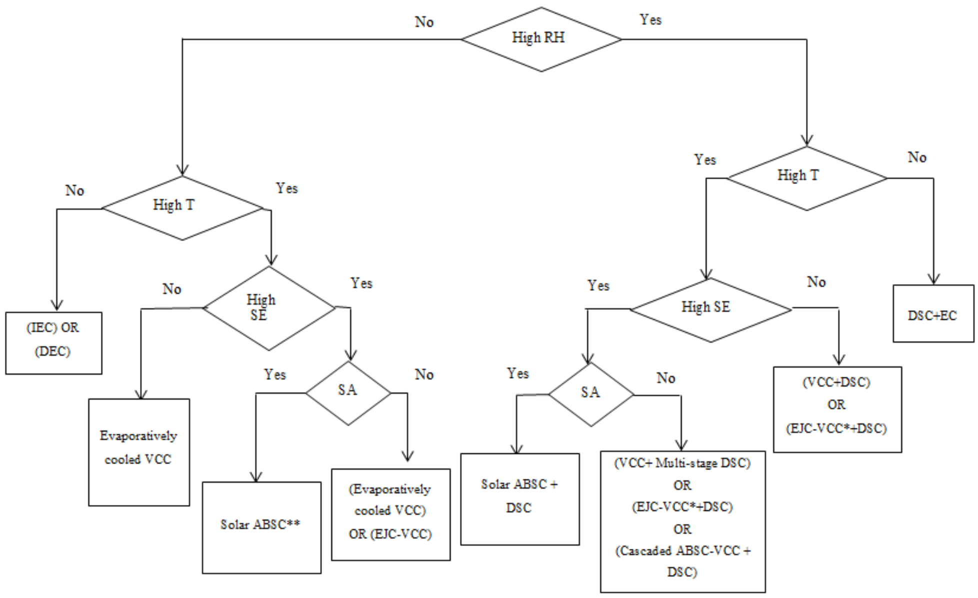
- The main advantage of the multi-evaporator cooling system is to eliminate the use of electric energy for cooling production during a period of the day especially at peak load time. Also, this hybrid system offers the benefits of simpler design compared to other hybrid systems, producing cooling at different temperature levels and better performance.

Based on the above-mentioned observations, it is found that different parameters must be considered in selecting a cooling system :

- Dry bulb temperature or sensible cooling load which is related to the size and the most appropriate type of cooling machine ;
- Relative humidity or latent cooling load which can be removed by a desiccant dehumidifier and reduces the size of cooling machine removing sensible load. The comfort level of relative humidity in habitable space is recommended generally in the range 30-60$\%$;
- Solar energy availability, which is related to the possibility of using thermally driven cooling machine which profits of the free renewable energy ;

- Surface availability, which makes possible the use of more than cooling machine and the installation of renewable energy systems such as solar collector, geothermal heat exchanger, etc.

A scheme for the selection of the best hybrid cooling system from energetic and environmental point of view is suggested in fig. 3.11. It is found that in some cases, individual cooling systems are more effective than hybrid systems. For instance, in very hot and arid climate, the removal of latent load separately by desiccant system becomes unnecessary, and thus, the comfort level could be reached using an electric or absorption chiller standing alone. At low ambient temperatures, an evaporative cooler is able to ensure comfort level. For regions characterized by high solar energy, the required energy for the generator of absorption chiller can be totally provided by a solar system. Therefore, a standing alone absorption chiller could be used.



⋆ Ejector as throttling valve or Ejector on the condenser

⋆⋆ The condenser of the absorption chiller could be evaporatively cooled

Figure 3.11 – Scheme for cooling system selection

3.7 Conclusions

The hybridization of cooling systems has proved to be an energy saving method compared to standalone machines. This chapter provides a detailed review of all types and research developments concerning hybrid cooling systems for residential use. A wide variety of existing studies performed on hybrid cooling systems has been investigated in order to emphasize the advantages and the benefits of the combination of more than processes or machines to cover the cooling need. Based on the review study, it was observed that most of hybrid cooling systems were able to achieve a *COP* improvement, an energy saving, a great reduction of primary energy consumption and CO_2 emissions, and also, it succeeds in the mitigation of individual cooling systems' disadvantages. These improvements vary in a wide range according to different parameters (e.g. climatic conditions, energy prices, availability of renewable energy sources, etc.). However, a hybrid cooling system may have a negative impact if it was not carefully selected. For instance, an ejector standing alone performs well than cascaded ejector-vapor compression cooling systems in regions with low cooling load during a short period of the year.

Despite the aforementioned advantages, hybrid cooling systems are not sufficiently integrated in the market, and the application of such system is still limited. This is due to the high initial and maintenance costs owing to the complexity of design which makes the use of hybrid system not appealing, especially for small applications. Thus, further studies should be more concerned about the development of simpler system design in order to encourage the use of hybrid cooling systems even for residential application.

In the next chapter, a hybrid cooling system is designed based on the review study performed in the present chapter. An optimal sizing method is presented in order to determine the optimal number of the components in the system. The problem of optimization is studied for different case studies.

4.Hybrid Cooling System : Optimal Sizing

Nomenclature

Abbreviations

AC	*Absorption Chiller*
CHW	*Chilled Water tank*
COP	*Coefficient of performance*
DR	*Discount Rate*
DSC	*Desiccant System*
ET	*Evacuated Tube*
ET,AC	*Evacuated tube solar collector for Absorption chiller*
ET,DSC	*Evacuated tube solar collector for Desiccant system*
HW	*Hot Water tank*
HX	*Heat Exchanger*
Hyb	*Hybrid*
NPV	*Net Present Value*
PV	*Photovoltaic*
VCC	*Vapor Compression Chiller*
WT	*Wind Turbine*

Variables

A	*Area* $[m^2]$
C	*Cost* $[\$]$
COP	*Coefficient of Performance* $[-]$
E	*Energy* $[KJ]$
g-value	*Solar Heat Gain Coefficient* $[\%]$
N	*Number of Units/years* $[Units/years]$
RH	*Relative Humidity* $[\%]$
T	*Temperature* $[^oC\ or\ K]$
U-value	*Heat Transfer Coefficient* $[W/m^2.K]$

Subscripts

a	*Ambient*
chw	*Chilled water*
cv	*Conventional*

e	*Electric*
gen	*Generator*
i	*Input*
l	*Latent*
l	*Load*
lt	*Lifetime*
max	*Maximum*
min	*Minimum*
o	*Output*
reg	*Regeneration*
s	*Sensible*
sp	*Set point*
st	*Storage*
t	*Thermal*
u	*Unit*
y	*Years*

4.1 Introduction

In order to overcome the harmful impact and the high energy consumption of cooling systems, numerous systems were designed and studied in the literature. Hybrid cooling technology has been developed for high efficiency and energy saving cooling systems. A wide range of hybrid cooling systems has been investigated previously. Most of them concentrated on the hybridization of absorption and mechanical compression processes to produce cooling effect. This type of hybrid system achieved a great energy saving and coefficient of performance COP improvement by more than 50$\%$ ([213] [151] [149]). Furthermore, multi-evaporator hybrid cooling systems, where cooling is produced by more than one cooling machine, have been proved to be energy efficient and economically feasible. For instance, in [251], the authors studied a hybrid system consisted of cogenerates, a liquid desiccant system, a vapor compression chiller, an absorption chiller and a gas boiler. The results for a building located in Beijing, China showed a payback period of 2 years.

Besides, solar cooling is one of the energy saving cooling technologies used all over the world ([8]and[7]). For instance, in Mediterranean areas, solar cooling systems can save about half of the primary energy consumption ([7]and [123]). In addition, a solar cooling machine reduces the dependency on electrical energy, which makes this technology suitable for countries suffering from shortage in electricity generation. Generally, solar cooling can be achieved either by solar thermal systems where the cooling is provided by a sorption machine thanks to a high temperature fluid heated by the solar energy, or solar electric systems where an electric chiller is driven by solar energy converted into electrical energy using photovoltaic panels. Most solar thermal cooling systems are based on LiBr/H2O absorption chiller, a proven technology for domestic use ([252]and [253]), and they are highly efficient in areas with high solar energy and sunny hours available. According to [252] and [253], in solar absorption cooling, the auxiliary cooling system, such as heat pump or air-cooled chiller, is preferred over auxiliary heating systems, such as gas boiler, electric heater, etc, when solar energy is not able to provide all cooling load. Rendering to [6], the capital investment of solar electric cooling system is expected to be the lowest in 2030 with lower projected CO_2 emission values. Another energy saving method is the separate sensible and latent cooling, where sensible load is met by chiller (e.g. absorption chiller, vapor compression chiller, etc.) ; while a desiccant system is used to meet the latent load. This method provides an efficient control of air humidity and temperature in parallel and the comfort level could be reached by a higher supply chilled water temperature (the chilled water temperature can be raised from 7^oC to 13^oC [6]. Consequently, it improves COP and reduces the energy consumption as well as the size of the cooling chiller ([155]and [144]). Moreover, desiccant systems have the benefit of being driven by low-grade heat (about 70 – 80^oC), such as solar free energy, industrial waste heat, etc. They are usually hybridized with cooling systems such as vapor compression chiller, absorption chiller, adsorption chiller or evaporative cooler. The use of the hybrid desiccant cooling system becomes more beneficial in zones with high latent load fraction([74] [247] [254][162] [159]) where it can produce a COP improvement and an energy saving of more than 30$\%$ compared with a standalone chiller ([157] [220] [160] [213].

The objective of this chapter is to present an optimal sizing method that defines, in a specific region, a hybrid cooling energy system, economically feasible with maximum renewable energy share. The method tends to minimize the dependency of the cooling system on grid electricity and to reduce the nonrenewable energy consumption. Based on the above, an optimal sizing method for hybrid cooling system is proposed and studied in this chapter.

The present chapter is organized as follow. In section 2, a hybrid cooling system for building use is proposed and designed. In the third section, a standard small house is defined in order to ensure the application of the hybrid system. In section four, an optimal sizing method is proposed where an objective function, under a number of constraints, is defined and presented. Then, The optimization problem is investigated for different case studies, where different geographical regions are chosen to assess the proposed method according to diverse climatic conditions, component prices and electricity costs. Finally, the results obtained are compared and main conclusions are deduced.

4.2 System description

The main objective of this study is to investigate a proposed optimal sizing method used for hybrid energy systems. For this purpose, a residential hybrid cooling system that benefits from renewable energy resources is designed and modeled using Trnsys 17. The system, schematized in fig.4.1, mainly consists of two types of components ; i) thermal energy components composed of a solar absorption chiller, a vapor compression chiller, a solid desiccant system, two evacuated tube solar thermal collectors, two hot water storage tanks and a chilled water storage tank and ii) electric energy components composed of a wind turbine, a PV cell and two electric storage batteries.

This designed cooling system is used to ensure the cooling need of a standard small house. The total cooling load is separated into its two components ; the sensible load that is related to the dry bulb temperature and the latent load that is related to the wet bulb temperature or the relative humidity. Each load is treated separately by using separate components of the hybrid system. The desiccant dehumidifier component is used to meet the latent load, while chiller components are utilized to meet the sensible load and excess latent load, if found. This strategy of separating both kinds of loads has been proved to be an efficient energy-saving method since it raises the evaporator temperature in the sensible cooling machine ([255] and [256]).

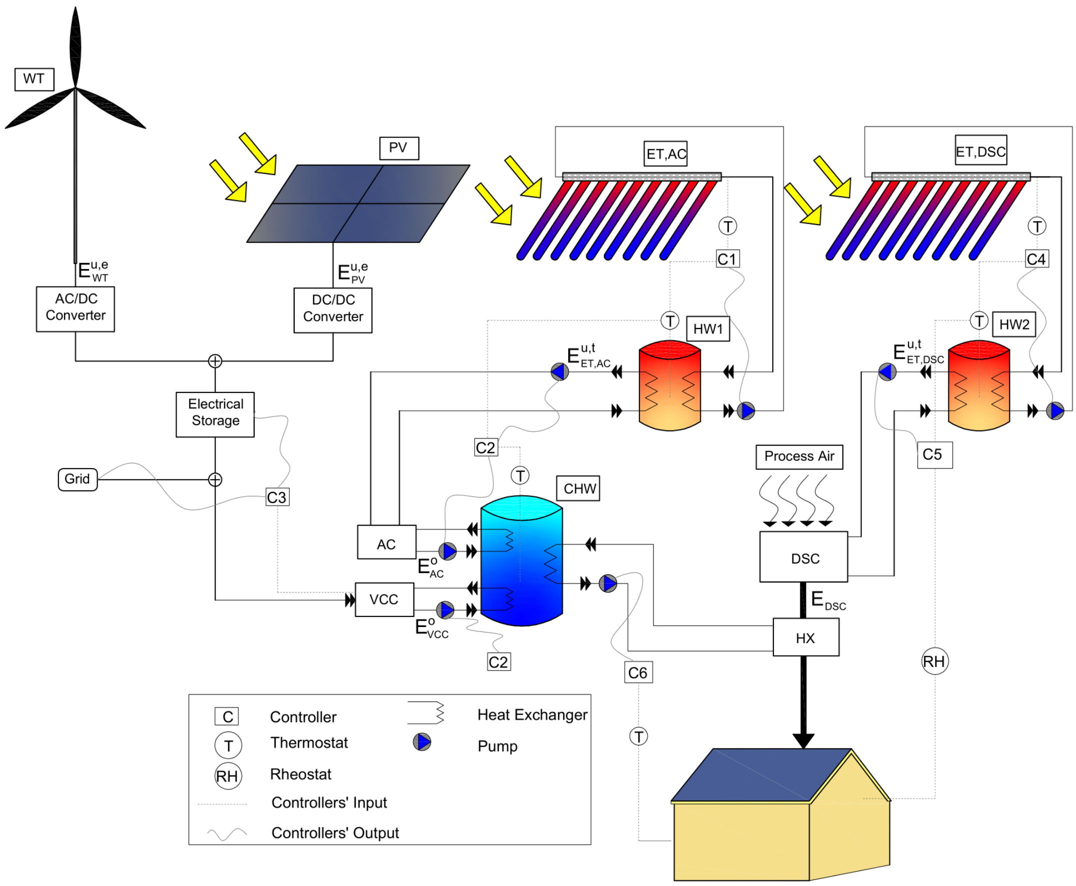


Figure 4.1 – schematic representation of the proposed residential hybrid cooling system

The desiccant wheel, driven by a solar thermal collector, is utilized because it is an effective method of meeting the latent cooling load. In addition, a solid type desiccant is used because it is preferred to a liquid type one, especially for residential applications, due to the simplicity of its structure and its common regeneration by a thermal solar energy source [257]. Besides, the solar cooling absorption chiller is chosen to meet the sensible load because it is highly effective and clean, due to the convergence of peak cooling loads with the available solar radiation as well as the use of environmentally friendly refrigerants. In addition, an electric vapor compression chiller is utilized as a backup for the absorption machine since it has high coefficient of performance. Moreover, auxiliary cooling systems are preferred to auxiliary heating ones, such as gas boilers or electric heaters, which are used to provide the necessary thermal energy for solar absorption chillers as a backup for the solar thermal system ([252] and [54]). Furthermore, both thermal solar cooling systems, the desiccant wheel and the absorption chiller, could be effectively driven by flat plate or evacuated tube solar collectors ([252] [258] [259] [260]). However, in this study, evacuated tube solar collectors are chosen for the application because they require smaller roof area and are more efficient when high temperature difference between the working fluid and ambient temperature exists ([261] and [262]) as well as in locations subjected to abrupt variations in solar insulation. According to [263], evacuated-tube collectors are more likely to maintain their efficiency over a wide range of ambient temperatures and heating requirements. In the present system, each evacuated tube collector is connected to one of the two mentioned cooling systems via a hot water storage tank. The total separation of both solar thermal systems is considered to allow independent control and operation, especially that each one has different operating conditions.

The electric vapor compression chiller is another utilized cooling energy system, which operates using electrical energy. Hence, in order to minimize the grid electricity consumption and consequently approach low energy and low carbon building concepts, a hybrid wind turbine/solar photovoltaic system is utilized to generate part of the electricity needed by the vapor compression chiller.

Moreover, a chilled water tank is integrated to the hybrid cooling system in order to improve the period of solar absorption cooling as well as to ensure the operation stability of vapor compression chiller by reducing the number of start-up operations. Both chillers are connected in parallel to the chilled water tank.

Concerning the control method of the system, the absorption chiller is initiated when the temperature in the chilled water tank is greater than the chilled water temperature set point (T^{sp}_{chw}) and the temperature of water in the hot water storage *HW1* (refer to fig. 4.1) is greater than the minimum temperature required by the generator of the absorption chiller (T^{min}_{gen}) to operate. If the absorption chiller is not able to meet the chilled water need, then the electric vapor compression chiller starts operating, where its electricity need is secured from the electrical energy renewable energy systems, PV cells, wind turbines and batteries, if stuffiest. Else, grid electricity is utilized. As for the desiccant system, it starts operation when the relative humidity in the house is greater than RH^{sp}_{h} and the desiccant material is regenerated by solar thermal energy stored in the hot water tank water *HW2* (refer to fig. 4.1) when the temperature of water exceeds the threshold of the desiccant wheel regeneration (T_{reg}^{min}). Besides, process air treated in the desiccant system consists of 80$\%$ return air from the house mixed with 20$\%$ fresh air from the ambient for health and comfort purposes. After being dehumidified in the desiccant wheel, the process air is cooled by the chilled water through a water-to-air heat exchanger (*HX*) before passing into the house.

4.3 House Description

A non-thermally isolated house of $144\ m^2$ total surface area, North oriented and connected to the electrical grid, is considered in this study. It is composed of two levels ; the basement is divided into two thermal zones with a total area of $72\ m^2$, while the upper level is totally considered as one thermal zone, having the same surface area as beneath. Two types of walls are adopted, external walls with U-value of $0.25\ W/m^2K$ and internal walls with U-value of $0.126\ W/m^2K$. The floor of the basement satisfies Dirichlet boundary condition with a constant Temperature of 0^oC. The ceiling of the basement is itself the floor of the first level acquiring a U-value of 0.896 W/m^2K and stills the ceiling of the upper level, which is considered to satisfy Neumann boundary condition with a U-value of $0.148\ W/m^2K$.

As for the windows, they constitute 9% of the total area of external walls with an infiltration rate of about $0.1h^{-1}$, all being of the same type with a U-value of $1.06\ W/m^2K$. In addition, the heat gains due to occupants, electric appliances and other equipment are neglected for the sake of simplicity.

The dry bulb temperature and the relative humidity of each thermal zone are intended to be maintained close to pre-specified set points, using the proposed hybrid cooling system. Annual cooling energy load of the house (E_L) is separated into annual sensible (E_s) and annual latent (E_l) cooling loads, so that each could be treated alone using the appropriate thermal cooling system, as discussed before.

4.4 Optimal Sizing Method

The core objective of this chapter is to find an optimal sizing method applicable to hybrid energy systems in the aim of defining the best system configuration from energetic and economic standpoints. Accordingly, a highly efficient hybrid cooling energy system that benefits from renewable energy resources is designed and illustrated in section 4.2 as well as it is applied to a suggested house model described in the previous section. Hence, for this hybrid system, the energetic objective aims to minimize the total grid-electrical energy consumption and consequently maximize the renewable energy share, and the economic objective aims to obtain a positive net present value (NPV) for the proposed system, which is also greater than that of conventional electric vapor compression chiller over a pre-specified period of time.

The optimal sizing method is based on the concept of simulating the proposed hybrid cooling system using small base-units for the renewable energy components, solar and wind systems : PV cell, wind turbine, evacuated tube solar collector connected to hot water storage tank for both absorption chiller and desiccant system. Note that, the evacuated tube solar collector connected to hot water storage tank is considered as one unit. These base units are supposed to produce their maximum possible energy during the studied period of time. Thus, a multiple of a base unit would multiply its maximum energy output for the same climatic and boundary conditions. Results of the sizing problem should then find the appropriate number of individual renewable base units. This method has been first proposed in [264] however, in this study, another application is investigated and another formulation and problem solution are presented.

4.4.1 Mathematical formulation

The goal of this subsection is to give the mathematical formulation of the sizing problem. More precisely, I propose a formulation for choosing the optimal configuration for the hybrid system. The optimal configuration is obtained taking into account energy balance, the available surface and the economic constraints.

4.4.1.1 Statement of optimization problem

A.Objective function In this section, an objective function that maximizes the renewable energy share and consequently, minimizes the grid electricity consumption is defined. It is determined according to the following demonstration :

The energy output (E^o)of an energy system is related to its energy input (E^i) and its COP as shown in equation 4.1.

$$COP = \frac{E^o}{E^i} \tag{4.1}$$

Hence, relations between input and output energies of each of the absorption chiller (E^i_{AC} and E^o_{AC}), the vapor compression chiller (E^i_{VCC} and E^o_{VCC})and the desiccant wheel could be deduced as follows :

$$E^o_{AC} = COP_{AC}E^i_{AC} = COP_{AC}N^{AC}_{ET}E^{u,t}_{ET,AC} \tag{4.2}$$

$$E^o_{VCC} = COP_{VCC}E^i_{VCC} = COP_{VCC}(N_{PV}E^{u,e}_{PV} + N_{WT}E^{u,e}_{WT} + E_{grid}) \tag{4.3}$$

$$E^o_{DSC} = COP_{DSC}E^i_{DSC} = COP_{DSC}N^{DSC}_{ET}E^{u,t}_{ET,DSC} \tag{4.4}$$

Where :

- N_{PV}, N_{WT}, N^{AC}_{ET} and N^{DSC}_{ET} represent the number of photovoltaic cells, wind turbines, and evacuated tube solar collectors for the absorption chiller as well as for the desiccant system, respectively.
- $E^{u,e}_{PV}$ and $E^{u,e}_{WT}$ are the electrical energy produced by one base-unit of each of the PV and wind turbine systems, respectively.
- $N^{u,t}_{ET,AC}$ and $N^{u,t}_{ET,AC}$ are the thermal energy produced by one base-unit of each of the evacuated tube collectors used for feeding the absorption chiller and the desiccant system, respectively.
- COP_{AC}, COP_{DSC} and COP_{VCC} are the respectively the coefficient of performance of each of the absorption chiller, desiccant wheel and vapor compression chiller.

Knowing that one of the principal and obvious constraints is the ability of the hybrid system to meet the total cooling load of the house, it could be concluded that the sum of energy produced by the absorption chiller, the vapor compression chiller and the desiccant system should be at least equal to the cooling load of the house. This constraint is presented in equation 4.5.

$$E^o_{AC} + E^o_{VCC} + E^o_{DSC} \geq E_L \tag{4.5}$$

Using equations 4.2 to 4.5, the following equation is obtained :

$$COP_{VCC}(N_{PV}E^{u,e}_{PV} + N_{WT}E^{u,e}_{WT} + E_{grid}) + COP_{AC}N^{AC}_{ET}E^{u,t}_{ET,AC} +$$

$$COP_{DSC}N^{DSC}_{ET}E^{u,t}_{ET,DSC} \geq E_L \tag{4.6}$$

Hence, equation 4.6 could be reformulated to give equation 4.7 :

$$E_{grid} \geq E_L/COP_{VCC} - (COP_{AC}/COP_{VCC}N_{ET}^{AC}E_{ET,AC}^{u,t}+$$

$$COP_{DSC}/COP_{VCC}N_{ET}^{DSC}E_{ET,DSC}^{u,t} + N_{PV}E_{PV}^{u,e} + N_{WT}E_{WT}^{u,e}) \quad (4.7)$$

The second term of the right part of equation 4.7 represents the total renewable energy share of the hybrid system and maximizing this term will minimize the grid electricity consumption as it is obvious from this equation. Therefore, the objective function stated in equation 4.8 is the maximum of this term of equation 4.7.

$$\max f = \frac{COP_{AC}}{COP_{VCC}} N_{ET}^{AC} \; E_{ET,AC}^{u,t} + \frac{COP_{DSC}}{COP_{VCC}} N_{ET}^{DSC} \; E_{ET,DSC}^{u,t}+$$

$$N_{PV}E_{PV}^{u,e} + N_{WT}E_{WT}^{u,e} = \left\langle \hat{N},\ \beta \right\rangle \quad (4.8)$$

With

$$\hat{N} = \begin{pmatrix} N_{PV} \\ N_{WT} \\ N_{ET}^{AC} \\ N_{ET}^{DSC} \end{pmatrix}$$

And

$$\beta = \begin{pmatrix} E_{PV}^{u,e} \\ E_{WT}^{u,e} \\ \frac{COP_{AC}}{COP_{VCC}} \; E_{ET,AC}^{u,t} \\ \frac{COP_{DSC}}{COP_{VCC}} \; E_{ET,DSC}^{u,t} \end{pmatrix}$$

B.Constraints The energy balance constraints of the system are the following :

(C1) The hybrid system meets the total cooling load of the house. This constraint has been previously illustrated in equation 4.5.

(C2) The electricity consumed by the vapor compression chiller is at most equal to the electrical energy produced by the PV and WT systems in addition to the consumed grid electricity. This constraint is presented in equation 4.9.

$$N_{PV}E_{PV}^{u,e} + N_{WT}E_{WT}^{u,e} + E_{grid} = E_{VCC}^{i} \tag{4.9}$$

(C3) Equations 4.10 and 4.11 illustrate the thermal energy balances between the output of each evacuated tube solar system with the input of the corresponding thermal cooling system, i. e. the absorption chiller and the desiccant wheel.

$$N_{ET}^{AC}E_{ET,AC}^{u,t} = E_{AC}^{i} \tag{4.10}$$

$$N_{ET}^{DSC}E_{ET,DSC}^{u,t} = E_{DSC}^{i} \tag{4.11}$$

(C4) The maximum cooling energy delivered by the desiccant wheel should not exceed the total latent cooling load as formulated in the following equation :

$$\langle \hat{N}, \rho \rangle \leq E_1 \tag{4.12}$$

Where

$$\rho = \begin{pmatrix} 0 \\ 0 \\ 0 \\ COP_{DSC}E_{ET,DSC}^{u,t} \end{pmatrix}$$

Furthermore, given that the hybrid cooling system is applied to a residential with limited roof surface area, it is necessary to impose area constraints related to the number of renewable energy systems installed on the roof. Equation 4.13 represents this constraint as follows :

$$\left\langle \hat{N} \ , \gamma \right\rangle \ \leq \ A_T \tag{4.13}$$

Where $\gamma = \begin{pmatrix} A_{PV} \\ A_{WT} \\ A_{ET}^{AC} \\ A_{ET}^{DSC} \end{pmatrix}$; and A_T is the useful total surface area of the house roof.

Additionally, from an economic point of view, any system is said to be economically feasible if it has a positive net present value (NPV) over a specified period of time. Moreover, when comparing two systems, the one with greater NPV is the preferred. Thus, in the case considered in this study, the proposed hybrid cooling system economically compared to a conventional electric vapor compression chiller, this constraint could be translated into the following equation :

$$NPV_{VCC}^{cv} - NPV_{hyb} \leq 0 \tag{4.14}$$

Where NPV_{VCC}^{cv} *and* NPV_{hyb} are the net present values of the conventional vapor compression chiller and the hybrid system, respectively.

Since there is no energy savings considered for the conventional system, then its net present value is simply its capital cost during the studied period of time. Hence, equation 4.14 could be written as follows :

$$\left[\left(\sum_{X\in H1} C_X^{hyb} + \sum_{X\in H2} C_X^{hyb}\right) - C_{VCC}^{cv}\right] - \left[\alpha\left(N_y\right)C^e\left(E_{grid}^{Cv} - \left(\frac{E_L}{COP_{VCC}} - \left\langle \hat{N},\ \beta\right\rangle\right)\right)\right] \leq 0 \tag{4.15}$$

Where

- $H1 = \{PV; WT; ET, AC; ET, DSC; st^e; st^{t1}; st^{t2}\}$
- $H2 = \{AC;\ VCC;\ DSC\}$
- C_X^{hyb} is the cost of one component of the hybrid system.

- C_{VCC}^{cv} is the cost of the conventional system.

- $\alpha\left(N_y\right)$ is defined as follow :

$$\alpha\left(N_y\right) = \; N_y \; DR^{-1}\left(1 - \; (1 + DR)^{-N_y}\right)$$

  with N_y being the number of operation years and DR being the discount rate of the considered case study.

- C^e is the electricity cost of the considered case study.

- E_{grid}^{Conv} is the grid electrical energy consumed by the conventional electric vapor compression chiller.

The cost of each component X of the hybrid system is calculated using equations 4.16 and 4.17, respectively.

$$C_X = \left(\left[\frac{N_y - 1}{N_X^{lt}}\right] + 1\right) C_X^{hyb} = \varphi_{N_y}\left(N_X^{lt}\right)\; C_X^{hyb} \qquad for \quad X \in H1 \tag{4.16}$$

Where N_X^{lt} is the life time of the component X and $\left[\frac{N_y-1}{N_X^{lt}}\right]$ denotes the integer part of $\frac{N_y-1}{N_X^{lt}}$.

In addition, the storage electrical and thermal systems are used jointly with the corresponding renewable energy system. Thus, the number of storage is null when the number of corresponding renewable system is zero. So,

$N_{st^e} = 0$ if $(N_{PV} + N_{WT}) = 0$, $\quad N_{st^{t1}} = 0$ if $N_{ET}^{AC} = 0$ and

$N_{st^{t2}} = 0$ if $N_{ET}^{DSC} = 0$.

The cost of cooling and de-humidification systems is calculated as follow :

$$C_X = \sigma\;(Z)\varphi_{N_y}\left(N_X^{lt}\right)\; C_X^{hyb} \qquad for \quad X \in H2 \tag{4.17}$$

Where $\quad \sigma\left(Z\right) = \begin{cases} 0 \;\; if \quad Z \leq 0 \\ \\ 1 \;\; if \quad Z > 0 \end{cases}$; with Z being defined as follows :

$$Z = \begin{cases} E_L - \left(COP_{AC}\ \ N_{ET}^{AC}\ \ E_{ET,AC}^{u,t} + \ COP_{DSC}\ \ N_{ET}^{DSC}\ \ \ E_{ET,DSC}^{u,t}\right) & if\ X = VCC \\ N_{ET}^{AC} & if\ X = AC \\ N_{ET}^{DSC} & if\ X = DSC \end{cases}$$

C.Optimization problem Following the defined method objectives and applying for the introduced hybrid cooling system, the objective function is defined as follows :

Finally, considering the objective function and constraints, the optimal sizing problem can be stated as follows :
the objective function

$$Max\ \left\langle \hat{N},\ \beta \right\rangle$$

$$\begin{cases} \left\langle\ \hat{N}\ \ ,\gamma \right\rangle \ \ \leq\ A_T \\ \\ \left\langle\ \hat{N}\ \ ,\rho \right\rangle \ \ \leq\ L \\ \\ \psi(\hat{N}) = \left[\left(\sum_{X\in H1} C_X^{hyb}\ +\ \sum_{X\in H2} C_X^{hyb}\ \right) -\ C_{VCC}^{cv}\right] - \\ \qquad\qquad \left[\alpha\left(N_y\right) C^e \left(E_{grid}^{Cv} - \left(\frac{E_L}{COP_{VCC}} - \left\langle \hat{N},\ \beta\right\rangle\right)\right)\right]\ \ \leq 0 \end{cases}$$

With $\hat{N} = \begin{pmatrix} N_{PV} \\ \\ N_{WT} \\ \\ N_{ET}^{AC} \\ \\ N_{ET}^{DSC} \end{pmatrix}$ and all N are positive integers

4.4.1.2 Problem resolution

The problem is stated as a discrete problem with a limited number of cases, namely number of renewable energy systems. Thus, it can be solved by using a straight forward algorithm using Matlab. The problem is solved taking into consideration the different possibilities -in regard the number of renewable energy

systems used- as shown in Algorithm 1.
+

4.4.2 Case studies

The introduced optimal sizing method applied for the proposed hybrid cooling system is investigated in two case studies. One represents Beirut city, the capital of Lebanon and the other represents Marseille, the second largest city in France. Both cities are Mediterranean with moderate climate characterized by mild rainy winters and hot humid summers.

In the following subsection, input data concerning the climate of a typical meteorological year and the corresponding cooling load of the considered house model as well as the cost of electricity and energy systems are illustrated for both case studies.

4.4.2.1 Input data

In this section, input climatic, load and economic data related to the presented case studies are illustrated. For the climatic data, a typical meteorological year data file obtained from "Meteonorm" software is used as an external file for the data weather component in the prepared Trnsys simulation model. Fig. 4.2 and 4.3, respectively, show the hourly variations of both ambient temperature and relative humidity during a typical year for both cases, Beirut and Marseille. It is clear that in both cases, high relative humidity exists all through the year, which indicates the existence of latent load year round. Thus, it is predicted that the integration of a de-humidification system , namely the desiccant system is important in both case studies. This could reduce greatly the energy consumption from electricity, since the desiccant system driven by free energy deals with the latent load that consists about half of the total load (refer to fig. 4.4). Concerning the ambient temperature, Marseilles is generally colder than Beirut. In winter season, the temperature in Marseilles reaches the zero degree level, while a minimum of approximately 5^oC is attained in Beirut. Similarly, in summer, Beirut attains higher temperatures. Furthermore, temperature and humidity profiles of Beirut are more stable than those of Marseille, which experiences abrupt changes year round.

The proposed hybrid cooling system is designed using small base units for the renewable energy components, PV, WT and ET energy systems, as introduced before. On the other hand, each of the absorption chiller, vapor compression chiller and desiccant systems is chosen so as it has acceptable capacity for the developed application. Main characteristics of the hybrid system components used

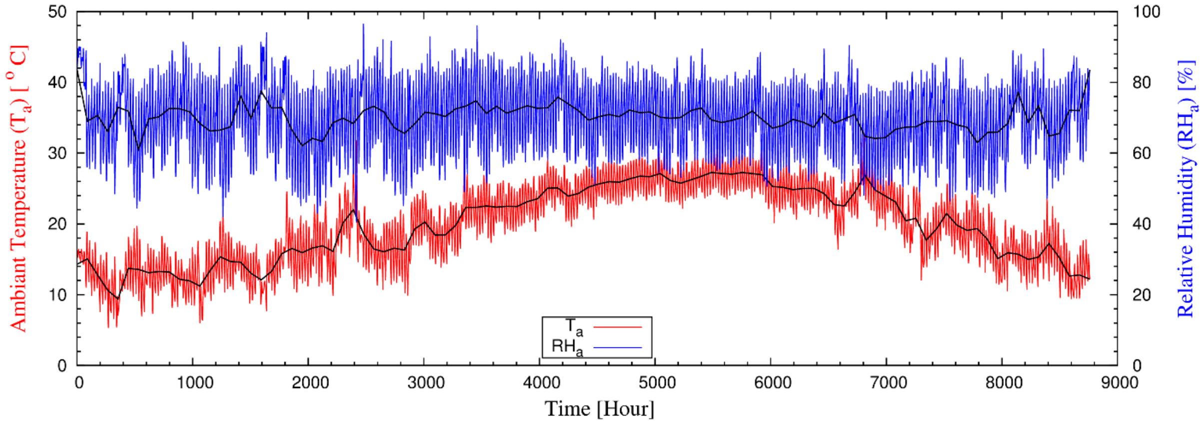


Figure 4.2 – Hourly variation of ambient temperature and relative humidity during a typical meteorological year of Beirut

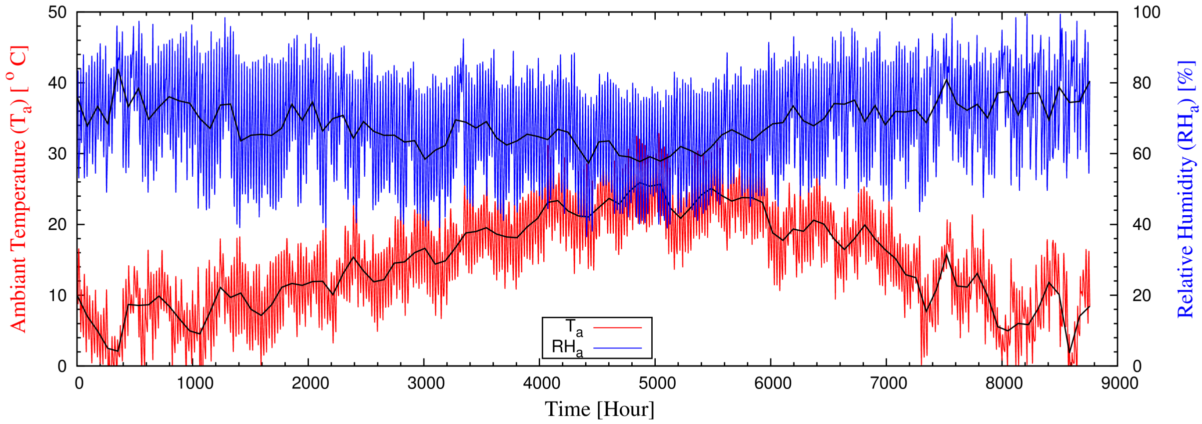


Figure 4.3 – Hourly variation of ambient temperature and relative humidity during a typical meteorological year of Marseille

in simulations are illustrated in table 4.1. The house described previously is modeled, and simulated for a typical meteorological year using the software Trnsys. As a result, the sensible and the latent load are obtained using the built-in models integrated in Trnsys. The results for Beirut and Marseille are presented in fig. 4.4 and 4.5, respectively. Generally, the sensible load and the latent load are closely related to the ambient temperature and relative humidity, respectively shown in fig. 4.2 and 4.3. Just as predicted, latent cooling loads exist year round in both cases due to high relative humidity, while sensible loads appear during and near the summer season. The total cooling load is obviously the sum of both sensible and latent loads.

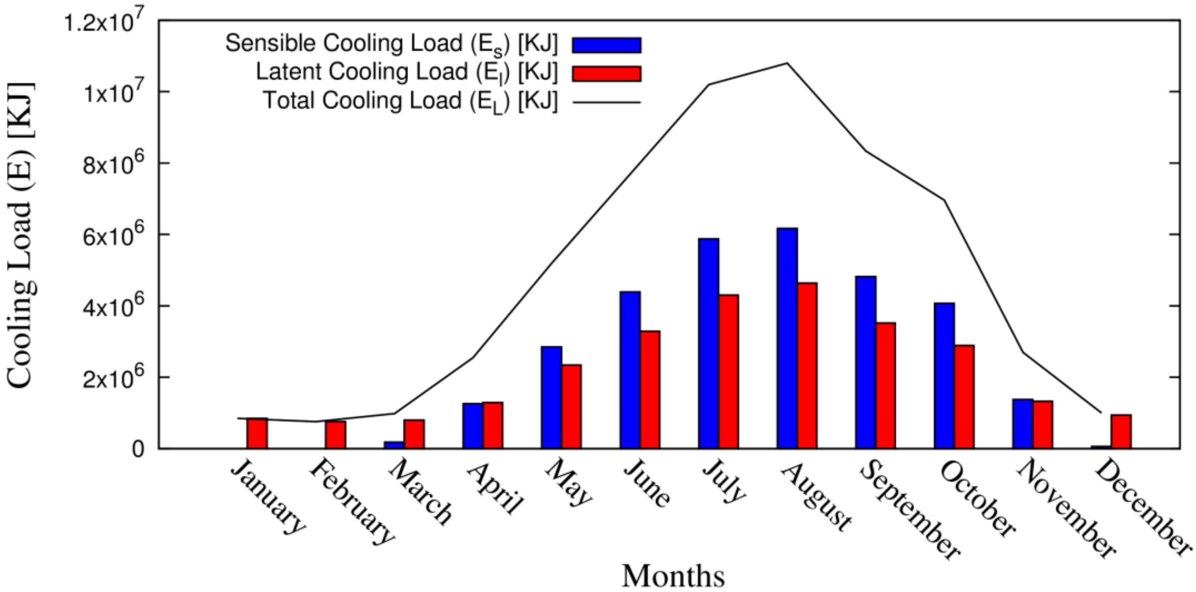


Figure 4.4 – Monthly values of sensible, latent and total cooling loads during a typical meteorological year of Beirut

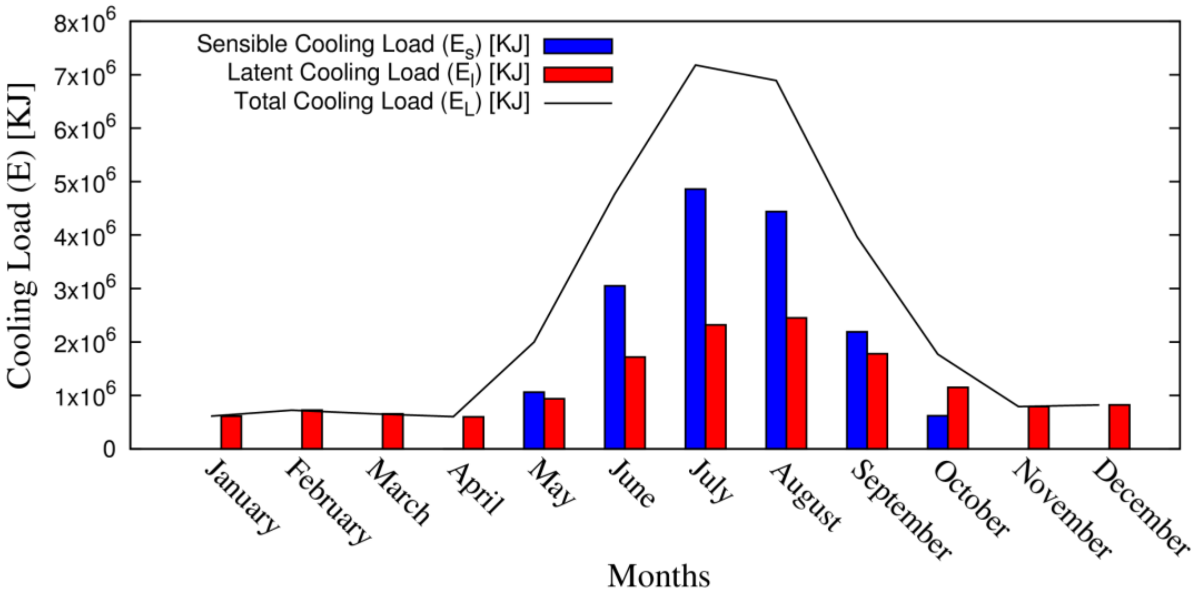


Figure 4.5 – Monthly values of sensible, latent and total cooling loads during a typical meteorological year of Marseille

As shown in the algorithm 1, the energy outputs from renewable energy systems is required to solve the optimization problem. Thus, the renewable energy systems are modeled and simulated using Trnsys. Consequently, the electric and the thermal output of these systems are obtained. The electric energy outputs of PV and WT base units for the two studied cases are presented for monthly values in fig. 4.6 and 4.7, respectively. It is shown that PV unit is much more efficient from an energetic point of view. This is mainly attributed to low wind speeds and high solar radiation in Beirut and Marseille. Concerning thermal energy, monthly values of the energy output in a typical meteorological year of one base unit of each of the adopted evacuated tube solar thermal systems, the one used for AC and that used for VCC, are illustrated for Beirut and Marseille in fig. 4.8 and 4.9, respectively. Thermal energy is low in the winter season and increases gradually as approaching the summer season, with the output from ET, DSC is greater from that of ET, AC, since the former's area of absorption is larger as illustrated in the system description.

The yearly values of the energy outputs are obtained by computing the sum of the monthly values. Noting that for cooling systems that remove the sensible load such as the absorption chiller and the vapor compression chiller, the values for the months where sensible load is null are not considered. For instance, in Beirut, the monthly energy outputs of PV, WT and ET,AC in January are not considered , since there is no sensible load in Beirut during January. So, both absorption and vapor compression chiller are turned off during January. However, in case of desiccant systems, the energy output of ET, DSC is token into consideration all through the year, since the latent load exists for all months.

Additionally, one constraint of the optimal sizing problem is economic, which needs the price and life-time of each of the various system components as inputs in the way for obtaining the solution. Accordingly, table 4.2 presents the market prices as well as the life time of hybrid cooling system components in Beirut and Marseilles.

It is worth noting that the climatic data are inputs for the developed Trnsys simulations, while the annual integrated values of different types of the cooling loads as well as the integrated useful energy outputs for renewable energy base-units all along with the life time and price of each component of the hybrid system as well as the total available surface area on roof are inputs for the presented optimal sizing problem.

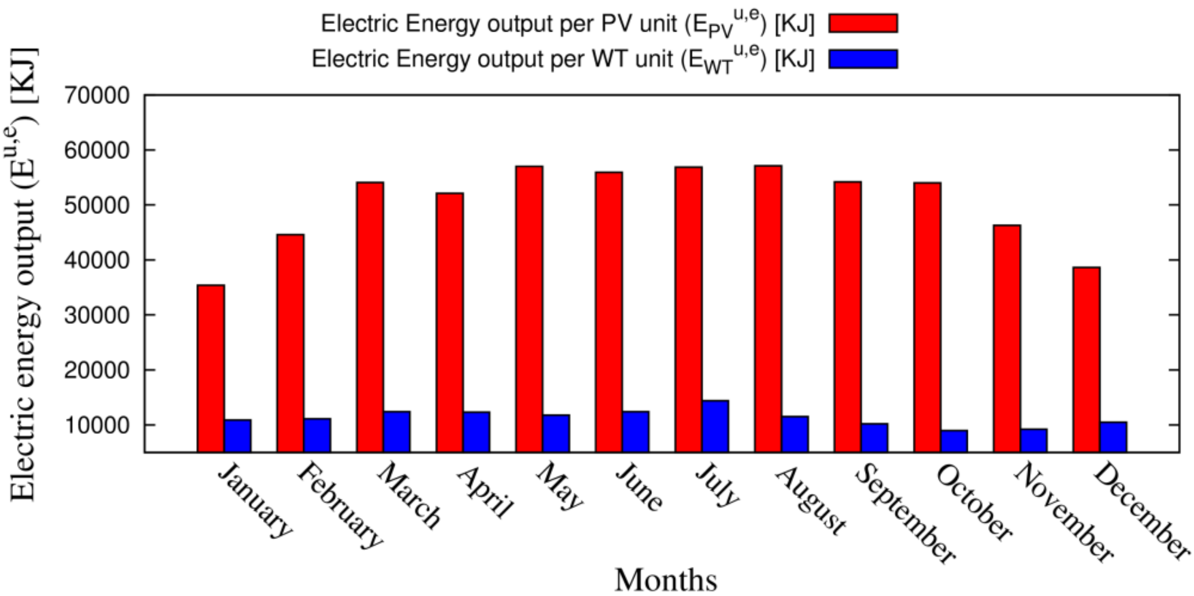


Figure 4.6 – 6 Monthly values of electrical energy output of a base unit of each of PV and WT systems during a typical meteorological year of Beirut

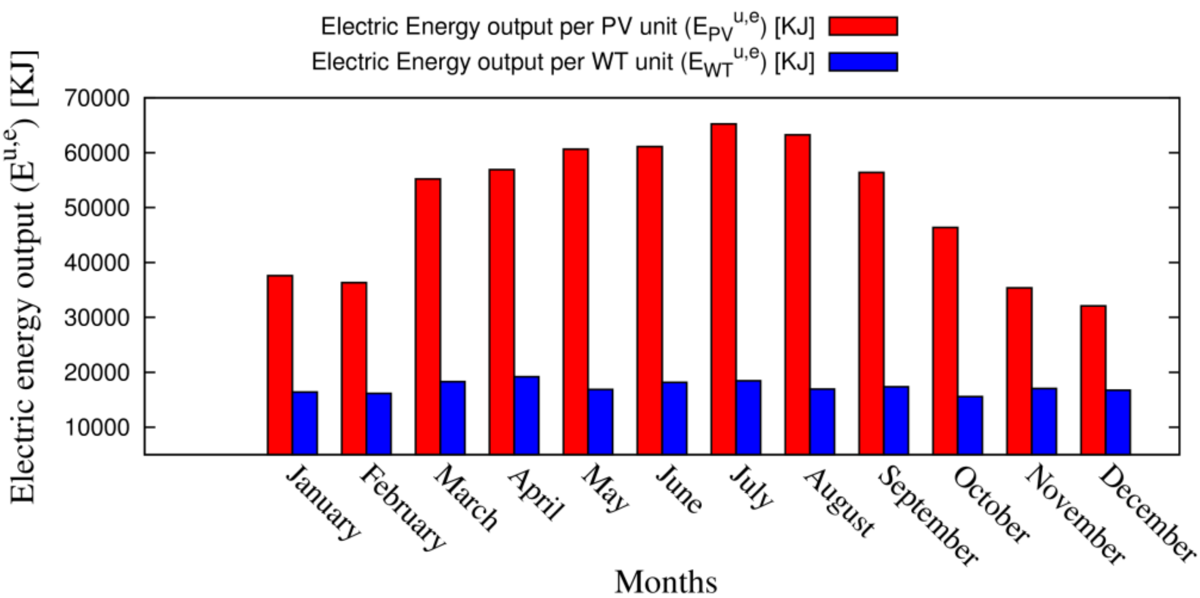


Figure 4.7 – Monthly values of electrical energy output of a base unit of each of of PV and WT systems during a typical meteorological year of Marseille

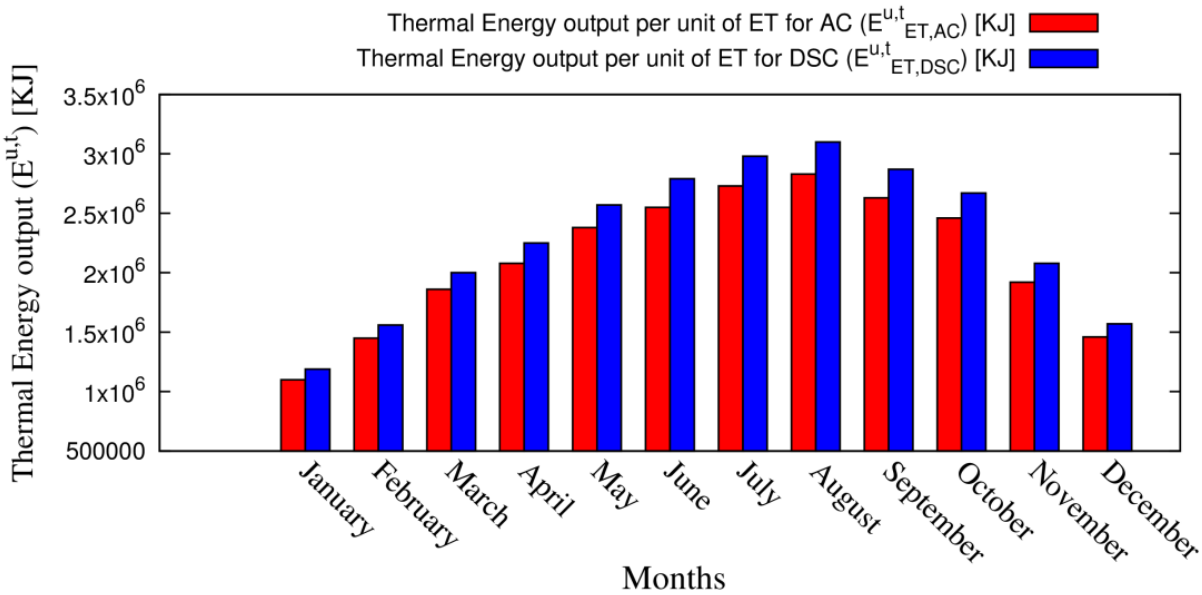


Figure 4.8 – Monthly values of thermal energy output of base units of ET, AC and ET, DSC systems during a typical meteorological year of Beirut

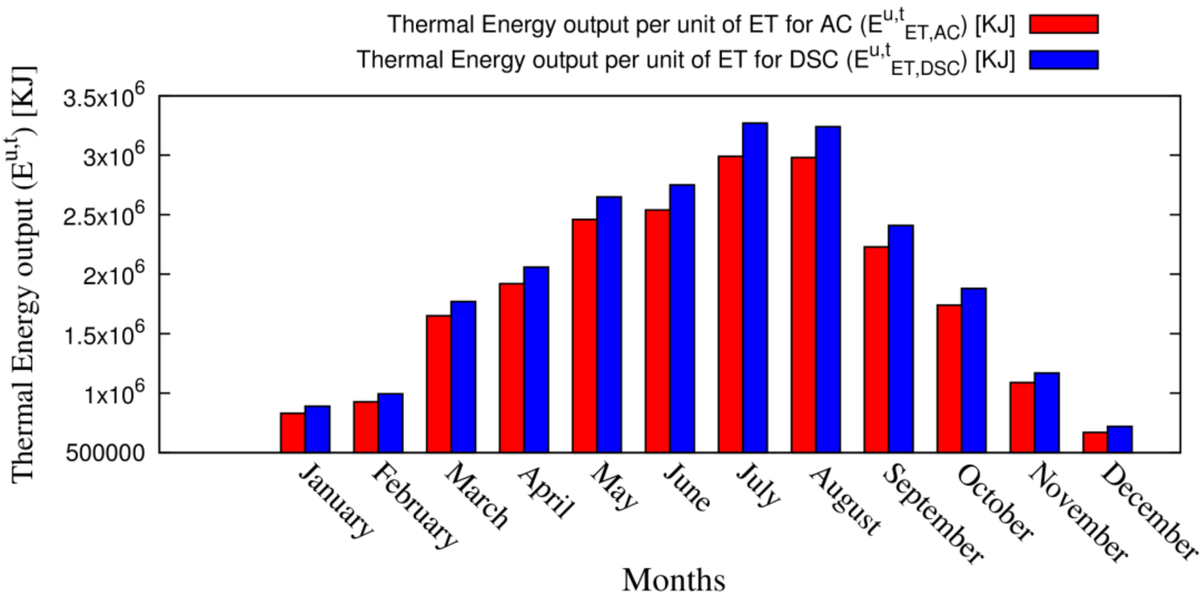


Figure 4.9 – Monthly values of thermal energy output of base units of ET, AC and ET, DSC systems during a typical meteorological year of Marseille

4.4.2.2 Results and discussion

The presented optimal sizing method for a proposed residential hybrid cooling system is investigated, where two case studies are examined, Beirut and

Marseille. Actually, the importance of the first case study lies in the fact that Lebanon is a country that suffers from a huge shortage in electricity ([265] and [266]) with a high population density of about four million people living in a small area [267]. Moreover, the policies and the plans of ministers turn into increasing the tariff of grid electricity [268]. Hence, the use of hybrid cooling based on renewable energy systems in Lebanon is an efficient method that could reduce the dependency on grid electricity. For this purpose, the hybrid system, applied to a suggested house, is modeled using Trnsys software. Simulations for a typical meteorological year of both case studies are developed in the aim of obtaining the necessary energy inputs of the problem. Other economic and characteristic inputs are adopted from the local market in each city. After that, Matlab software is utilized to model the optimal sizing method, where two time periods, 7 and 10 years, are considered for each case. Simulation results yield the optimum number of each energy unit composing the hybrid system. Table 4.3 illustrates these results for the each considered case all along with the capital cost of the system and the annual operating-cost savings compared with a conventional electric vapor compression chiller. It is shown that for the case of Beirut, the same components-configuration of the hybrid system is the same for both studied operation periods (7 and 10 years). As for Marseilles, the electric vapor compression chiller is preferred to any other hybrid configuration over the 7-years operation period. This is mainly attributed to lower electricity costs in the country compared with that in Beirut and thus, annual operating-cost savings would not be able to meet the economic criterion. On the other hand, for 10-years operation period, the same hybrid system configuration obtained for the case of Beirut is found. Moreover, comparing with the annual savings in both cities, it is clear that they are much higher in Beirut due to higher cooling loads and higher electricity tariff.

Furthermore, although the capital costs of the obtained hybrid system is high in both cities ; however, annual cost-savings are also considerable. Hence, if good financial incentives and loans exist, there would be a large spread of such high efficient energy systems.

Algorithm 1: Optimization problem resolution

Input E_s, E_l, E_L, C^{cv}, N_y, N_y^{lt}, A_X, C_X^{hyb}, E_X^u, COP_X where $X \in H1 \cup H2$

Ouput N_{PV}, N_{WT}, $N_{ET,AC}$, $N_{ET,DSC}$

Initialize Max, N_{PV}, N_{WT}, $N_{ET,AC}$, $N_{ET,DSC}$
$Max \leftarrow 0$; $N_{PV} \leftarrow 0$; $N_{WT} \leftarrow 0$; $N_{ET,AC} \leftarrow 0$; $N_{ET,DSC} \leftarrow 0$.

Compute $\alpha(N_y)$, β, γ, ρ, $\psi(\hat{N})$, A_t, L

for $i = 1$ to N_{PV}^{Max} do

  for $j = 1$ to N_{WT}^{Max} do

    for $k = 1$ to $N_{ET,AC}^{Max}$ do

      for $l = 1$ to $N_{ET,DSC}^{Max}$ do

      Compute $\alpha(N_y)$, β, γ, ρ, $\psi(\hat{N})$

$$\hat{N} = \begin{pmatrix} i \\ j \\ k \\ l \end{pmatrix}$$

        If $[(\langle \hat{N}, \gamma\rangle \leq A_T)$ **and** $(\langle \hat{N}, \rho\rangle \leq L)$ and $(\psi(\hat{N}) \leq 0)$ **and** $(\langle \hat{N}, \beta\rangle > Max)]$

          $Max \leftarrow (\langle \hat{N}, \beta\rangle$

          $N_{PV} \leftarrow i$

          $N_{WT} \leftarrow j$

          $N_{ET,AC} \leftarrow k$

          $N_{ET,DSC} \leftarrow l$

        end if

      end for

    end for

  end for

end for

Table 4.1 – Main characteristics of the main components of the proposed hybrid cooling system

Absorption Chiller	
Type	Air-cooled LiBr/Water Absorption Chiller
Nominal cooling capacity [KW]	4.5
Nominal generator heat input [KW]	7.2
Nominal inlet temperature to generator [°C]	90
COP [-]	0.7
Vapor Compression Chiller	
Type	Air-cooled vapor compression Chiller
Nominal cooling capacity [KW]	5.2
Nominal compressor power [KW]	2.07
COP [-]	2.51
Desiccant system	
Type	Rotary wheel desiccant system
Process air flow rate [CFM]	1200
Regeneration temperature [°C]	70
Percentage of return air to the process air [%]	80
COP [-]	0.6
Photovoltaic panel	
Nominal power [W]	90
Area [m^2]	1.08
Wind Turbine	
Nominal power [KW]	1
Rotor diameter [m]	2.7
Evacuated tube solar system for absorption chiller	
Collector area [m^2]	9.585
Tank volume [l]	250
Evacuated tube solar system for desiccant system	
Collector area [m^2]	20
Tank volume [l]	250

Table 4.2 – Average market prices of hybrid cooling system components in Beirut and Marseilles

Unit	Cost in Beirut	Cost in Marseille	Life time (years)
Photovoltaic Panel	1.7 US$/W	0.8 Euro/W	25
Wind Turbine	2100 US$/KW	800 Euro/KW	20
Evacuated Tube Solar Collector	335 US$/$m^2$	200 Euro/m^2	40
Vapor compression chiller	115 US$/KW	190 Euro/KW	15
Absorption Chiller	1770 US$/KW	1110 Euro/KW	23
Desiccant System	12.5 US/CFM	8.75 Euro/CFM	10
Electric Storage (Battery + Inverter)	1200 US$	500 Euro	5
Thermal Storage (Tank + Insulation)	200 US$	100Euro	10
Cost of Electricity	0.24 US$/KWh	0.121 Euro/KWh*	
*The cost of electricity for Marseille is calculated as the average of electricity prices in France			

Table 4.3 – Optimal sizing results for Beirut and Marseilles

Case Study	Beirut-Lebanon		Marseilles-France	
Parameter	for N_y= 7 years	for N_y= 10 years	for N_y= 7 years	for N_y = 10 years
Number of PV [unit]	10	10	0	10
Number of WT [unit]	2	2	0	2
Number of ET, AC [unit]	2	2	0	2
Number of ET, DSC [unit]	1	1	0	1
Number of AC [unit]	1	1	0	1
Number of VCC [unit]	1	1	1	1
Number of DSC [unit]	1	1	0	1
Total Capital Cost [US$]	32130	32130	1087	22242
Annual Saving [US$/year]	795	795	0	253
The currency exchange rate conversion factor from Euro to US$ used is 1.1.				

General Conclusions and Perspectives

In the present thesis, we are interested in cooling systems for building use.

The first contribution in this study is the Effectiveness Factor ($\mathcal{EF}$) defined. It serves as a comparative factor of the behavior of solar absorption chiller in different cities situated on different latitudes. The effectiveness factor combines the effect of different energetic and economic parameters. It was found that the behavior of solar absorption chiller -defined by the effectiveness factor-is closely related to the location of a region on latitude lines. The system studied has recorded the highest yearly $\mathcal{EF}$ in cities located between $10 \circ N$ and $25 \circ N$. Then, the factor decreases -symmetrically- from both sides out of this latitude intervals.

A hybrid cooling system is one of the energy efficient technology investigated in the literature. However, it could have a negative repercussion if it does not match the climatic zone where it will be used. The second contribution of this thesis is the suggestion of a selection scheme to recommend the best hybrid cooling system. It allows to find the cooling system with the minimum energy consumption and pollution emissions according to different parameters.

The third contribution focuses on the sizing of a hybrid cooling system. A hybrid system based on renewable energy sources is proposed. The system consisted of three cooling machines : absorption chiller, vapor compression chiller and desiccant dehumidifier. The system is driven by different type of energy captors : solar evacuated tube collectors, photovoltaic panels and wind turbine. An optimal sizing method is proposed in order to determine the optimal number of the hybrid system's components. An objective function that tends to minimize the electrical energy from the grid is defined. The optimization problem is solved under different energetic and economic constraints. Also, the possibility of installing renewable energy systems is investigated. This method aims to find a cooling system economically feasible with maximum energy share.

The studies performed in the present thesis provided renewed perspectives that may be the object of future works.

More improvement needs to be made on the $\mathcal{EF}$ so it combines the effect of all parameters related to the machine operation and cost as well as the impact on environment and CO_2 emissions. In addition, an index that compare the behavior of different machines should be implemented.

Concerning the hybrid system, the optimal sizing method is necessary to find the best configuration of cooling system. However, this method need to be improved, and an optimal strategy for energy management and control -according to the cooling demand, energy input, costs, etc. -should be implemented. Hence, it allows the components of the system to operate at its maximal performance. Finally, both concepts introduced in this work (1) the effectiveness factor and (2) the optimal sizing method must be combined in the future studies.

The works of the present thesis are the subject of three papers in international journals :

[1] Farah Kojok, Farouk Fardoun, Rafic Younes, and Rachid Outbib "Hybrid cooling systems : A review and an optimized selection scheme"
Renewable and Sustainable Energy Reviews (IF 6.798), Volume 65, pp. 57-80, 2016.

[2] Rafic Younes, Farah Kojok, Farouk Fardoun and Rachid Outbib "Effectiveness of solar absorption machine in representative locations"
In revision and will be re-submitted to *Solar Energy* (IF 3.685).

[3] Farah kojok, Rachid Outbib, Oussama Ibrahim and Rafic Younes "Novel optimal sizing method for a renewable energy-based hybrid cooling system"
Under preparation and will be submitted to *Energy Conversion and Management* (IF 4.801).

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