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Faculty of Engineering III

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Energy section

RECOVERY OF WASTE HEAT BY THE PRODUCTION OF COOLING EFFECT

EXERGY ANALYSIS AND COMPARISON BETWEEN DIFFERENT SYSTEMS

BY

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A dissertation submitted to the Department of Mechanical Engineering in Conformity with the Department Requirements as partial fulfillment for the diploma in Mechanical Engineering-Energy section

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Preface

There is a strong tendency to use our energy resources as efficient as possible, all over the world, and any disesteem in the use of energy has a great negative influence on the subsequent generations.

Energy is universally acknowledged to be the mainstay of an industrial society. Without an adequate supply of energy, the stability of the social and economic order, as well as the political structure of a society is in jeopardy. As the world supply of inexpensive, but nonrenewable, fossil energy sources decreases, the need for energy conservation as well as for developing renewable technologies becomes ever more critical.

Recently, the issue of energy efficiency and conservation emerged as a serious challenge because it was recognized that burning of fossil fuels is one of the main contributors to global warming. Global warming is largely the result of the emission of radiation-trapping gases, such as carbon dioxide and methane, into the atmosphere. It is now the consensus of the scientific community that artificial CO_2 pollution is largely responsible for the increase in the average global temperature. Improving energy efficiency and conservation in the use of fossil fuels is therefore an important challenge for the engineering community.

In this study, I have studied one of the situations where heat is wasted to the environment; it's the case where hot exhausts are rejected to the environment without any benefit from the great energy which is hidden in these exhausts, I have studied the exergetic state of a hot exhaust with each of the three different thermodynamic cycles (Absorption, drying, and Rankin cycles), identifying the situations where the extraction of the energy from these exhausts is economic, to identify the most efficient cycle to implement in the production of the cooling effect from any hot exhaust, finally I have done a conclusion of my work in this domain.

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ALI M. EZZEDDINE

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Acronyms and Abbreviations

Nomenclature

h	enthalpy of a unit mass (kJkg^{-1})
ex	exergy of a unit mass (kJkg^{-1})
ṁ	mass flow rate (kgs^{-1})
T	temperature (K)
COP	coefficient of performance
s	entropy of a unit mass (kJkg^{-1}K^{-1})
v	velocity (m/s)
Q	heat (J)
W	work (J)
T	temperature (K)
N	number of mole (moles)
FLT	first law of thermodynamics
SLT	second law of thermodynamics
V	volume (m^3)
X	mass fraction of the absorbent or desiccant
P	pressure (bar)
LiBr	Lithium bromide
t	temperature (○C)
Y	air absolute humidity (kg/kg)
Gs,d	specific desiccant flow in the dehumidifier
Ga,d	specific air flow in the dehumidifier
Gs,r	specific desiccant flow in the regenerator
Ga,r	specific air flow in the regenerator (Kg/s.m2)
U	internal energy (J)
KE	kinetic energy (J)
PE	potential energy (J)
m	mass (Kg)
E	total internal energy (J)
Z	Height (m)

Subscripts

s	strong
w	weak/water
r	refrigerant
o	environmental state
ch	chemical
ph	physical
f	flow
i	ranking index
d	destruction
g	global
a	available
abs	absorber
b	boiler
evap,e	evaporator
P	pump
cond,c	condenser
c	cycle
comp	compressor
1s,2s…	ideal isotropic state
n	nozzle
m	mixing

exp	expansion valve
ex	exergy
en	energy
sys	system
rev	reversible
surr	surrounding
gen	generator, generated
cw	chilled water
acw	absorber cooling water
ccw	condenser cooling water
TV	throttling valve
deh	dehumidifier
SHX	solution heat exchanger
clg	cooling water
EXRA	external heating of regenerator air
CWHE	cooling water heat exchanger
EHWE	external heating of weak desiccant

Greek symbols

μ	chemical potential
η	efficiency
β	temperature difference ratio
ε	effectiveness
ρ	density
Δ	difference

Superscripts

o	dead state
2,3,4,…	power

Chapter 1: WASTE HEAT

Waste heat is heat, which is generated in a process by way of fuel combustion or chemical reaction, and then "dumped" into the environment even though it could still be reused for some useful and economic purpose. The essential quality of heat is not the amount but rather its "value".

Waste heat has no useful application is found, and is regarded as a waste by-product. When produced by humans, or by human activities, it is a component of anthropogenic heat, which additionally includes unintentional heat leakage, such as from space heating. Waste heat is thought by some to contribute to the urban heat island effect.

Waste heat quality

Depending upon the type of process, waste heat can be rejected at virtually any temperature from that of chilled cooling water to high temperature waste gases from an industrial furnace or kiln. Usually higher the temperature, higher the quality and more cost effective is the heat recovery. In any study of waste heat recovery, it is absolutely necessary that there should be some use for the recovered heat. Typical examples of use would be preheating of combustion air, space heating, or pre-heating boiler feed water or process water. With high temperature heat recovery, a cascade system of waste heat recovery may be practiced to ensure that the maximum amount of heat is recovered at the highest potential. An example of this technique of waste heat recovery would be where the high temperature stage was used for air pre-heating and the low temperature stage used for process feed water heating or steam raising.

In considering the potential to recover heat, it is useful to note all the possible sources of waste and their quality and possible uses (table 1).

Waste heat quantity

In any heat recovery situation it is important to know the amount of recoverable heat and also its usage. The total heat that could potentially be recovered can be calculated using this formula:

$$Q = q \times \rho \times C_p \times \Delta T$$

Where,
Q   is the heat content in Kw
q   is the flow rate of the substance in m^3/s
ρ is density of the flue gas in kg/m^3
C_p is the specific heat of the substance in kJ/kg ^{O}C
ΔT is the temperature difference in ^{O}C

Source of waste heat	Quality of waste heat and possible use
Heat in flue gases	The higher the temperature, the greater the potential value for heat recovery
Heat in vapor streams	As for heat in flue gases, but when condensed, latent heat is also recoverable
Convective & radiant heat lost from exterior of equipment	Low grade – if collected, may be used for space heating or air preheats
Heat losses in cooling water	Low grade – useful gains if heat is exchanged with incoming fresh water
Heat losses in providing chilled water or in the disposal of chilled water	1. High grade if it can be utilized to reduce demand for refrigeration 2. Low grade if refrigeration unit used as a form of Heat pump
Heat stored in products leaving the Process	Quality depends upon temperature
Heat in gaseous & liquid effluents leaving process	Poor, if heavily contaminated & thus require alloy heat exchanger

Table 1: Waste Heat general Sources and Quality [1]

Classifications and Sources of waste heat

Waste heat can be recovered from various industrial processes. A distinction is made between high, medium and low temperatures of waste heat [1]

- The high temperature range refers to temperature above 650 ^{o}C;
- The medium temperature range is between 232 ^{o}C and 650 ^{o}C ;
- The low temperature range is below 232 ^{o}C.

High and medium temperature waste heat can be used to produce process steam. If high temperature waste heat exists, instead of producing steam directly, consider the possibility of using the high temperature energy to do useful work before the waste is extracted. Both gas and steam turbines are useful and fully developed heat engines. In the low temperature range, waste energy which would be otherwise useless can sometimes be made useful by application of mechanical work through a device called the heat pump.

Tables 2 through 4 illustrate the sources of waste heat according to their classification.

Types of Devices	Temperature (^{o}C)
Nickel refining furnace	1370 – 1650
Aluminum refining furnace	650 –760
Zinc refining furnace	760 – 1100
Copper refining furnace	760 – 815
Steel heating furnace	925 – 1050
Copper reverberatory furnace	900 – 1100
Open hearth furnace	650 – 700
Cement kiln (Dry process)	620 – 730
Glass melting furnace	1000 – 1550
Hydrogen plants	650 – 1000
Solid waste incinerators	650 – 1000

Types of Devices	Temperature (^{O}C)
Steam boiler exhaust	232 – 480
Gas turbine exhaust	370 – 540
Reciprocating engine exhaust	315 – 600
Reciprocating engine exhaust (turbo charged)	232 – 370
Heat treatment furnace	425 – 650
Drying & baking ovens	232 – 600
Catalytic crackers	425 – 650
Annealing furnace cooling systems	425 – 650

Table 3: Typical Waste Heat Temperature at Medium Temperature Range from Various Sources [1]

Types of Devices	Temperature (^{O}C)
Process steam condensate	55-88
Cooling water from: Furnace doors	32-55
Bearings	32-88
Welding machines	32-88
Injection molding machines	32-88
Annealing furnaces	66-230
Forming dies	27-88
Air compressors	27-50
Pumps	27-88
Internal combustion engines	66-120
Air conditioning and refrigeration condensers	32–43
Liquid still condensers	32-88
Drying, baking and curing ovens	93-230
Hot processed liquids	32-232
Hot processed solids	93-232

Table 4: Typical Waste Heat Temperature at Low Temperature Range from Various Sources [1]

Waste Heat recovery

What is waste heat recovery?

Waste heat recovery refers to the recuperation of heat that is discharged as a byproduct from one process to provide supplemental energy needed by another process. For example, by recovering the heat from your cooling system, you can heat water for free. Heat recovery is cost effective technology for restaurants, supermarkets, and food processors...any business that needs hot water and uses refrigeration or air conditioning.

Why recover heat?

With increasing oil prices and higher environmental taxes many industrial processes can make substantial cost savings by installing a heat recovery system. During the last 15 years, oil prices have increased 60-70%. The incentive to recover energy is and will become more and more evident.

Heat recovery also means lower exhaust gas volumes and lower emissions to the atmosphere. Heat recovery gives direct savings in terms of lower fuel consumption as well as indirect savings in lower environmental impact. Last, but not least, it facilitates smaller and more economic furnaces.

The question should in fact be: Why not recover heat? Amazingly, almost 100 percent of the electrical energy input to a compressor is turned into heat. Approximately 2 percent of this energy radiates away from the compressor package into the ambient surroundings and 2 percent remains in the compressed air. This means that with installation of heat recovery systems, up to 96 % of the energy dedicated to compressed air production can be recovered and reused. Therefore, in view of continuously soaring energy prices and concerns regarding the environment, the potential benefits and savings are enormous.

Benefits of Waste Heat Recovery

Benefits of 'waste heat recovery' can be broadly classified in two categories:

Direct Benefits:

Recovery of waste heat has a direct effect on the efficiency of the process. This is reflected by reduction in the utility consumption & costs, and process cost.

Indirect Benefits:

- **Reduction in pollution**: A number of toxic combustible wastes such as carbon monoxide gas, sour gas, carbon black off gases, oil sludge, Acrylonitrile and other plastic chemicals etc, releasing to atmosphere when burnt in the incinerators serves dual purpose i.e. recovers heat and reduces the environmental pollution levels.

- **Reduction in equipment sizes**: Waste heat recovery reduces the fuel consumption, which leads to reduction in the flue gas produced. This results in reduction in equipment sizes of all flue gas handling equipments such as fans, stacks, ducts, burners, etc.

- **Reduction in auxiliary energy consumption**: Reduction in equipment sizes gives additional benefits in the form of reduction in auxiliary energy consumption like electricity for fans, pumps etc...

Our case study

In our case study we have a low temperature source, and we will do our study in order to benefit from this waste heat source as maximum as possible.

Study of the hot exhaust of the manufacturing processes of industrial gases

The manufacturing processes of industrial gases are major consumers of energy as electricity or fossil fuel. A big part of this energy ends up as heat in the process gas at temperatures around 150 ° C. Today, this thermal energy is generally released to the atmosphere through cooling towers (open circuit or closed circuit cooling towers).

Air-cooled huge compressors are one of the fields where air is exhausted at low temperatures; these compressors have very large surface to horsepower ratio which effectively dissipates the heat of compression through the compressor case. These compressors must have positive air flow whenever the compressors are operating in order to guarantee safe operating temperatures. The required air flow can be provided by a condenser fan of adequate size blowing on the compressor or by a separate fan which provides the required air flow. These fans must operate whenever the compressor operates, even under winter operating conditions.

So we have a hot stream originated from a compressor air-cooled system in an industrial gases manufacturing industry that has the following conditions:

Temperature(^{O}C)	Pressure(bar)	Water (%molar)	Flow rate(kg/s)	Dew point(^{O}C)	$\dot{E}x_{1f,3f}$ (Kw)
165	25	2	152.2	75	4950*

* $\dot{E}x_{1,3} = \dot{m}_f(ex_{1f}-ex_{3f})$; ex_1, ex_3 are taken from standard tables

Industry

Recovery of waste heat by the production of cooling effect; Exergy analysis and comparison between different systems
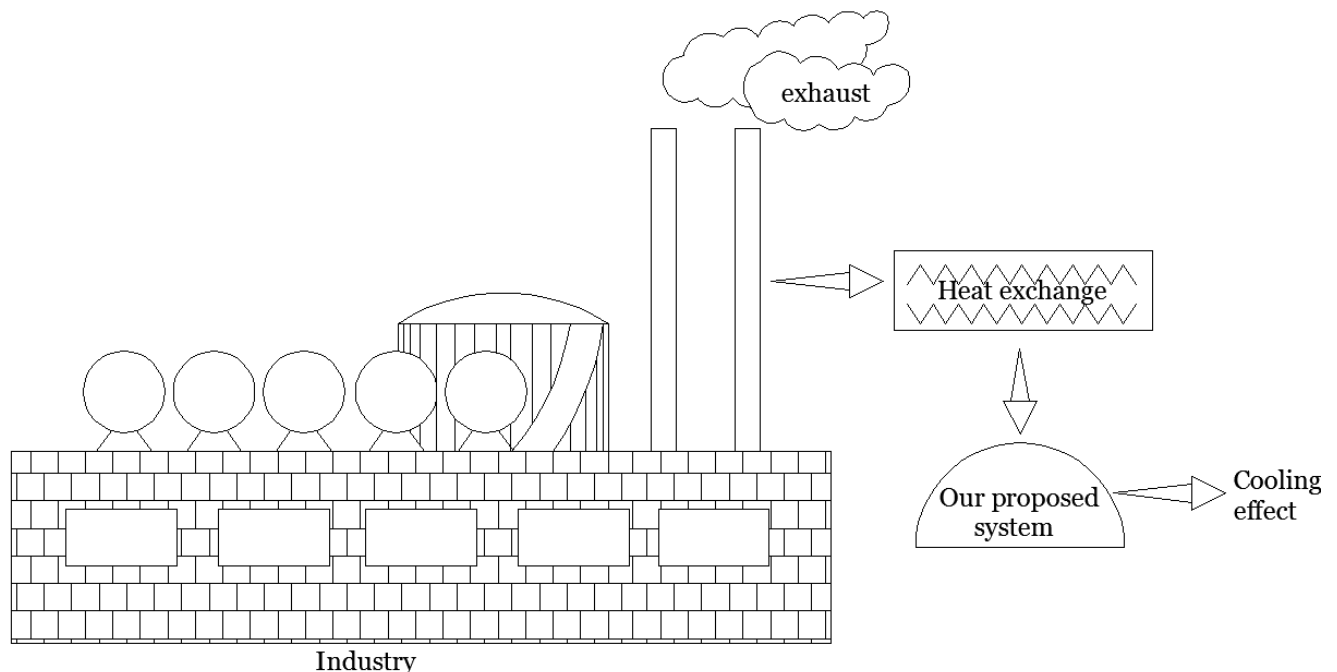
CHAPTER 2: ESTIMATING OUR RESOURCE POTENTIAL

Since we are talking about a thermodynamic system (exhaust hot stream resource analysis), which is a system where we have an interaction between the three basic concepts of thermodynamics, energy, entropy, and exergy, we must do a thermodynamic analysis for this system, concerning the behavior, performance, and efficiency of extracting energy from that hot exhaust.

How to estimate our resource potential?

An energy analysis of a system is essentially an accounting of the energies entering and exiting. Efficiencies are often evaluated as ratios of energy quantities, and are often used to assess and compare various systems. Power plants, heaters, refrigerators and thermal storages, for example, are often compared based on energy efficiencies or energy-based measures of merit. However, energy efficiencies are often misleading in that they do not always provide a measure of how nearly the performance of a system approaches ideality. Further, the thermodynamic losses which occur within a system (i.e., those factors which cause performance to deviate from ideality) often are not accurately identified and assessed with energy analysis. The results of energy analysis can indicate the main inefficiencies to be within the wrong sections of the system, and a state of technological efficiency different than actually exists.

Exergy analysis permits many of the shortcomings of energy analysis to be overcome. Exergy analysis is based on the second law of thermodynamics, and is useful in identifying the causes, locations and magnitudes of process inefficiencies. The exergy associated with an energy quantity is a quantitative assessment of its usefulness or quality. Exergy analysis acknowledges that, although energy cannot be created or destroyed, it can be degraded in quality, eventually reaching a state in which it is in complete equilibrium with the surroundings and hence of no further use for performing tasks.

Exergy analysis states the theoretical limitations imposed on a system, clearly pointing out that no real system can conserve exergy and that only a portion of the input exergy can be recovered. Also, exergy analysis quantitatively specifies practical limitations by providing losses in a form in which they are a direct measure of lost exergy.

From the above comparison between energy analysis and exergy analysis, we infer that exergy analysis is more convenient for our case study.

Thermodynamics approach

In our case study, we're facing a domain where energy, entropy, and exergy exist. Thus I will introduce my understanding to these basic concepts.

Energy [2]

Energy concept

The concept of energy is so familiar to us today that it seems intuitively obvious to understand, yet we often have difficulty defining it precisely.

Energy is a scalar quantity that cannot be observed directly but can be recorded and evaluated by indirect measurements. The absolute value of the energy of a system is difficult to measure, whereas the energy change is relatively easy to evaluate. Energy is also associated with the structure of matter and can be released by chemical and atomic reactions.

General forms of energy

Energy manifests itself in many forms, which are either internal or transient. Energy can be converted from one form to another. In thermodynamic analysis, the forms of energy can be classified into two groups: macroscopic and microscopic.

Macroscopic forms of energy: are those which an overall system possesses with respect to a reference frame, e.g., kinetic and potential energies. The macroscopic energy of a system is related to motion and the influence of external effects such as gravity, magnetism, electricity and surface tension.

Microscopic forms of energy*:* are those related to the molecular structure of a system and the degree of molecular activity, and are independent of outside reference frames. The sum of all the microscopic forms of energy of a system is its internal energy. The internal energy of a system depends on the inherent qualities, or properties, of the materials in the system, such as composition and physical form, as well as the environmental variables (temperature, pressure, electric field, magnetic field, etc.). Internal energy can have many forms, including mechanical, chemical, electrical, magnetic, surface and thermal.

The structure of thermodynamics involves the concept of equilibrium states and postulates that the change in the value of thermodynamic quantities, such as internal energy, between two equilibrium states of a system does not depend on the thermodynamic path the system takes to get from one state to the other. The internal energy thus is referred to as a state function, or a point function, i.e., a function of the state of the system only, and not its history.

The first law of thermodynamics

The first law of thermodynamics (FLT) is the law of the conservation of energy, which states that, although energy can change form, it can be neither created nor destroyed. The FLT defines internal energy as a state function and provides a formal statement of the conservation of energy.

However, it provides no information about the direction in which processes can spontaneously occur, i.e., the reversibility aspects of thermodynamic processes. The FLT provides no information about the inability of any thermodynamic process to convert heat fully into mechanical work, or any insight into why mixtures cannot spontaneously separate or un-mix themselves. A principle to explain these

phenomena and to characterize the availability of energy is required to do this. That principle is embodied in the second law of thermodynamics (SLT).

Energy and the FLT

For a control mass, the energy interactions for a system may be divided into two parts: dQ, the amount of heat, and dW, the amount of work. Unlike the change in total internal energy dE, the quantities dQ and dW are not independent of the manner of transformation, so we cannot specify dQ and dW simply by knowing the initial and final states. Hence it is not possible to define a function Q which depends on the initial and final states, i.e., heat is not a state function. The FLT for a control mass can be written as follows:

$$dQ = dE + dW$$

When the above equation is integrated from an initial state 1 to a final state 2, it results in

$$Q_{1-2} = E_2 - E_1 + W_{1-2} \; or \; E_2 - E_1 = Q_{1-2} - W_{1-2}$$

Where E_1 and E_2 denote the initial and final values of the energy E of the control mass, Q_{1-2} is the heat transferred to the control mass during the process from state 1 to state 2, and W_{1-2} is the work done by the control mass during the process from state 1 to state 2.

The energy E may include internal energy U, kinetic energy KE and potential energy PE terms as follows:

$$E = U + KE + PE$$

For a change of state from state 1 to state 2 with a constant gravitational acceleration g, becomes

$$E_2 - E_1 = U_2 - U_1 + m\frac{(v_2^2 - v_1^2)}{2} + mg(Z_2 - Z_1)$$

Where m denotes the fixed amount of mass contained in the system, v the velocity and Z the elevation. The quantities dQ and dW can be specified in terms of the rate laws for heat transfer and work. For a control volume an additional term appears from the fluid flowing across the control surface (entering at state i and exiting at state e). The FLT for a control volume can be written as

$$Q_{cv} = \dot{E}_{cv} + \dot{W}_{cv} + \sum \dot{m}\,\hat{h}_e - \sum \dot{m}\,\hat{h}_i \; or \; \dot{E}_{cv} = Q_{cv} - \dot{W}_{cv} + \sum \dot{m}\,\hat{h}_i - \sum \dot{m}\,\hat{h}_e$$

Where $\dot{m}$ is mass flow rate per unit time, $\hat{h}$ is total specific energy, equal to the sum of specific enthalpy, kinetic energy and potential energy, i.e. $\hat{h} = h + {v^2}/{2} + gZ$

Entropy [2]

To understand the property 'entropy' we should know some basic concepts, like, ordered energy, disordered energy, reversibility, and irreversibility.

Ordered and disordered energy

Much of the internal energy of a substance is randomly distributed as kinetic energy at the molecular and sub-molecular levels and as energy associated with attractive or repulsive forces between molecular and sub-molecular entities, which can move closer together or further apart. This energy is sometimes described as being 'disordered' as it is not accessible as work at the macroscopic level in the same way as is the kinetic energy or gravitational potential energy that an overall system possesses due to its velocity or position in a gravitational field. The term disorder refers to the lack of information about exactly how much and what type of energy is associated at any moment with each molecular or sub-molecular entity within a system.

At the molecular and sub-molecular level there also exists 'ordered energy' associated with the attractive and repulsive forces between entities that have fixed mean relative positions. Part of this energy is, in principle, accessible as work at the macroscopic level under special conditions.

Heat transfer to a system increases its disordered energy, while heat transfer from a system reduces its disordered energy.

Reversibility and Irreversibility

Reversible process is a process that, after it has taken place, can be reversed and causes no change in either the system or its surroundings. In thermodynamic terms, a process "taking place" would refer to its transition from its initial state to its final state.

Irreversible process occurs when a change in the thermodynamic state of a system and all of its surroundings cannot be precisely restored to its initial state by infinitesimal changes in some property of the system without expenditure of energy. A system that undergoes an irreversible process may still be capable of returning to its initial state; however, the impossibility occurs in restoring the environment to its own initial conditions.

An irreversible process increases the entropy of the system, which is a measure of the microscopic disorder of the system whereas a reversible process does not.

Entropy concept

Entropy is a measure of the amount of molecular disorder within a system. A system possessing a high degree of molecular disorder (such as a high-temperature gas) has a high entropy and vice versa.
The change in entropy between states is defined as the integral of the ratio of the reversible heat transfer to the absolute temperature.

Characteristics of entropy

- The entropy of a system is a measure of its internal molecular disorder.
- A system can only generate, not destroy, entropy.
- The entropy of a system can be increased or decreased by energy transports across the system boundary.

The entropy of a system at some state is a measure of the probability of its occurrence, with states of low probability having low entropy and states of high probability having high entropy. The entropy of a system must increase in any transfer or conversion of energy, because the spontaneous direction of the change of state of a closed system is from a less to a more probable state. In any energy transfer or conversion within a closed system, the entropy of the system increases.

In open systems, energy conversions can occur which cause the entropy of part or all of a system to decrease. If the combination of the system and its surroundings is considered, however, the overall net effect is always to increase disorder.
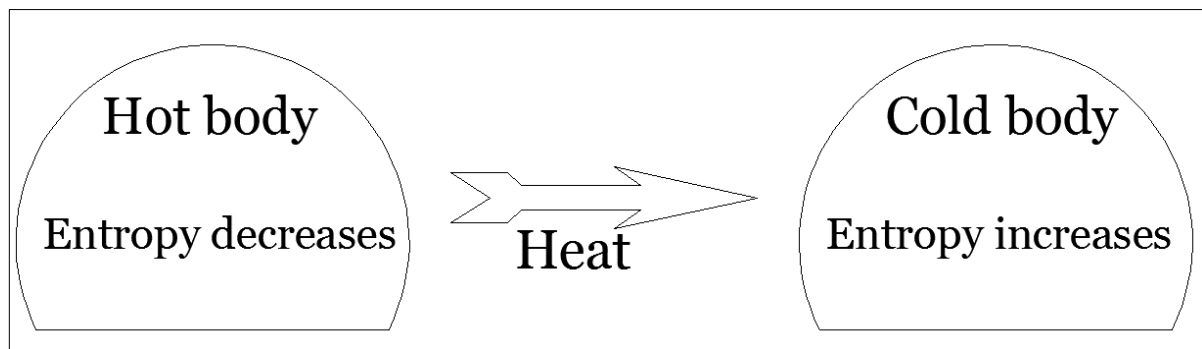
Figure 1: Illustration of entropy increase and decrease for cold and hot bodies during heat transfer

The second law of thermodynamics

The SLT establishes the difference in the quality of different forms of energy and explains why some processes can spontaneously occur while other cannot. The SLT is usually expressed as an inequality, stating that the total entropy after a process is equal to or greater than that before. The equality only holds for ideal or reversible processes. It also states that all spontaneous processes, both physical and chemical, proceed to maximize entropy, i.e., to become more randomized and to convert energy to a less available form. A corollary of the SLT is the statement that the sum of the entropy changes of a system and that of its surroundings must always be positive.

We have two well-known statements of SLT

- **Clausius statement**: It is impossible for heat to move of itself from a lower-temperature reservoir to a higher temperature reservoir. That is, heat transfer can only occur spontaneously in the direction of temperature decrease. For example, we cannot construct a refrigerator that operates without any work input.
- **Kelvin–Planck statement**: It is impossible for a system to receive a given amount of heat from a high-temperature reservoir and to provide an equal amount of work output. While a system

converting work to an equivalent energy transfer as heat is possible, a device converting heat to an equivalent energy transfer as work is impossible. Alternatively, a heat engine cannot have a thermal efficiency of 100%.

The Clausius inequality

$$\oint \frac{\delta Q}{T} \leq 0$$

The system undergoes only reversible processes (or cycles) if the cyclic integral equals zero, and irreversible processes (or cycles) if it is less than zero.

Without inequality

$$S_{\text{gen}} = -\oint \frac{\delta Q}{T}$$

Where

$$\Delta S_{gen} = \Delta S_{total} = \Delta S_{sys} + \Delta S_{surr}$$

The quantity S_{gen} is the entropy generation associated with a process or cycle, due to irreversibilities. The following are cases for values of S_{gen}:

- $S_{gen} = 0$ for a reversible process
- $S_{gen} > 0$ for an irreversible process
- $S_{gen} < 0$ for no process

One can write

- For reversible process

$$\Delta S_{sys} = \left(\frac{Q}{T}\right)_{rev} \; and \; \Delta S_{surr} = -\left(\frac{Q}{T}\right)_{rev}$$

- For irreversible process

$$\Delta S_{sys} > \Delta S_{surr}$$

Some useful relations

$\boldsymbol{\delta q = du + \delta w}$ (A FLT statement applicable to any simple compressible closed system)
$\boldsymbol{\delta q = du + pdv}$ (A FLT statement restricted to reversible processes for a closed system)
$\boldsymbol{Tds = du + \delta w}$ (A combined statement of the FLT and SLT, with $Tds = dq$)
$\boldsymbol{Tds = du + pdv}$ (A combined statement of the FLT and SLT valid for all processes between equilibrium states)

Exergy [2]

To understand the exergy concept, we will define some concepts related to the exergy property.

Reference environment [3]

An idealization of the natural environment that is characterized by a perfect state of equilibrium that is, the absence of any gradients or differences involving pressure, temperature, chemical potential, kinetic energy, and potential energy. The environment constitutes a natural reference medium with respect to which the exergy of different systems is evaluated.

Environmental state [3]

The state of a system when it is in thermal and mechanical equilibrium with the reference environment (i.e., at pressure P_o and temperature T_o of the reference environment)

Exergy destruction [3]

The exergy consumed or destroyed during a process due to irreversibilities within the system boundaries. Also known as dissipation, irreversibility, and lost work. Real processes are always irreversible (i.e., exergy is destroyed) in order to occur.

Dead state [3]

The state of a system when it is in thermal, mechanical, and chemical equilibrium with a conceptual reference environment (having intensive properties pressure P_o; temperature T_o; and chemical potential μ_{K0} for each of the reference substances in their respective dead states); under these conditions, the system cannot undergo any change of state starting from interactions with the ambient conditions [4].

Exergy concept and characteristics

Any lack of mutual stable equilibrium between a system and the environment can be used to produce shaft work. The exergy of a system is defined as the maximum shaft work that can be done by the composite of the system and a specified reference environment. Exergy is not simply a thermo-dynamic property, but rather is a property of both a system and the reference environment.

The term exergy comes from the Greek words "ex″ and "ergon″, meaning from and work. The following are some terms found in the literature that are equivalent or nearly equivalent to exergy: available energy, essergy, utilizable energy, available energy and availability.

Exergy has the characteristic that it is conserved only when all processes occurring in a system and the environment are reversible. Exergy is destroyed whenever an irreversible process occurs.

The thermodynamic imperfections can be quantified as exergy destructions, which represent losses in energy quality or usefulness. Like energy, exergy can be transferred or transported across the boundary of a system. For each type of energy transfer or transport there is a corresponding exergy transfer or transport.

Exergy efficiencies are measure of approach to ideality (or reversibility). This is not necessarily true for energy efficiencies, which are often misleading.

Exergy and energy comparison

Energy analysis is used traditionally in performing energy balances; it's based on the FLT, and evaluating energy efficiencies. But energy balance gives no information on the degradation of energy or resources during a process and doesn't qualify the usefulness or quality of the various energy and material stream flowing through a system a system and exiting as products and wastes.

The concept of exergy is based on both the FLT and the SLT. Exergy analysis indicates the locations of energy degradations; also quantify the quality of heat in a waste stream.

A comparison between energy and exergy is presented in the following table (table 5)

Energy	Exergy
Dependent on the matter or energy flow properties	Dependent on properties of both a matter or energy flow and the environment
Has values different from zero when it is in equilibrium with the environment (including being equal to mc^2 in accordance with Einstein's equation)	Equal to zero when in the dead state by virtue of being in complete equilibrium with the environment
Conserved for all processes, based on the FLT	Conserved for reversible processes and not conserved for real processes (where it is partly or completely destroyed due to irreversibilities), based on the SLT
Can be neither destroyed nor produced	Can be neither destroyed nor produced in a reversible process, but is always destroyed (consumed) in an irreversible process
Appears in many forms (e.g., kinetic energy, potential energy, work, heat) and is measured in that form	Appears in many forms (e.g., kinetic exergy, potential exergy, work, thermal exergy), and is measured on the basis of work or ability to produce work
A measure of quantity only	A measure of quantity and quality

Table 5: Comparison between energy and exergy

Exergy of a system

The exergy transfer takes place by three kinds of transfers, mass flow (Ex_{flow}), work (Ex_w), and heat (Ex_Q).

We have two types of systems

- Open (flow)
- Closed (non-flow)

The difference between the above mentioned types is that the first one allows the mass exchange with its environment in addition to heat and work exchange; while the second one only allows the heat and work exchange.

In our study we are concerned in open systems

The exergy of flow stream of matter is the sum of physical and chemical exergies

$$Ex_{flow} = Ex_{ph} + Ex_{ch}$$

$$Ex_{flow} = H - T_o S - (P - P_o)V - \sum N_i \mu_i^o + v^2/2 + gz$$

Physical exergy (Ex_{ph})
Physical exergy is equal to the maximum amount of obtainable work when the stream of substance is brought from its initial state to the environmental state by physical processes involving only an interaction with the environment [5]

$$Ex_{ph} = H - T_o S - (P - P_o)V - \sum N_{io} \mu_{io} + v^2/2 + gz \quad N_{io} = N_i$$

In the case of:

- Constant volume system
- Conservation of number of moles of each substance in the system (no chemical reaction)
- Negligible kinetic and potential energies

We can define the physical exergy as $Ex_{ph} = H - T_o S$

Chemical exergy (Ex_{ch})
Chemical exergy is equal to the maximum amount of obtainable work when the substance under consideration is brought from the environmental state to the dead state by processes involving and exchange of substances only with the environment [5] Then the chemical exergy is the flow exergy at the environmental state, ($P = P_o$; $T = T_o$; $N = N_{io}$; all parameters with o index)

$$EX_{ch} = \sum N_{io} (\mu_{io} - \mu_i^o)$$

Where P, V, T, are the pressure, volume, and temperature respectively, N_i and μ_i are the number of moles and the chemical potential of the species 'i' respectively, the index o refers to the environmental state of the system, and the exponent o refers to the dead state.
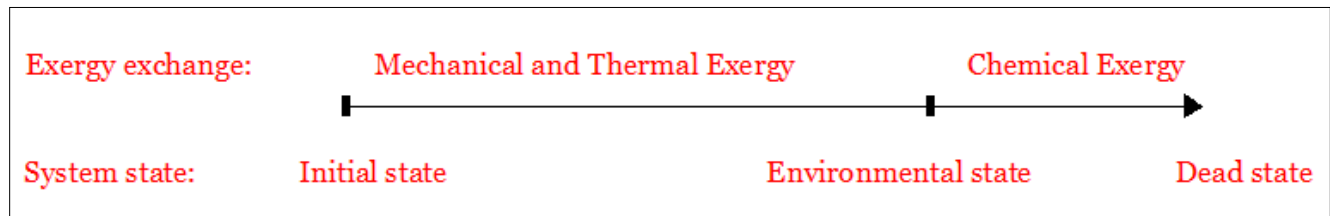
Figure 2: Exergy exchange in the different states of the system

Heat exergy
The maximum amount of obtainable work when we are dealing with heat source at temperature $T > T_o$ or $T < T_o$, by transferring heat Q. The maximum amount of work that can be obtained from a heat source is the work output from a Carnot heat engine which works between the heat source and the environment.

$$Ex_Q = Q\left(1 - \frac{T_o}{T}\right)$$

Work exergy

Exergy is the useful potential work. For boundary work, such as the expansion work of a piston-cylinder device, a portion of the work is used to push the atmosphere air away and it cannot be utilized. Thus, the exergy transfer by the expansion work equals the difference between the expansion work and the surroundings work.

$$Ex_w = W - P_o\Delta V$$

But in the case of constant volume system; then we can define the work exergy as

$$Ex_w = W$$

Exergy efficiencies

Energy efficiency is used traditionally to gauge the performance of any device; energy efficiency is based on the FLT, but we also a SLT-based efficiency which takes into account the limitations imposed by the energy efficiency.

Consider a control volume, and let's do an analogy between the 2 kinds of efficiency

Energy in = Energy output in product + Energy emitted in waste

Exergy in = Exergy output in product + Exergy emitted in waste + Exergy destruction

The efficiency from energy or exergy viewpoints;

➕ Energy efficiency

$$\eta_{en} = \frac{Energy\ output\ in\ product}{Energy\ input} = 1 - \frac{Energy\ loss}{Energy\ input}$$

➕ Exergy efficiency

$$\eta_{ex} = \frac{Exergy\ output\ in\ product}{Exergy\ input} = 1 - \frac{Exergy\ loss}{Exergy\ input} = 1 - \frac{(exergy\ waste\ emission\ +\ Exergy\ destruction)}{exergy\ input}$$
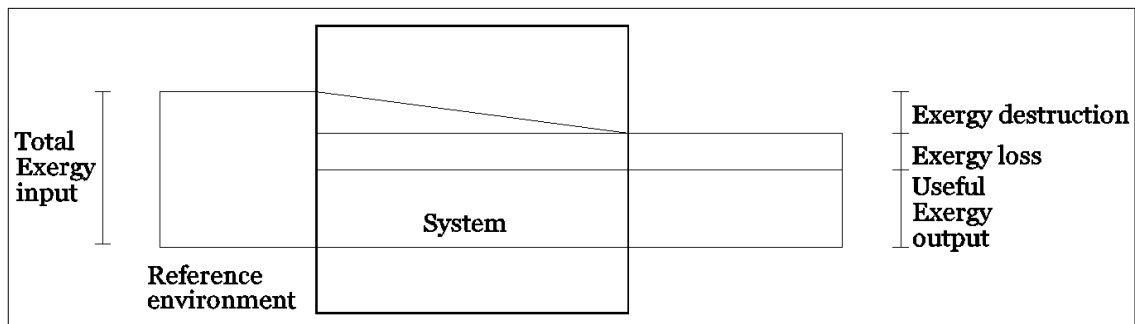
Figure 3: Exergy balance in a system

Exergy analysis

Exergy analysis is a method that uses the conservation of mass and conservation of energy principles together with the second law of thermodynamics for the analysis, design and improvement of energy and other systems. The exergy method is a useful tool for furthering the goal of more efficient energy-resource use, for it enables the locations, types and magnitudes of wastes and losses to be identified and meaningful efficiencies to be determined.

Bejan [6] defines exergy or availability analysis involving the simultaneous employment of the first and second laws of thermodynamics for the purpose of analyzing the performance of thermodynamic processes in the reversible limit and for estimating the departure from this limit (irreversibility).

Exergy analysis for a thermodynamic cycle and a hot flow

We can know the exergy efficiency of each component i in the cycle by knowing the exergy destruction in this component:

$$\eta_{ex,i} = 1 - \frac{Ex_{d,i}}{Ex_{a,i}}$$

Where $Ex_{d,i}$ and $Ex_{a,i}$ are the destructed and available exergy respectively at the element i

Also, we can know the global exergy efficiency of the system from the total destructed exergy in the cycle and the flow and the available exergy in the hot exhaust flow

$$\eta_{ex,g} = 1 - \frac{Ex_{d,g}}{Ex_{a,f}}$$

Where $Ex_{d,g}$ is the exergy destruction in the cycle and the hot flow

$$Ex_{d,g} = \sum_i Ex_{d,i} + Ex_{d,f}$$

And $Ex_{d,f}$ is the destructed exergy in the hot flow and $Ex_{a,f}$ is the available exergy in the hot flow.

Chapter 3: Exergy analysis of producing 'cooling effect' from the predefined hot exhaust using the 'Ejector refrigeration cycle'

Ejector refrigeration is a thermally driven technology that has been used for cooling applications for many years. In their present state of development they have a much lower COP than vapor compression systems but offer advantages of simplicity and no moving parts. Their greatest advantage is their capability to produce refrigeration using waste heat.

The first steam ejector refrigeration system was developed by Maurice Leblanc in 1910 and gained in popularity for air conditioning applications until the development of chlorofluorocarbon refrigerants in the 1930's and their use in the vapor compression cycle. Research and development continued however and the ejector technology found applications in many engineering fields particularly in the chemical and process industries.

The basic components of this system are: ejector, condenser, boiler, evaporator, expansion valve, and circulation pump.

Description of Basic Ejector refrigeration cycle

The system consists of two loops, the power loop and the refrigeration loop. In the power loop, low-grade heat, Q_b, (Q_b is part of the waste heat), is used in a boiler or generator to evaporate high pressure liquid refrigerant (process 1-2). The high pressure vapor generated, known as the primary fluid, flows through the ejector where it accelerates through the nozzle. The reduction in pressure that occurs induces vapor from the evaporator, known as the secondary fluid, at point 3. The two fluids mix in the mixing chamber before entering the diffuser section where the flow decelerates and pressure recovery occurs. The mixed fluid then flows to the condenser where it is condensed rejecting heat to the environment, Q_c. A portion of the liquid exiting the condenser at point 5 is then pumped to the boiler for the completion of the power cycle. The remainder of the liquid is expanded through an expansion device and enters the evaporator of the refrigeration loop at point 6 as a mixture of liquid and vapor. The refrigerant evaporates in the evaporator producing a refrigeration effect, Q_e, and the resulting vapor is then drawn into the ejector at point 3. The refrigerant (secondary fluid) mixes with the primary fluid in the ejector and is compressed in the diffuser section before entering the condenser at point 4. The mixed fluid condenses in the condenser and exits at point 5 for the repetition of the refrigeration cycle.

Little research work has been done to identify the effect of the refrigerant type on the performance of the cycle [7]. The results show that using water as refrigerant has very low COP values; the system using R152a (hydro-fluoro-carbon HFC; 1,1-difluoroethane) as refrigerant has better performance and the variation in COP values for various refrigerants is almost independent of system operating conditions [7].
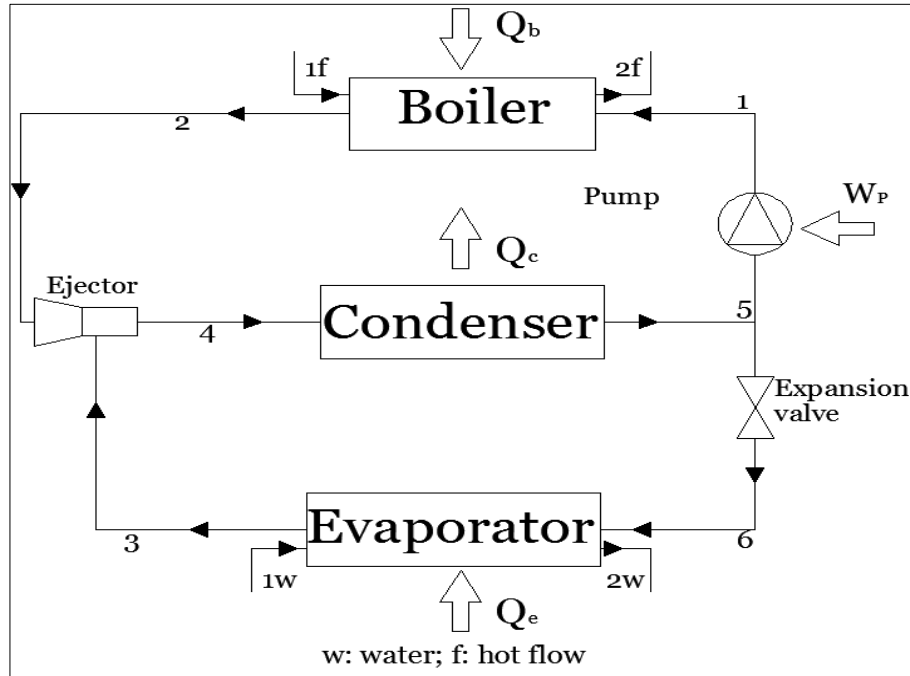
Figure 4: Basic ejector refrigeration system

The key component in the ejector refrigeration system is the ejector [7]; the cycle performance was very sensitive to the ejector efficiency [8] and ejector entrainment ratio [7]; the performance of the ejector is measured by its entrainment ratio, ω, which is the mass flow rate of the secondary flow to the drive flow:

$$\omega = \dot{m}_e / \dot{m}_b$$

Where $\dot{m}_e$ and $\dot{m}_b$ are mass flow rate from the evaporator and the boiler respectively

The structure of the ejector is characterized by the area ratio, ζ, which is defined as the cross-section area of the constant area section divided by that of the primary nozzle throat.

$$\zeta = A_m / A_t$$

Where A_m and A_t are the cross-sectional area of the constant area section and the convergent nozzle throat.

The ejector behaves as a turbine in series with a compressor, so we can adopt this analogy in the thermodynamic cycle.
The experiment of Da-Wen Sun [7] shows that R152a has the best performance among the most common refrigerants, at some operating conditions (T_{boiler}=90 ^{O}C, $T_{condeser}$=35 ^{O}C, $T_{evaporater}$= 5 ^{O}C) and ejector performance equals to 0.85, it has ω=0.2325, ζ=4.08
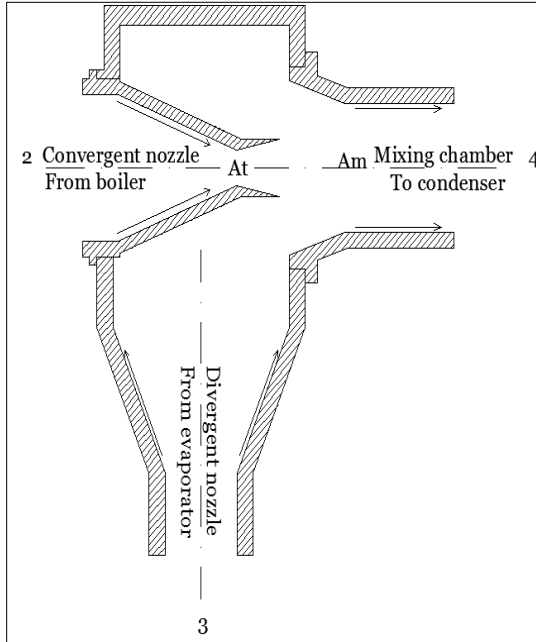
Figure 6: ejector structure
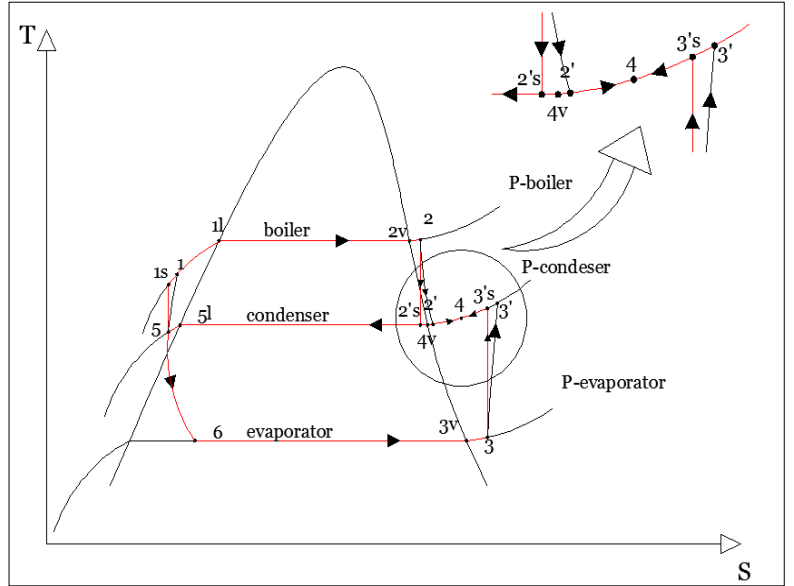
Figure 5: Basic ejector refrigeration system T-S cycle

Exergy analysis of the hot exhaust and the above mentioned system

We will work in the cycle according to the following conditions:

- The ejector is modeled by a turbine, mixing chamber and a compressor
- The used refrigerant is R152a
- Temperature of the boiler is function of its pressure
- Isentropic efficiency of the compressor $\eta_c=0.85$
- Isentropic efficiency of the turbine $\eta_t=0.85$
- Isentropic efficiency of the pump $\eta_p=0.85$
- Reference temperature To $=25\ ^{\text{O}}\text{C}$
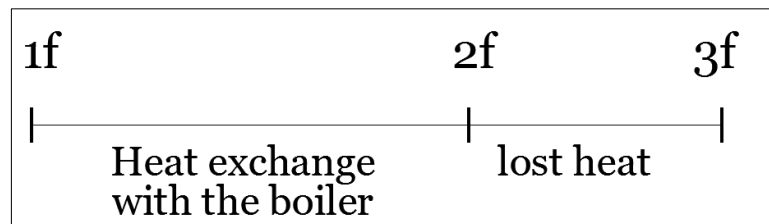
Figure 7: Heat exchange with the hot flow profile

We know that the exergy balance for an open system in the steady state is:

$$(\text{Exergy input}) - (\text{Exergy destructed}) = (\text{Exergy output})$$

So we will do the energy and exergy balance for each component in the cycle, calculating the exergy destruction rate at each component (internal irreversibilities) in the cycle and finally with the destructed exergy (external irreversibilities) in the hot flow we will get the global exergy efficiency in the whole system (cycle and hot flow)

Mass flow rate in the boiler $\dot{m}_b$
Mass flow rate in the evaporator $\dot{m}_e$
Mass flow rate in the condenser $\dot{m}_c = \dot{m}_b + \dot{m}_e$

Pump
✚ Energy balance

$$\dot{W}_p + \dot{m}_b(h_5 - h_1) = 0$$

The isentropic efficiency of the pump is the ratio of the ideal mechanical power and the actual power

$$\eta_{s,p} = \dot{m}_b \frac{h_{1s} - h_5}{\dot{W}_P}$$

Where h_{1s} is the enthalpy of the ideal state of point 1

Then

$$h_1 = \frac{h_{1s} - h_5}{\eta_{s,p}} + h_5$$

✚ Exergy balance

$$\dot{W}_p + \dot{m}_b(ex_5 - ex_1) - \dot{Ex}_{d,p} = 0$$

$$\dot{I}_{pump} = \dot{Ex}_{d,p} = \dot{W}_p + \dot{m}_b(ex_5 - ex_1)$$

By substituting the energy balance and work equations in the exergy balance equation; the irreversibility rate can be written as:

$$\dot{I}_p = \dot{Ex}_{d,p} = \dot{m}_b T_o(s_1 - s_5)$$

Boiler
✚ Energy balance

$$\dot{m}_f(h_{1f} - h_{2f}) + \dot{m}_b(h_1 - h_2) = 0$$

✚ Exergy balance

$$\dot{m}_f\left(ex_{1f} - ex_{2f}\right) + \dot{m}_b(ex_1 - ex_2) - \dot{Ex}_{d,b} = 0$$

By substituting the energy balance equation in the exergy balance equation; then the irreversibility rate can be written as:

$$\dot{I}_b = Q_b\, T_o\left[\frac{\Delta s_{12}}{\Delta h_{12}} - \frac{\Delta s_{1f2f}}{\Delta h_{1f2f}}\right]$$

Here we introduce the equivalent temperature of a transformation definition:
The equivalent temperature $\widetilde{T}_{12}$ and the Carnot factor $\tilde{\theta}_{12}$ of the transformation (1 to 2)

$$\widetilde{T}_{12} = \frac{\Delta h_{12}}{\Delta s_{12}} \qquad \tilde{\theta}_{12} = 1 - \frac{T_o}{\widetilde{T}_{12}}$$

The equivalent temperature $\widetilde{T}_{1f\,2f}$ and the Carnot factor $\tilde{\theta}_{1f\,2f}$ of the transformation (1f to 2f)

$$\widetilde{T}_{1f2f} = \frac{\Delta h_{1f2f}}{\Delta s_{1f2f}} \qquad \tilde{\theta}_{1f2f} = 1 - \frac{T_o}{\widetilde{T}_{1f2f}}$$

Then the irreversibility rate becomes

$$\dot{I}_b = \dot{Ex}_{d,b} = Q_b\left(\tilde{\theta}_{1f2f} - \tilde{\theta}_{12}\right)$$

Turbine
🞧 Energy balance
We can obtain the entrainment ratio of the cycle by proposing that the energy lost in the nozzle is equal to the energy gained in the compressor

$$\dot{m}_b(h_2 - h_{2'}) = \dot{m}_e(h_{3'} - h_3)$$

If we assume that there is no heat exchange with the environment then

$$\dot{m}_b(h_2 - h_{2'}) + \dot{m}_b\frac{\left(v_2^2 - v_1^2\right)}{2} = 0$$

And

$$\eta_{s,n} = \frac{h_2 - h_{2'}}{h_2 - h_{2's}} \qquad then \quad h_{2'} = (h_{2's} - h_2)\eta_{s,n} + h_2$$

🞧 Exergy balance

$$\dot{m}_b(ex_2 - ex_{2'}) - \dot{Ex}_{d,nozzle} = 0$$

By substituting the energy balance equation in the exergy balance equation; then the irreversibility rate can be written as:

$$\dot{I}_{nozzle} = \dot{Ex}_{d,nozzle} = \dot{m}_b T_o(s_{2'} - s_2)$$

Mixing chamber
  ➕ Energy balance

$$\dot{m}_e h_{3'} + \dot{m}_b h_{2'} - (\dot{m}_e + \dot{m}_b)h_4 = 0$$

  ➕ Exergy balance

$$\dot{m}_e ex_{3'} + \dot{m}_b ex_{2'} - (\dot{m}_e + \dot{m}_b)ex_4 - \dot{Ex}_{d,m} = 0$$

By substituting the energy balance equation in the exergy balance equation; then the irreversibility rate can be written as:

$$\dot{I}_m = \dot{Ex}_{d,m} = (\dot{m}_e + \dot{m}_b)T_o s_4 - \dot{m}_e T_o s_{3'} - \dot{m}_b T_o s_{2'}$$

Compressor
  ➕ Energy balance

$$\dot{m}_e(h_3 - h_{3'}) + \dot{m}_e \frac{(v_3^2 - v_{3'}^2)}{2} = 0$$

And

$$\eta_{s,c} = \frac{h_{3's} - h_3}{h_{3'} - h_3} \quad then \quad h_{3'} = \frac{(h_{3's} - h_3)}{\eta_{s,c}} + h_3$$

  ➕ Exergy balance

$$\dot{m}_e(ex_3 - ex_{3'}) - \dot{Ex}_{d,comp} = 0$$

By substituting the energy balance equation in the exergy balance equation; then the irreversibility rate can be written as:

$$\dot{I}_{comp} = \dot{Ex}_{d,comp} = \dot{m}_e T_o(s_{3'} - s_3)$$

Condenser
  ➕ Energy balance

$$-Q_c + \dot{m}_c(h_4 - h_5) = 0$$

Exergy balance

$$-Q_c\left(1 - \frac{T_o}{T}\right) + \dot{m}_c(ex_4 - ex_5) - \dot{Ex}_{d,cond} = 0$$

As $Q_c\left(1 - \frac{T_o}{T}\right)$ tends to zero;

$$\dot{Ex}_{d,cond} = \dot{m}_c(ex_4 - ex_5)$$

By substituting the energy balance equation in the exergy balance equation; then the irreversibility rate can be written as:

$$\dot{I}_{cond} = \dot{Ex}_{d,cond} = Q_c\left[1 - T_o\frac{\Delta s_{45}}{\Delta h_{45}}\right]$$

The equivalent temperature $\widetilde{T}_{45}$ and the Carnot factor $\widetilde{\theta}_{12}$ of the transformation 4 to 5

$$\widetilde{T}_{45} = \frac{\Delta h_{45}}{\Delta s_{45}} \qquad \widetilde{\theta}_{45} = 1 - \frac{T_o}{\widetilde{T}_{45}}$$

Then the irreversibility rate becomes

$$\dot{I}_{cond} = \dot{Ex}_{d,cond} = Q_c\left[1 - \frac{T_o}{\widetilde{T}_{45}}\right] = Q_c\widetilde{\theta}_{45}$$

Expansion valve
Energy balance

$$\dot{m}_e(h_5 - h_6) = 0$$

Exergy balance

$$\dot{m}_e(ex_5 - ex_6) - \dot{Ex}_{d,exp} = 0$$

By substituting the formula of specific exergy and the energy balance then

$$\dot{I}_{exp} = \dot{Ex}_{d,exp} = \dot{m}_e T_o(s_6 - s_5)$$

Evaporator
Energy balance

$$\dot{m}_{cw}(h_{1w} - h_{2w}) + \dot{m}_e(h_6 - h_3) = 0$$

Where $\dot{m}_{cw}$ is the mass flow rate of the water to be cooled

➕ Exergy balance

$$\dot{m}_{cw}(ex_{1w} - ex_{2w}) + \dot{m}_e(ex_6 - ex_3) - \dot{Ex}_{d,evap} = 0$$

Here we introduce the concepts of "warm exergy" and "cold exergy", when the temperature of the system is greater than the reference temperature, we call the heat exergy in the system warm exergy, and when the temperature of the system is lower than the reference temperature, we call the heat exergy in the system cold exergy, as in our case here $(1 - \frac{T_o}{\widetilde{T}_{63}})$ is smaller than zero $(T_o > \widetilde{T}_{63})$, and the system is losing exergy.

Note that the exergy, unlike energy, flows in one direction only, to the side of the ambient environment.

By substituting the energy balance equation in the exergy balance equation; then the irreversibility rate can be written as:

$$\dot{I}_{evap} = \dot{Ex}_{d,evap} = Q_e T_o \left[\frac{\Delta s_{63}}{\Delta h_{63}} - \frac{\Delta s_{1w2w}}{\Delta h_{1w2w}} \right]$$

Where Q_e is the heat transferred from water to the evaporator

The equivalent temperature and the Carnot factor $\widetilde{\theta}_{63}$ of the transformation (6 to 3)

$$\widetilde{T}_{63} = \frac{\Delta h_{63}}{\Delta s_{63}} \qquad \widetilde{\theta}_{63} = 1 - \frac{T_o}{\widetilde{T}_{63}}$$

The equivalent temperature and the Carnot factor $\widetilde{\theta}_{1w2w}$ of the transformation (1w to 2w)

$$\widetilde{T}_{1w2w} = \frac{\Delta h_{1w2w}}{\Delta s_{1w2w}} \qquad \widetilde{\theta}_{1w2w} = 1 - \frac{T_o}{\widetilde{T}_{1w2w}}$$

Then the irreversibility rate becomes

$$\dot{I}_{evap} = \dot{Ex}_{d,evap} = Q_e \left(\widetilde{\theta}_{1a2a} - \widetilde{\theta}_{63} \right)$$

Finally the internal exergy destruction of the global system is the sum of the individual exergy destruction at each component:

$$\sum_i \dot{Ex}_{d,i}$$

$$\sum_i \dot{Ex}_{d,i} = \dot{I}_P + \dot{I}_b + \dot{I}_{nozzle} + \dot{I}_m + \dot{I}_{comp} + \dot{I}_{cond} + \dot{I}_{exp} + \dot{I}_{evap}$$

The exergy loss in the hot flow is between 2f and downstream 3f, that is $\dot{Ex}_{2f3f}$, and the available exergy in the flow is $\dot{Ex}_{a,f}=\dot{Ex}_{1f3f} + \dot{Ex}_{pump}$, thus we can calculate the global exergy efficiency of the system. With the help of the REFPROP [9] software, working fluid properties are calculated.

Results of the exergy analysis of the ejector refrigeration cycle

The table below shows the different properties of the ejector refrigeration cycle at almost all the points that can reflect the behavior of the cycle according to some specific operating conditions

Point	Temperature (^{O}C)	Pressure (bar)	h (KJ/Kg)	s (KJ/Kg K)	Quality
1f	165	25	489.78	6.3910	-
2f	30	25	343.23	6.0113	-
1	25.07	40	244.52	1.1419	-0.00027
1L	106.89	40	426	1.6723	0
2	109.89	40	541.23	1.9756	1.57707
2v	109.89	40	526.34	1.9363	1
2'	26	6.14	497.84	2.0012	0.90713
3	8	3.14	513.85	2.1283	1.12175
3v	5	3.14	510.48	2.1163	1
3'	39.07	6.14	539.99	2.1409	1.57274
4	26.39	6.14	524.16	2.0892	1.01411
4v	26	6.14	523.67	2.0876	1
5	23	6.14	240.14	1.1397	-4.139E-06
5L	26	6.14	245.53	1.1578	0
6	5	3.14	240.14	1.1444	0.10463

*for P(boiler)=40 bar, T(evaporator)=5°C, T(condenser)=26°C, T(overheating)=3°C, T(overcooling)=3°C, T(overheating-boiler)=3°C, global efficiency=35.2%

Table 6: State properties of the ejector cycle at some specific operating conditions

The pie chart below shows the contribution of the different components in the cycle in the overall irreversibility rate according to the specific conditions in the preceding table
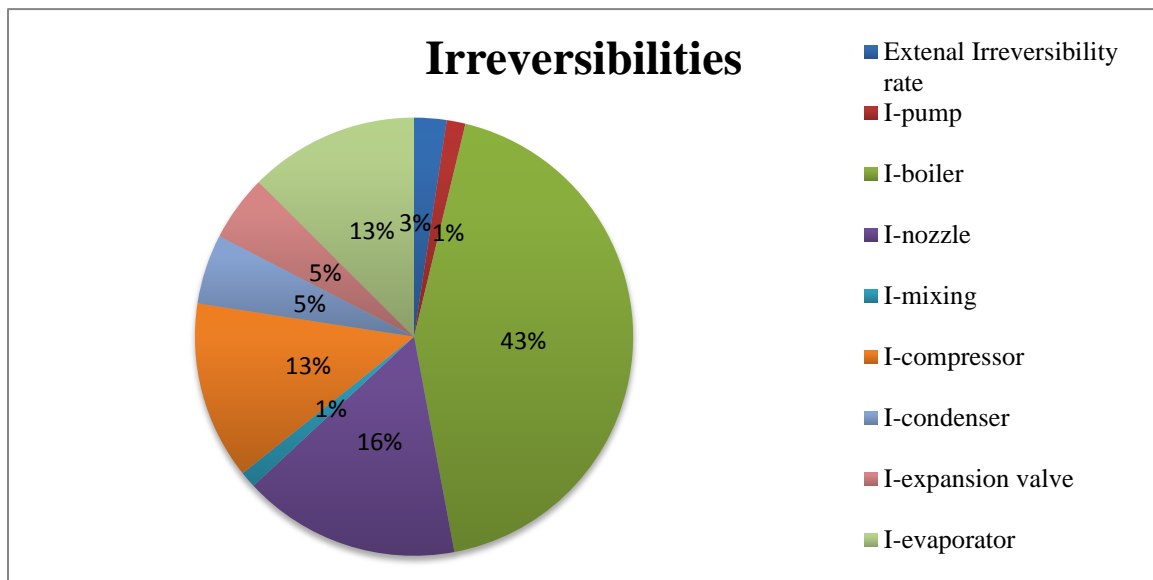
Figure 8:Contribution of each component in the overall irreversibility rate

Figure 9 shows the effect of the temperature of the hot flow at the end of heat exchange on the global efficiency, it shows clearly that the global exergy efficiency decreases when the temperature increases by a slope of 0.00254 %/ °C
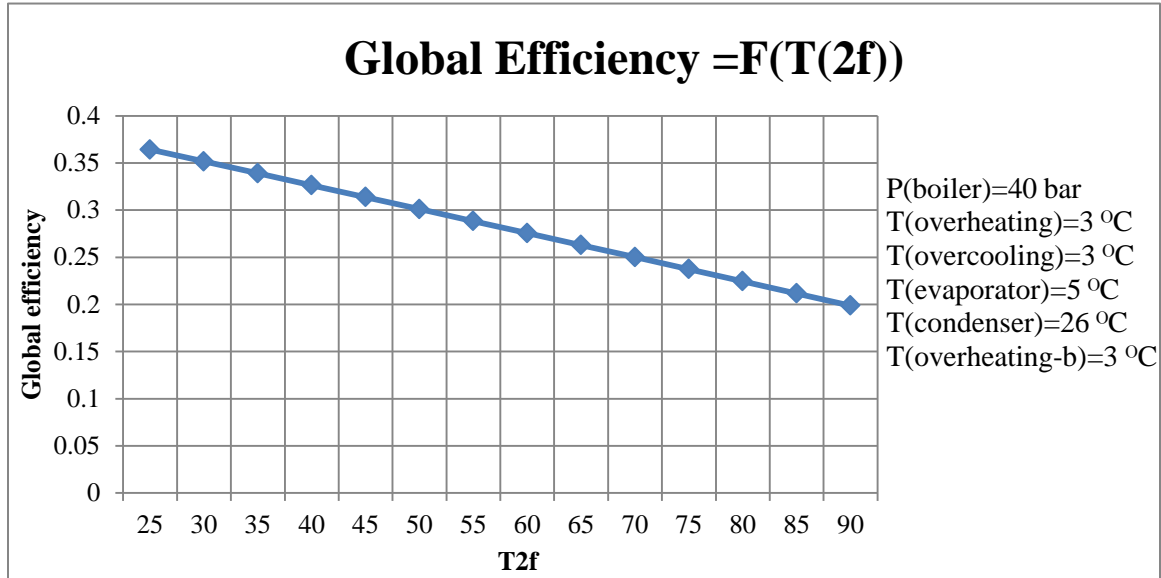
Figure 9: influence of the hot stream at the end of the heat exchange on the global exergy efficiency

Figure 10 shows the effect of the boiler pressure on the global exergy efficiency of the cycle, figure 10 shows clearly that the global exergy efficiency increases with the increase of boiler pressure in a parabolic way

Figure 10: influence of boiler pressure on the global exergy efficiency

Figure 11 shows the effect of the overheating temperature on the global exergy efficiency of the cycle, this figure shows that the global exergy efficiency decreases with the increase of overheating temperature in the evaporator also in a parabolic behavior
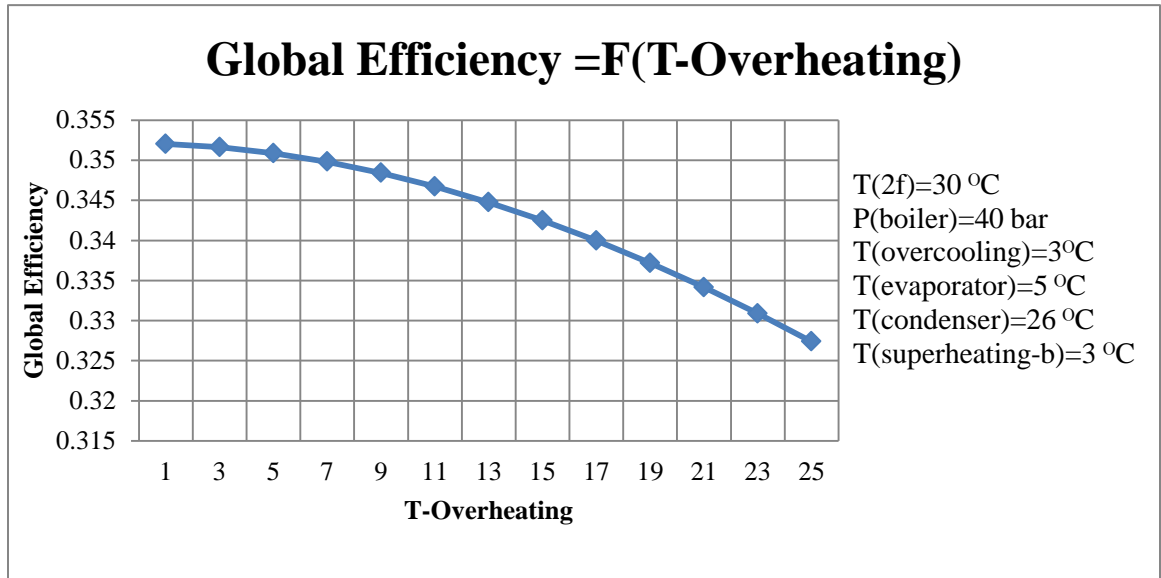
Figure 11: Influence of the oveheating on the global exergy efficiency

Figure 12 shows the effect of the overcooling temperature on the global exergy efficiency of the cycle, it shows that the global exergy efficiency increases with the increase of overcooling in the boiler by a slope of 0.00034 %/ ^{O}C
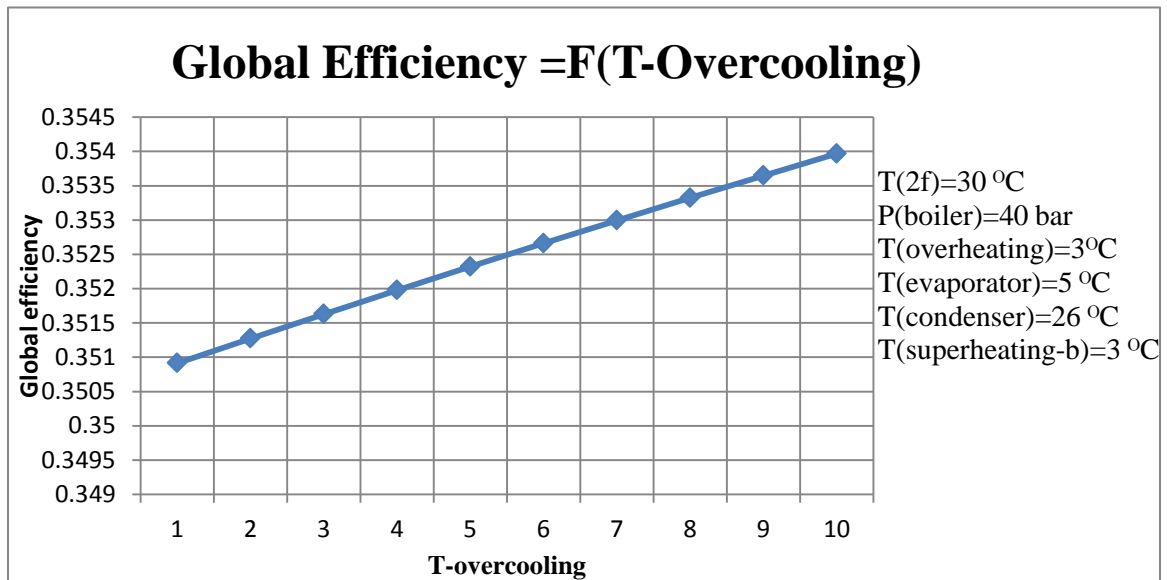
Figure 12: Influence of the ovecooling on the global exergy efficiency

Figure 13 shows the effect of the condenser temperature on the global exergy efficiency of the cycle, in figure 13 we can notice that the global exergy efficiency decreases with the increase of condenser temperature
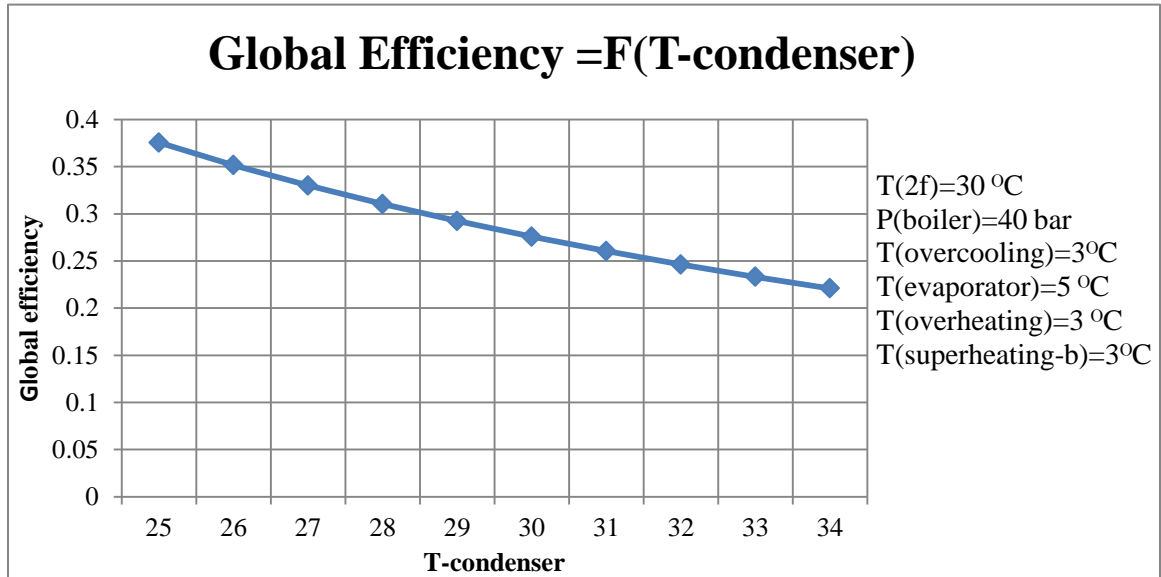
Figure 13: Influence of the condenser temperature on the global exergy efficiency

Figure 14 shows the effect of the evaporator temperature on the global exergy efficiency of the cycle, also the global exergy efficiency decreases with the increase of evaporator temperature
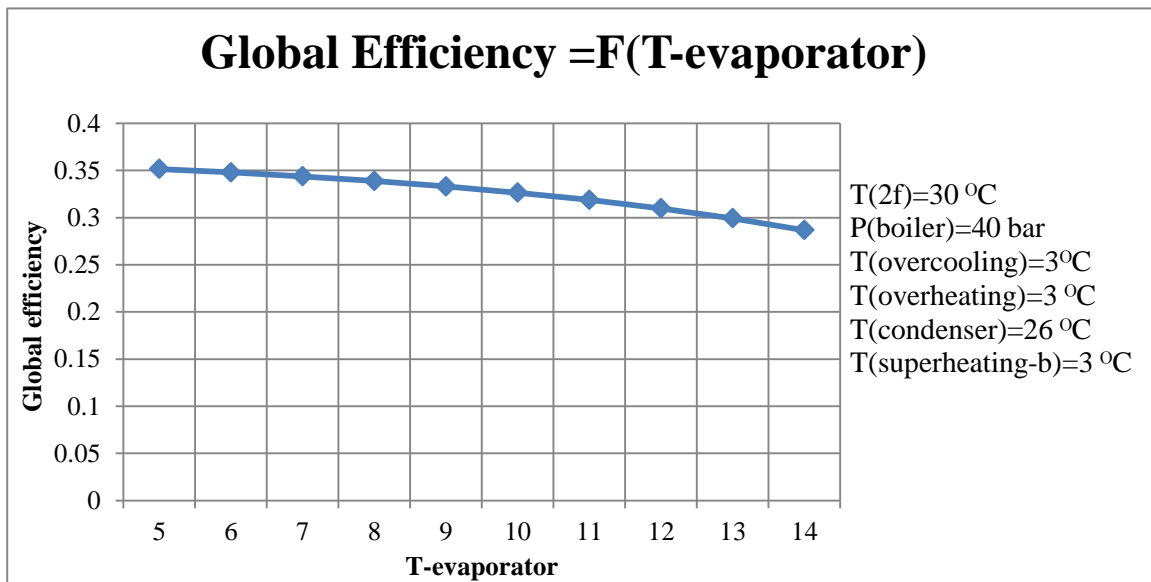
Figure 14: Influence of the evaporator temperature on the global exergy efficiency

Figure 15 shows the effect of the boiler overheating temperature on the global exergy efficiency of the cycle, this figure shows clearly that the global exergy efficiency increases with the increase of the boiler overheating temperature
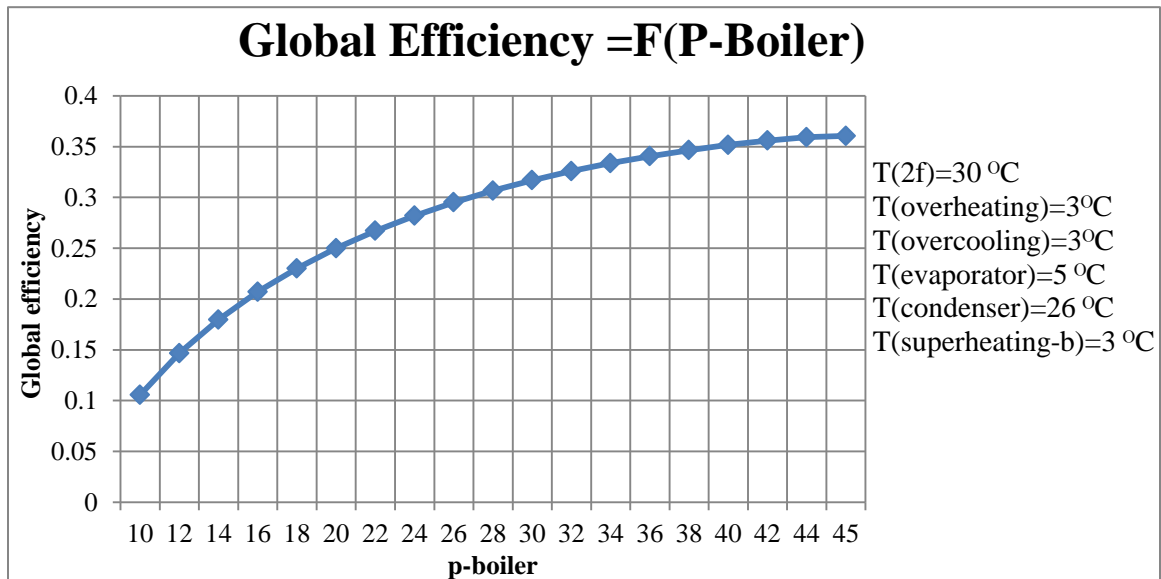
Global efficiency=F(boiler overheating)

T(2f)=30 $^{\circ}$C
P(boiler)=40 bar
T(overcooling)=3 $^{\circ}$C
T(overheating)=3 $^{\circ}$C
T(condenser)=26 $^{\circ}$C
T(evaporator)=5 $^{\circ}$C
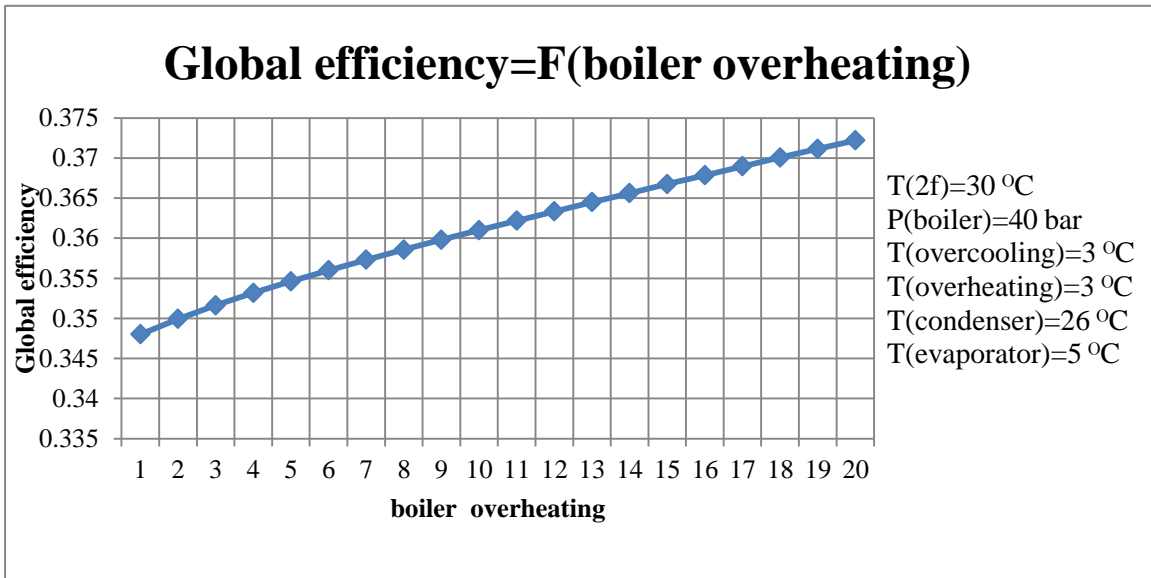
Figure 15: Influence of the boiler overheating temperature on the global exergy efficiency

Chapter 4: Exergy analysis of producing 'cooling effect' from the predefined hot exhaust using the 'absorption cycle'

Absorption cycle

Absorption technology has been applied to the air-conditioning and process cooling/heating for 150 years. Absorption cooling is an example of a sustainable solution where waste heat may be used to produce cooling. By using waste heat for cooling production the electricity need for compression cooling can be reduced. In addition, reduced electricity consumption could lower the global CO_2 emissions, if the electricity is produced by fossil based fuels. This environmental aspect is a heavy incentive for the use of heat-driven cooling.

Absorption cooling systems use heat, instead of mechanical energy, to provide cooling. The mechanical vapor compressor is replaced by a thermal compressor that consists of an absorber, a generator, a pump, and a throttling device. The refrigerant vapor from the evaporator is absorbed by a solution mixture in the absorber. This solution is then pumped to the generator where the refrigerant is re-vaporized using a waste steam heat source. The refrigerant-depleted solution is then returned to the absorber via a throttling device. The two most common refrigerant/absorbent mixtures used in absorption chillers are water/lithium bromide and ammonia/water.
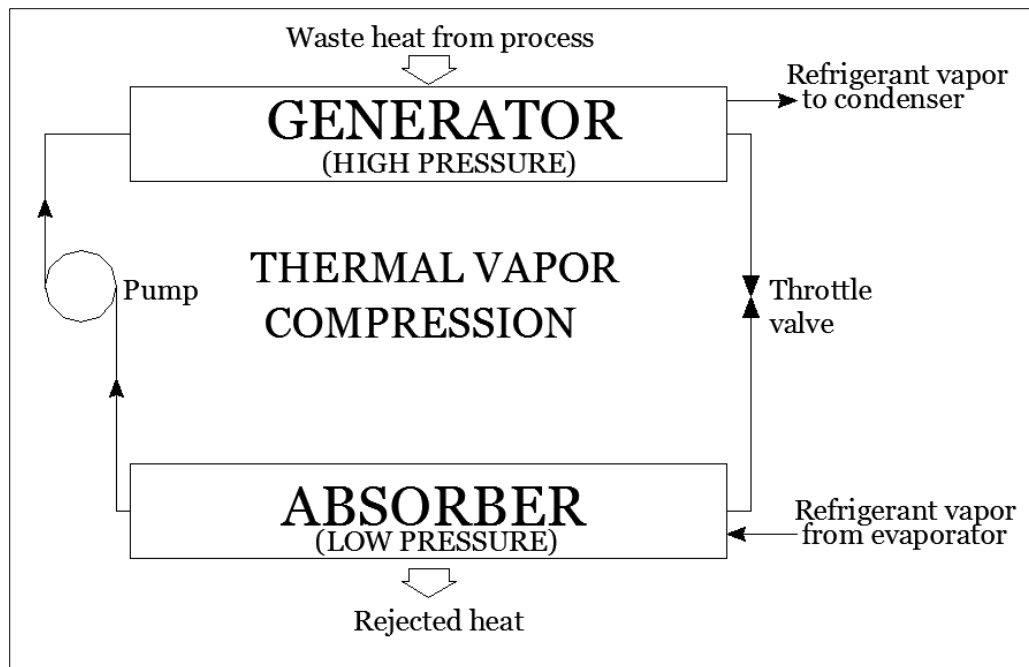
Figure 16: Thermal vapor compression in the absorption cycle

Vapor compression chillers are the most commonly used chiller type in the world today. A potential disadvantage with the absorption chillers is the low coefficient of performance, COP, which lies somewhere between 0.6-0.8. This should be compared to the COP of vapor compression chillers', which varies between 2 and 5 depending on the temperature lift.

Another disadvantage for the absorption chiller is the need for external cooling to cool the absorber and the condenser. This need is much larger per unit cooling effect in absorption chillers as compared to the need in compression chillers. If there is no natural heat sink close by, such as a lake or a river, the extra cooling tower capacity required for absorption cooling will result in a higher investment cost for this technology.

Despite these disadvantages absorption cooling has the potential to compete with compression cooling, both economically and environmentally. When using absorption chillers instead of compression chillers the consumption of electricity may be reduced. If cheap driving heat is available for the absorption chiller, it can become a more economical alternative. Also, if waste heat or heat from a renewable source is used, the CO_2 emissions will be lowered by the reduced electricity consumption. In addition, since water can be used as the refrigerant in absorption chillers, the problem with the environmentally harmful refrigerants used in vapor compression chillers is avoided.

Figure 17: Single-effect absorption cycle

Figure 18 shows the different processes in the absorption cooling cycle expressing at which absorbent concentration these processes is happened
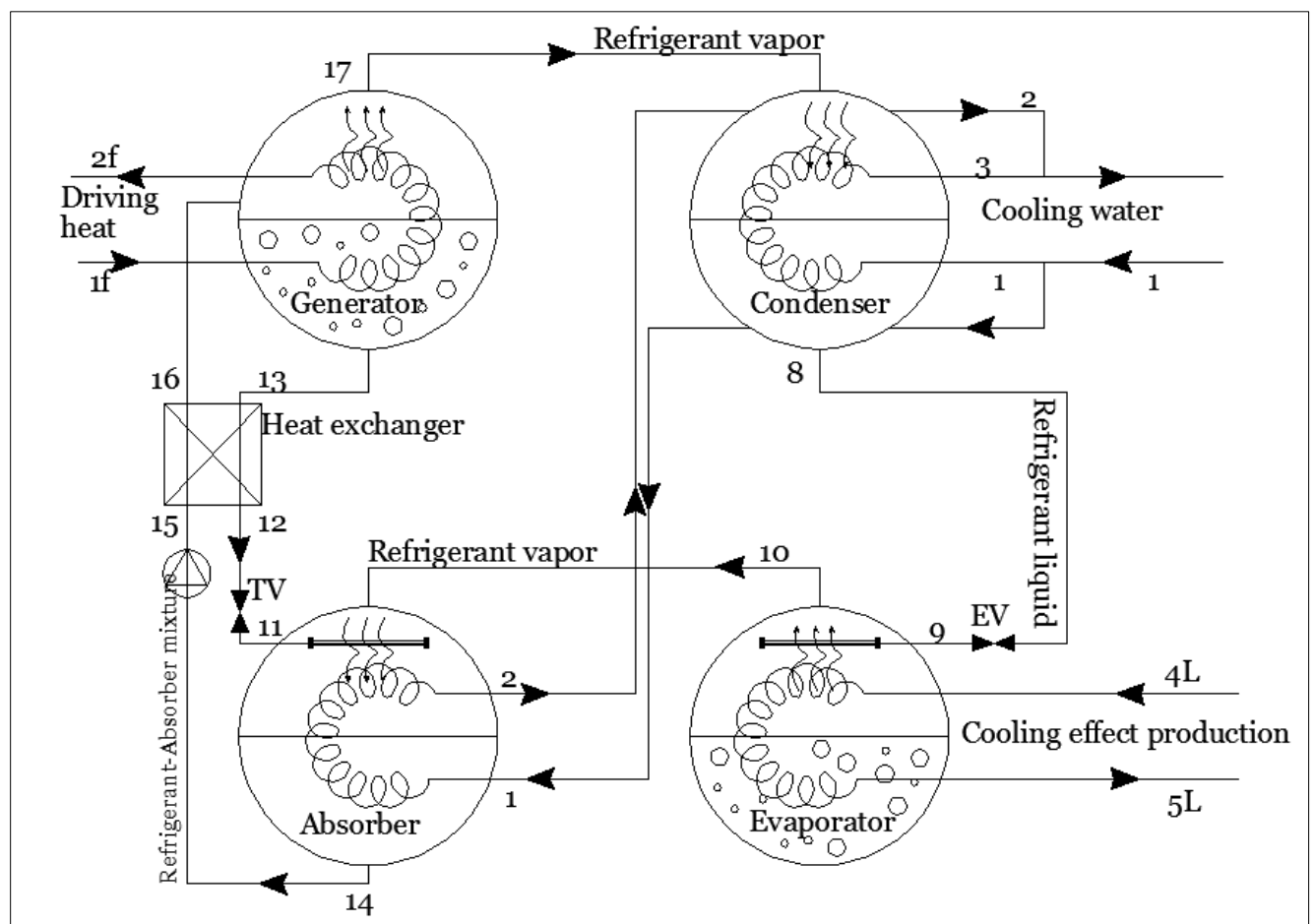
1-2: The weak solution from the absorber at point 1 is pumped through the heat exchanger to the generator; point 2 indicates the properties of the solution at the outlet of the heat exchanger. During the process 1-2, the concentration of the weak solution is held constant

2-3: shows the sensible heating of the weak solution in the generator

3-4: indicates the boiling of water vapor from the solution at the constant condensing pressure P_c (although the boiling pressure is little higher than the condensing pressure, the difference is negligible). During this process, the weak solution becomes strong solution

Te, Ta, Tc, Tg, Pe, Pc, x1, and x2 are the evaporator temperature, absorber temperature, condenser temperature, generator temperature, evaporator pressure, condenser pressure, and the mass fraction of the absorbent in the weak solution and the strong solution respectively
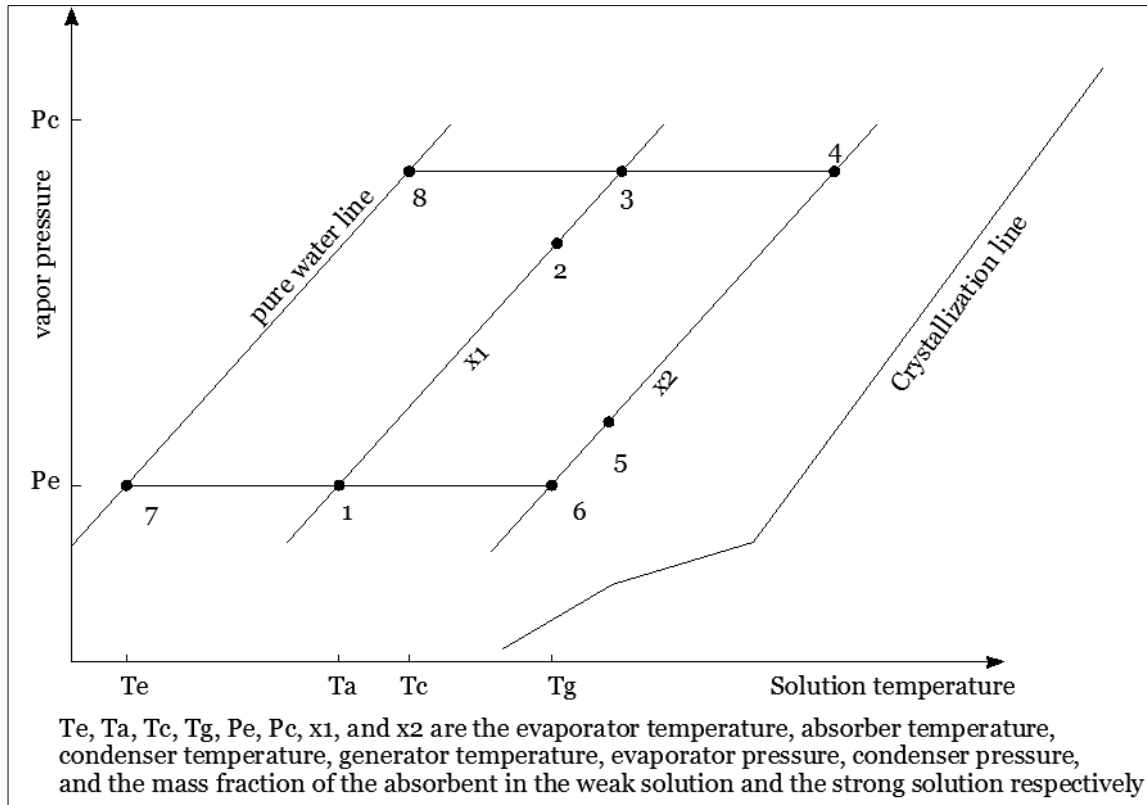
Figure 18: Working solution temperature-vapor pressure profile

4-5: the strong solution passing to the absorber through the heat exchanger, in which it preheats the weak solution flowing from the absorber to the generator. During this process, the concentration of the strong solution is constant

5-6-1: Indicates the idealized process of absorption of water vapor from the evaporator by the strong solution in the absorber

3-8: Denotes the condensation of water vapor in the condenser by the cooling water from the cooling tower, at constant condensing pressure P_c

8-7: Shows the flow of condensed water from the condenser to the evaporator

7-1: Indicates the evaporation of the water in the evaporator due to the prevailing low pressure P_e

Also the water absorbs the heat from the space to be cooled. The water vapor from the evaporator is in turn absorbed by the strong solution in the absorber, thus completing the cycle of refrigeration.

Exergy analysis of the hot exhaust and the above mentioned system

```
1f                          2f              3f

|————————————————————————————|———————————————|
    Heat exchange                  lost heat
    with the generator
```
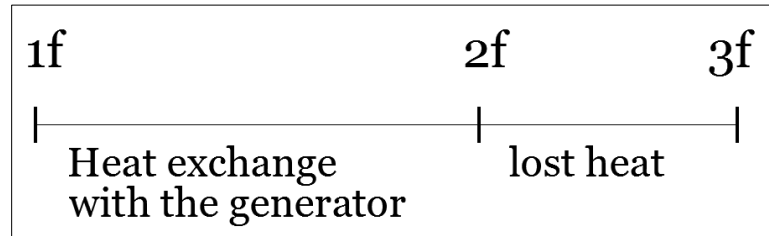
Figure 19: Heat exchange with the hot flow profile

Generator
➕ Energy balance

$$\dot{m}_f(h_{1f} - h_{2f}) + \dot{m}_s h_{16} - \dot{m}_w h_{13} - \dot{m}_r h_{17} = 0$$

$$\dot{m}_s = \dot{m}_w + \dot{m}_r$$

➕ Exergy balance

$$\dot{m}_f(ex_{1f} - ex_{2f}) + \dot{m}_s ex_{16} - \dot{m}_w ex_{13} - \dot{m}_r ex_{17} - \dot{Ex}_{d,gen} = 0$$

$$\dot{I}_{gen} = \dot{Ex}_{d,gen} = T_o\left(\dot{m}_r s_{17} + \dot{m}_w s_{13} - \dot{m}_s s_{16} - \dot{m}_f(s_{1f} - s_{2f})\right)$$

Absorber
➕ Energy balance

$$\dot{m}_{acw}(h_1 - h_2) + \dot{m}_w h_{11} - \dot{m}_s h_{14} + \dot{m}_r h_{10} = 0$$

➕ Exergy balance

$$\dot{m}_{acw}(ex_1 - ex_2) + \dot{m}_w ex_{11} - \dot{m}_s ex_{14} + \dot{m}_r ex_{10} - \dot{Ex}_{d,abs} = 0$$

$$\dot{I}_{abs} = \dot{Ex}_{d,abs} = T_o\left(\dot{m}_s s_{14} - \dot{m}_w s_{11} - \dot{m}_r s_{10} + \dot{m}_{acw}(s_2 - s_1)\right)$$

Heat exchanger
The effectiveness of the heat exchanger is

$$\varepsilon = \frac{T_{13} - T_{12}}{T_{13} - T_{15}}$$

➕ Energy balance

$$\dot{m}_w(h_{13} - h_{12}) + \dot{m}_s(h_{15} - h_{16}) = 0$$

Exergy balance

$$\dot{m}_w(ex_{13} - ex_{12}) + \dot{m}_s(ex_{15} - ex_{16}) - \dot{E}x_{d,HE} = 0$$

$$\dot{I}_{HE} = \dot{E}x_{d,HE} = T_o\big(\dot{m}_w(s_{12} - s_{13}) + \dot{m}_s(s_{16} - s_{15})\big)$$

Throttling valve
Energy balance

$$\dot{m}_w h_{12} - \dot{m}_s h_{11} = 0$$

Exergy balance

$$\dot{m}_w ex_{12} + \dot{m}_s ex_{11} - \dot{E}x_{d,TV} = 0$$

$$\dot{I}_{TV} = \dot{E}x_{d,TV} = \dot{m}_w T_o(s_{11} - s_{12})$$

Condenser
Energy balance

$$\dot{m}_{ccw}(h_1 - h_3) + \dot{m}_r(h_{17} - h_8) = 0$$

Exergy balance

$$\dot{m}_{ccw}(ex_1 - ex_3) + \dot{m}_r(ex_{17} - ex_8) - \dot{E}x_{d,cond} = 0$$

$$\dot{I}_{cond} = \dot{E}x_{d,cond} = T_o\big(\dot{m}_{ccw}(s_3 - s_1) + \dot{m}_r(s_8 - s_{17})\big)$$

Evaporator
Energy balance

$$\dot{m}_{cw}(h_{4L} - h_{5L}) + \dot{m}_r(h_9 - h_{10}) = 0$$

Exergy balance

$$\dot{m}_{cw}(ex_{4L} - ex_{5L}) + \dot{m}_r(ex_9 - ex_{10}) - \dot{E}x_{d,evap} = 0$$

$$\dot{I}_{evap} = \dot{E}x_{d,evap} = T_o\big(\dot{m}_{cw}(s_{5L} - s_{4L}) + \dot{m}_r(s_{10} - s_9)\big)$$

Expansion valve
- Energy balance

$$\dot{m}_r h_8 - \dot{m}_r h_9 = 0$$

- Exergy balance

$$\dot{m}_r ex_8 - \dot{m}_r ex_9 - \dot{Ex}_{d,EV} = 0$$

$$\dot{I}_{EV} = \dot{Ex}_{d,EV} = \dot{m}_r T_o(s_9 - s_8)$$

Pump
- Energy balance

$$\dot{m}_s(h_{14} - h_{15}) + \dot{W}_P = 0$$

$$\eta_{s,p} = \dot{m}_s \frac{h_{15s} - h_{14}}{\dot{W}_P}$$

Where h_{15s} is the enthalpy of the ideal state of point 15

Then

$$h_{15} = \frac{h_{15s} - h_{14}}{\eta_{s,p}} + h_{14}$$

- Exergy balance

$$\dot{m}_s(ex_{14} - ex_{15}) + \dot{W}_P - \dot{Ex}_{d,P} = 0$$

$$\dot{I}_{pump} = \dot{Ex}_{d,P} = \dot{m}_s T_o(s_{15} - s_{14})$$

Finally the internal exergy destruction of the global system is the sum of the individual exergy destruction at each component:

$$\sum_i \dot{Ex}_{d,i}$$

$$\sum_i \dot{Ex}_{d,i} = \dot{I}_{gen} + \dot{I}_{abs} + \dot{I}_{HE} + \dot{I}_{TV} + \dot{I}_{cond} + \dot{I}_{evap} + \dot{I}_{EV} + \dot{I}_{pump}$$

The exergy loss in the hot flow is between 2f and downstream 3f, that is $\dot{Ex}_{2f3f}$, and the available exergy in the flow is $\dot{Ex}_{a,f} = \dot{Ex}_{1f3f} + \dot{Ex}_{pump}$, thus we can calculate the global exergy efficiency of the system.

By proposing the set of temperatures of the evaporator, condenser, absorber, and the generator, also the solution heat exchanger effectiveness and the available heat input; according to the desired operating conditions, then we can calculate the different pressure levels and the different absorbent concentrations in the cycle according to the equations in the reference [10], then with the help of

REFPROP [9] and [11] we can get the different thermodynamic properties at the different points in the cycle.

Results of the exergy analysis of the absorption cycle

The table below shows the different properties of the absorption cycle at almost all the points that can reflect the behavior of the cycle according to some specific operating conditions using LiBr as an absorbent

Point	Temperature(°C)	X(KgLiBr/Kgsolution)	Pressure(Kpa)	h (KJ/Kg)	s(KJ/Kg K)
1f	165	NA	2525	490.78	6.391
2f	30	NA	2525	343.23	6.011
1	29	NA	101	121.55	0.4229
2	32	NA	101	134.09	0.4642
3	32	NA	101	134.09	0.4642
4L	12	NA	101	50.410	0.1806
5L	7.5	NA	101	31.520	0.1138
8	32	0	0.0469	134.09	0.4642
9	6.687	0	0.0098	134.09	0.4804
10	7	0	0.0098	2513.74	8.984
11	35.04	0.5717	0.0098	93.446	0.2022
12	35.04	0.5717	0.0469	93.446	0.2022
13	70	0.5717	0.0469	162.87	0.4156
14	32	0.5213	0.0098	69.781	0.21107
15	32	0.5213	0.0469	69.781	0.21107
16	70.506	0.5213	0.0469	151.987	0.4641
17	70	0	0.0469	2631.28	8.6407

*for T(generator)=70°C, T(evaporator)=7°C, T(condenser)=T(absorber)=32°C, effectiveness=0.92.global efficiency=35.66%
Table 7:State properties of the absorption cycle at some specific operating conditions

The pie chart below shows the contribution of the different components in the cycle in the overall irreversibility rate according to the specific conditions in the preceding table
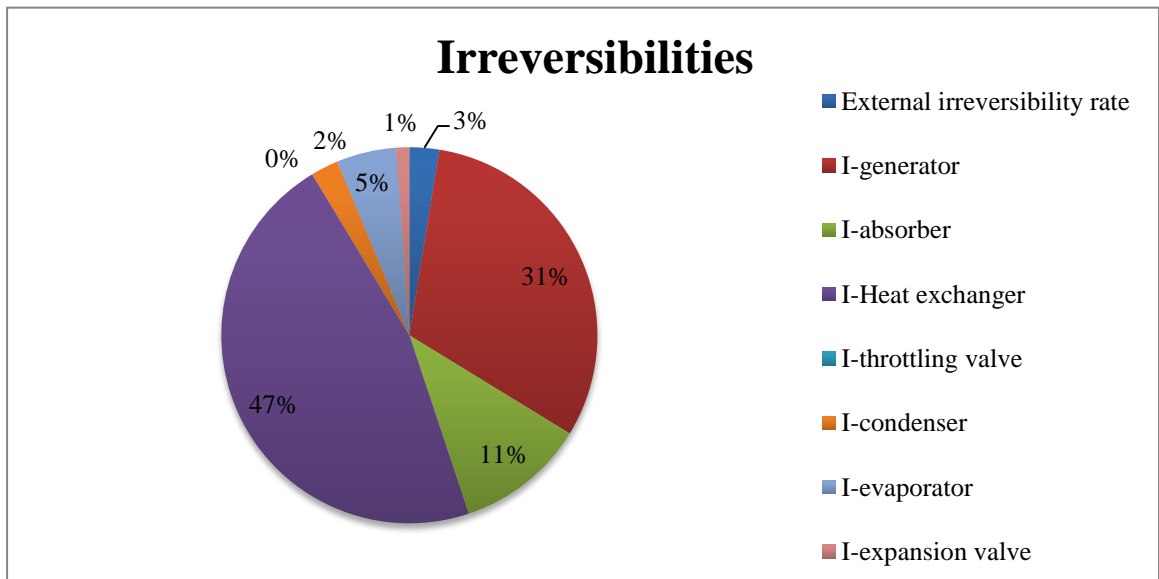

Figure 20: Contribution of each component in the overall irreversibility rate

Figure 21 shows the end of heat exchange temperature (T2F) on the global exergy efficiency, so that the global exergy efficiency decreases as the temperature T2F increases by the slope of 0.00337 %/$^{\circ}$C
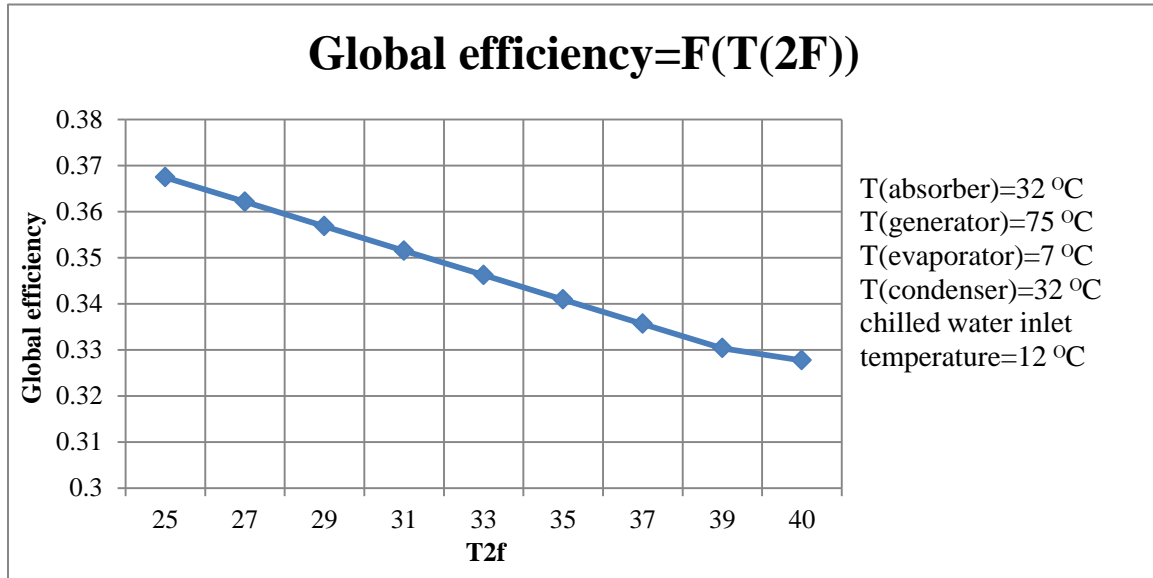
Figure 21: Influence of the hot flow temperature at the end of heat exchange on the global exergy efficiency

Figure 22 shows the influence the of the generator temperature on the global exergy efficiency, it shows clearly that the most convenient generator temperature range is between 65 and 75 $^{\circ}$C

Figure 22: Influence of the generator temperature on the global exergy efficiency

Figure 23 shows the influence of the evaporator temperature on the global exergy efficiency, it shows that the global exergy efficiency increases with the increase of evaporator temperature by a slope 0.002 %/ $^{\circ}$C

Figure 23: Influence of the evaporator temperature on the global exergy efficiency

Figure 24 shows the influence of the condenser temperature on the global exergy efficiency, this figure shows that the global exergy efficiency decreases with the increase of condenser temperature in a parabolic manner
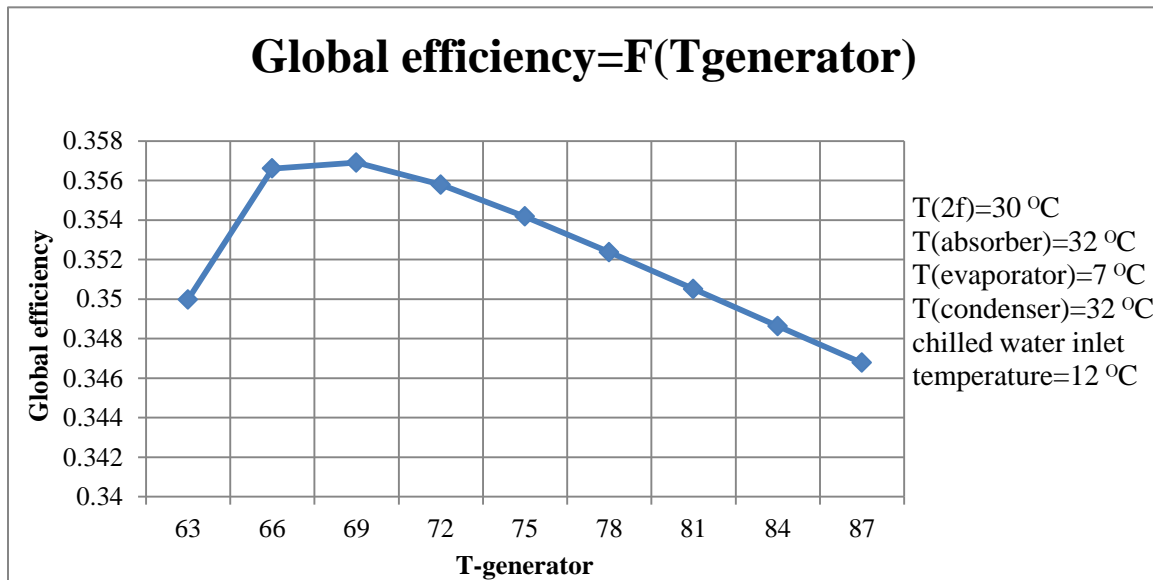

Figure 24: Influence of the condenser temperature on the global exergy efficiency

Figure 25 shows the influence of the absorber temperature on the global exergy efficiency, it shows that the global exergy efficiency decreases with the increase of absorber temperature

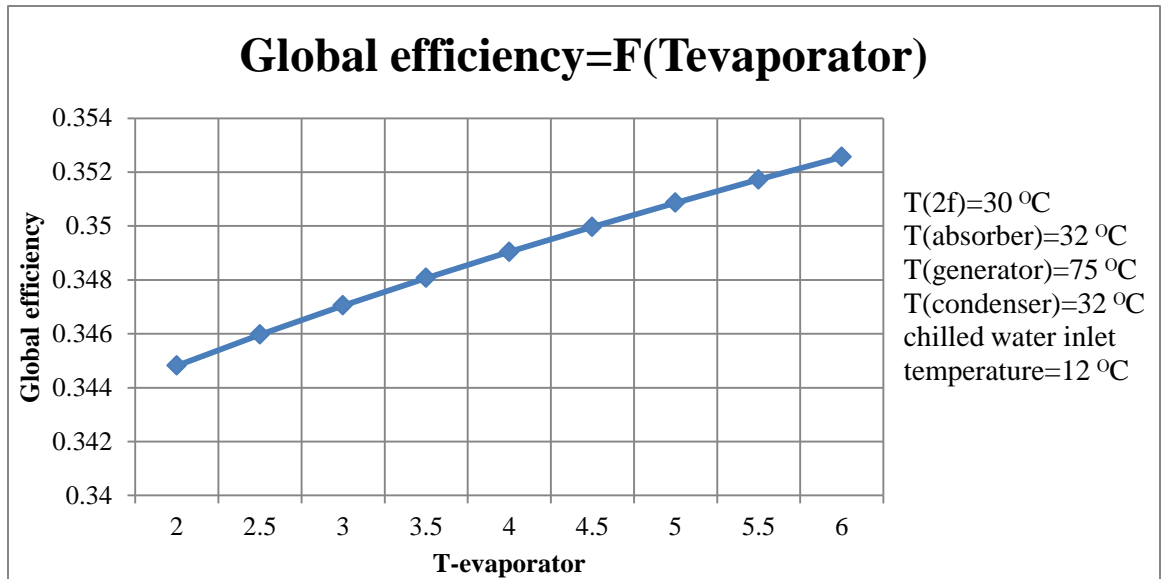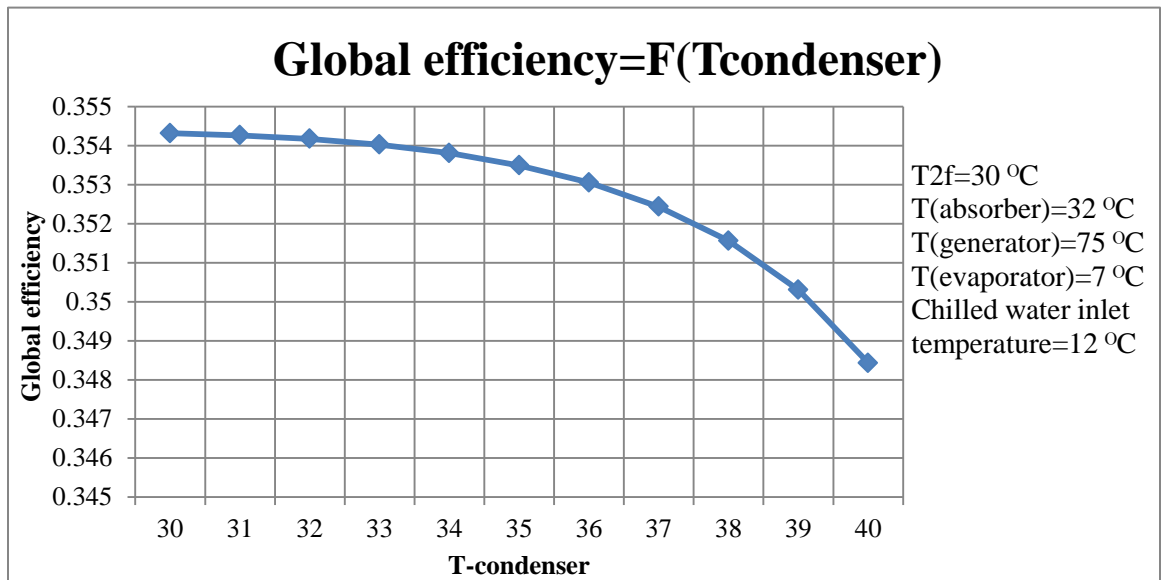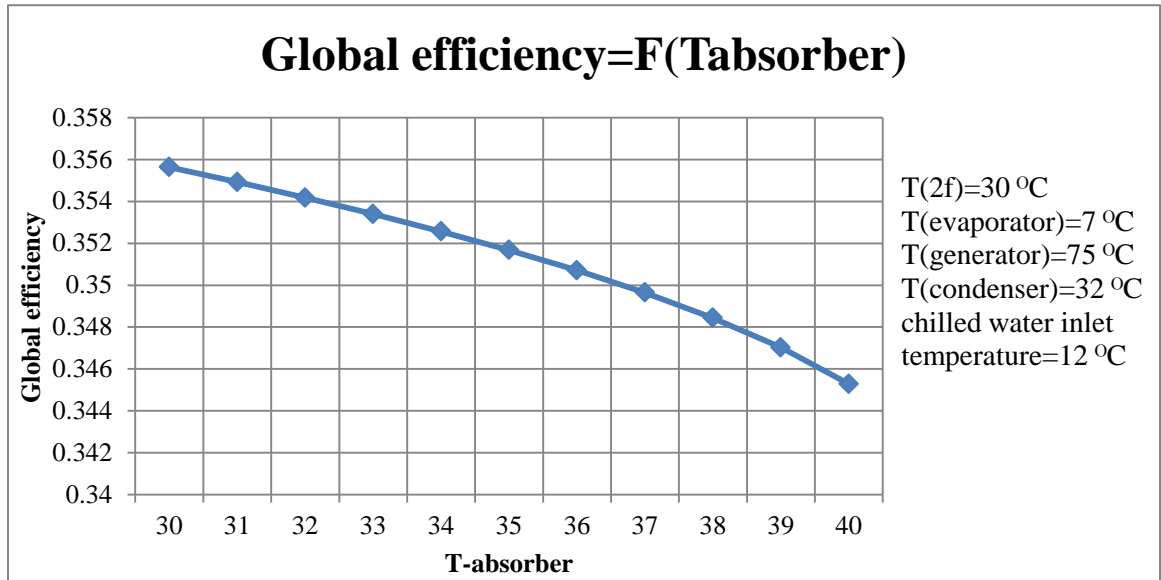Figure 25: Influence of the absorber temperature on the global exergy efficiency

Figure 26 shows the influence of the chilled water inlet temperature on the global exergy efficiency, this figure shows that the decreases with the increase of chilled water inlet temperature with a slope of 0.0073 %/ ^{O}C
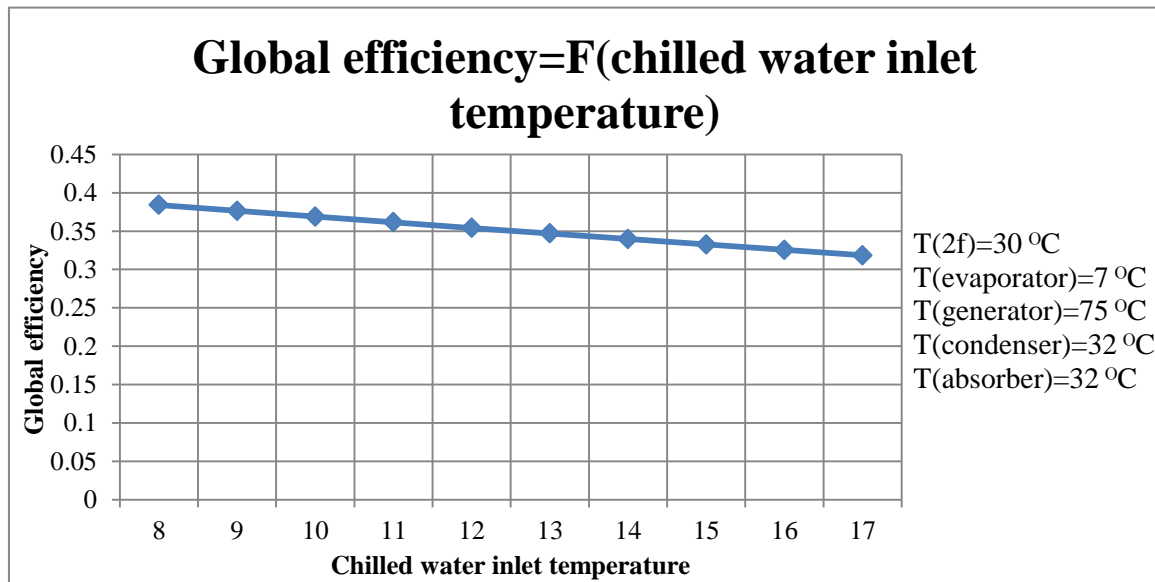

Figure 26: Influence of the chilled water inlet temperature on the global exergy efficiency

Chapter 5: Exergy analysis of producing 'cooling effect' from the predefined hot exhaust using the 'desiccant cooling cycle'

When we talk about desiccants then we instantly remind dehumidification, which minimizes the humidity quantity in the processed air, controlling humidity is an important thing to do in order to reach the comfort level.

Desiccant cooling

Desiccant cooling systems are those that use desiccant dehumidification and cooling (evaporative cooling, conventional vapor compression cooling or any other cooling system) in order to attain the desired processed air conditions. Desiccant cooling uses liquid or solid desiccant in order to dehumidify the processed air, uses also, low grade energy in the regeneration process (waste heat for example) in an open cycle manner, so desiccant cooling is environmental-friendly.

Desiccant cooling has long been adopted for both industrial and agricultural purposes, and is now taking a more and more prominent role in the air-conditioning field. Desiccant dehumidification is economical and has effective humidity control at low and moderate temperature really dwarfs the conventional method of humidity control, since it makes full use of surface vapor pressure difference to realize moisture transfer between the process air and the liquid desiccant.

Desiccant dehumidification can be done by either liquid or solid desiccant, Liquid desiccants have many advantages over solid ones, even though they give the same dehumidification effect, but liquid desiccants are better than the solid ones, when looking for humidity control, energy consumption, performance (we don't need to replace the liquid desiccant), indoor air quality (Natural disinfectant of airborne particulates & microorganisms), installation, weight, maintenance; Also, the regeneration temperature required for liquid desiccants is lower than that of the solid desiccants; the pressure drop through a liquid desiccant system is smaller than the pressure drop through a solid desiccant wheel; liquid desiccants can be used as a heat transfer medium in a heat exchanger, and the energy is stored as chemical energy rather than thermal energy. We must talk also about the energy storage capacity, in liquid desiccants such as lithium chloride or calcium chloride is up to 3.5 times higher compared to solid desiccants such as zeolites or silica-gel related to the same dehumidification process.

Using desiccant dehumidification with Vapor compression air conditioning might have more cooling production than the second one alone by 20 to 30% [12].

Description of Basic hybrid liquid desiccant cooling system

Firstly, let's talk about the dehumidification process:

Dehumidification unit is the most important part in these kinds of cooling, since this unit can handle the total latent load in the air conditioning process; the task of dehumidification unit is to remove the moisture of the inlet air by bringing into contact with sprinkled liquid desiccant.

The surface vapor pressure difference between liquid-desiccant film and air is acting as the driving force for mass transfer of water vapor in the air to liquid water in the liquid desiccant (condensation process). The liquid desiccant has a vapor pressure lower than that of water at the same temperature and the air passing over the solution approaches this reduced vapor pressure and is dehumidified. Vapor pressure of a liquid desiccant is directly proportional to its temperature and inversely proportional to its concentration .As the desiccant content in the mixture increases its vapor pressure decreases.

To regenerate the liquid desiccant from the diluted liquid we must heat the last one, heating this solution will increase the liquid surface water pressure, hence the water is then evaporated to the external air where the partial pressure of the water vapor is less. The heat source in our case is the predefined hot exhaust.

Among many liquid desiccants available aqueous solution of Lithium chloride, Calcium chloride, Ethylene glycol; Lithium chloride is found to be most effective since it has the lowest vapor pressure among these[13].

Packed bed towers are more effective than the other methods of heat and mass transfer in the dehumidifier, since they provide large rate of heat and mass transfer per unit volume and hence a compact design is possible [20].
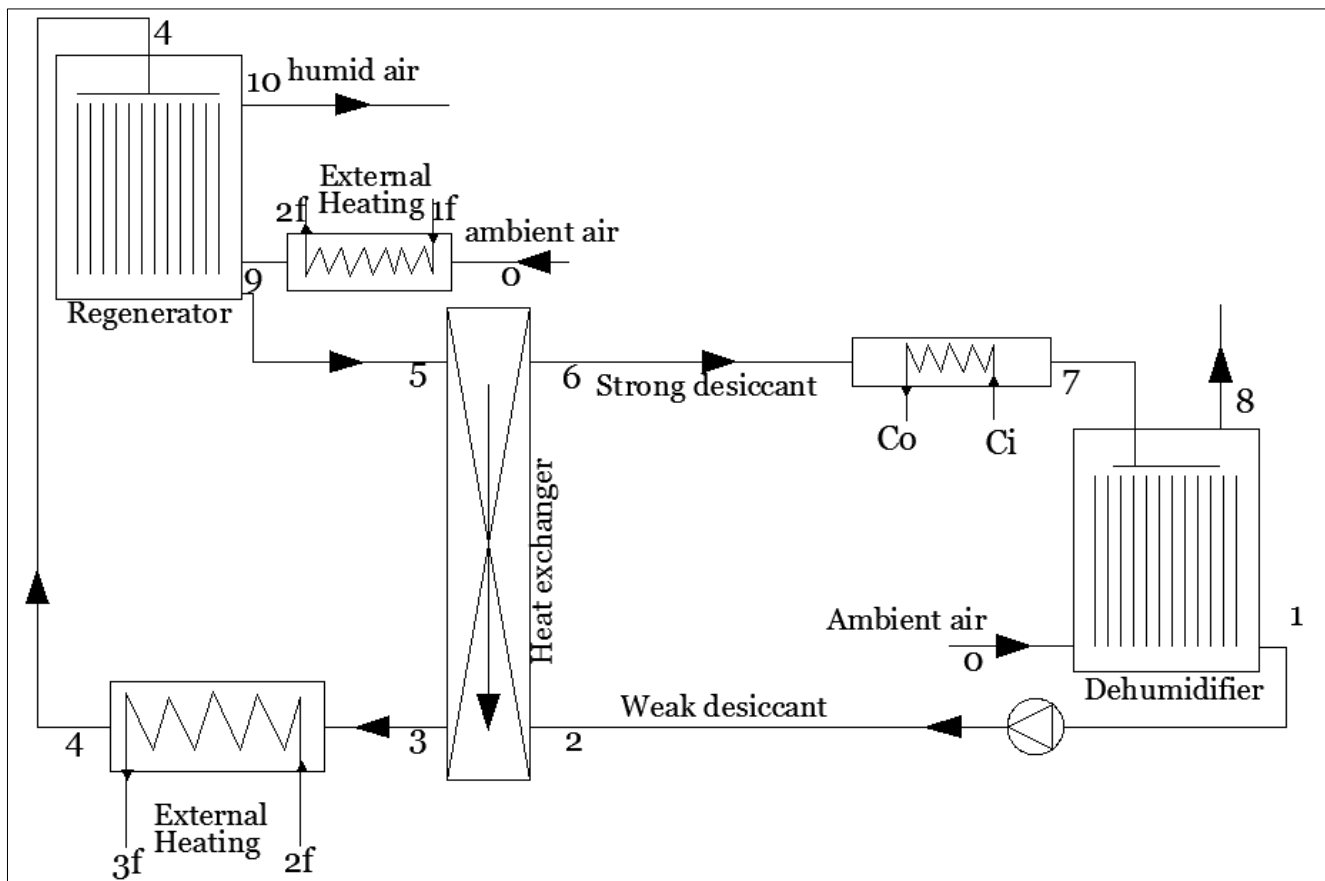

Figure 27: Schematic drawing of the liquid desiccant system

The cooling load is composed of sensible load and latent one, in the conventional vapor compression cycle these loads are handled by the evaporator, while by adding the dehumidifier to the cycle, we can use the evaporator for handling the sensible load only, hence we can reduce the size of the evaporator to the less than the half of that in the conventional vapor compression cycle, and hence the whole size of the conventional vapor compression system is reduced. Also, with desiccant systems, air humidity and temperature are controlled separately, enabling better control of humidity.

According to P. Gandhidasan [14], we can predict the water flow rate that is absorbed from the air flow in the dehumidifier by the desiccant, the outlet desiccant concentration, and the outlet desiccant temperature

$$\dot{m} = \frac{1}{\lambda} \left[\frac{G_{s,deh} c_{P,s,deh} \varepsilon_{HE}}{(1 - \varepsilon_{HE})} (t_7 - t_{c,i}) - G_{a,deh} c_{P,a} \beta (t_0 - t_7) \right]$$

$$X_{out} = \frac{G_{s,deh} \cdot S_{deh} \cdot X_{in}}{G_{s,deh} \cdot S_{deh} + m_{study}}$$

$$t_1 = \frac{t_7 - \varepsilon_{HE} t_{ci}}{1 - \varepsilon_{HE}}$$

Where λ, β, and ε_{HE} are the latent heat of water, temperature difference ratio of the dehumidifier, and the temperature effectiveness of the cooling water-desiccant heat-exchanger

$$\beta = \frac{t_0 - t_8}{t_0 - t_7} \qquad \varepsilon_{HE} = \frac{t_6 - t_7}{t_6 - t_{ci}}$$

Also, by doing an analogy between the dehumidifier and the regenerator we can obtain a simplified model to predict the mass flow rate of the evaporated water in the regenerator

$$\dot{m} = \frac{1}{\lambda S_{reg}} \left[G_{s,reg} S_{reg} c_{P,s,reg} (t_4 - t_5) - G_{a,reg} S_{reg} c_{P,a} \beta (t_9 - t_4) \right]$$

Where

$$\beta = \frac{t_9 - t_{10}}{t_9 - t_4}$$

The energy balance of the desiccant in the whole cycle yields

$$Q_{solution\ heating} + Q_{dehumidifier} = Q_{regenerator} + Q_{solution\ cooling}$$

Then

$$\dot{m}_f(h_{2f} - h_{3f}) + G_{s,deh}S_{deh}c_{p,s,deh}(t_1 - t_7) = G_{s,reg}S_{reg}c_{p,s,reg}(t_4 - t_5) + \dot{m}_{cw}(h_{co} - h_{ci})$$

Where we can get the desiccant output temperature (table 8)

$T_{5,exp}$	58.60	57.80	56.6	57.4	57.6	57.2	58.3	57.0	57.9	54.2	60.0	56.8	57.5
$T_{5,study}$	60.0	59.4	58.7	58.7	58.9	58.5	59.5	58.9	59.6	55.6	62.4	57.6	59.5
%error	2.4	2.8	3.7	1.9	2.2	2.2	2.1	3.3	2.9	2.6	4	1.4	3.4

Table 8: Comparison data of the proposed model outlet desiccant temperature results and the corresponding experimental data

By the comparison with the experimental data done by D. Y. GOSWAMI [15] then we get the following table (table 9)

Inlet						Outlet				Results		
$G_{a,reg}$	T_9	Y	$G_{s,reg}$	T_4	X(%)	T_{10}	Y	T_5	X(%)	m_{exp}	m_{study}	%error
0.833	30.4	0.0183	6.463	65.0	34.0	58.9	0.0579	58.6	34.5	1.55	1.443	6.9
1.098	30.1	0.018	6.206	65.1	34.1	59.3	0.0532	57.8	34.8	1.81	1.710	5.5
1.438	29.8	0.0177	6.479	65.1	34.5	57.5	0.0488	56.6	35.2	2.10	2.074	1.2
1.097	35.1	0.0180	6.349	65.1	33.4	58.5	0.0551	57.4	34.1	1.91	1.753	8.2
1.102	40.0	0.0178	6.354	65.0	33.6	58.9	0.0548	57.6	34.2	1.91	1.580	17.3
1.132	30.2	0.0143	6.370	65.2	34.0	57.6	0.0513	57.2	34.7	1.97	1.844	6.4
1.097	29.4	0.0210	6.440	65.5	33.6	58.5	0.0541	58.3	34.2	1.70	1.785	5.0
1.116	30.3	0.0182	5.185	65.4	34.4	57.6	0.0507	57.0	34.9	1.71	1.621	5.2
1.101	29.9	0.0180	7.541	65.2	34.3	59.0	0.0556	57.9	34.9	1.95	1.907	2.2
1.111	30.0	0.0187	6.245	60.3	34.4	55.8	0.0447	54.2	34.8	1.36	1.511	11.1
1.084	29.7	0.0184	6.315	70.0	34.5	62.6	0.0666	60.0	35.3	2.45	2.074	15.3
1.099	29.7	0.0177	6.400	64.8	32.8	57.6	0.0542	56.8	33.4	1.89	2.000	5.8
1.116	30.3	0.0182	6.428	65.0	34.9	57.9	0.0501	57.5	35.4	1.67	1.664	0.3

Table 9: Comparison data of the proposed model evaporated water results and the corresponding experimental data

So,

$$\dot{m} = \frac{1}{\lambda S_{reg}}\left[G_{s,deh}S_{deh}c_{P,s,deh}(t_1 - t_7) + \dot{m}_f(h_{2f} - h_{3f}) - \dot{m}_{cw}(h_{co} - h_{ci}) - G_{a,reg}S_{reg}c_{P,a}\beta(t_9 - t_4)\right]$$

Finally; the output desiccant concentration can be calculated from

$$X_{out} = \frac{G_{s,reg}.S_{reg}.X_{in}}{G_{s,reg}.S_{reg} - m_{study}}$$

Also by the comparison with the same experimental results in the preceding table we get (table 10)

$X_{out,exp}$	34.5	34.8	35.2	34.1	34.2	34.7	34.2	34.9	34.9	34.8	35.3	33.4	35.4
$X_{out,study}$	34.2	34.3	34.7	33.60	33.77	34.2	33.8	34.63	34.48	34.57	34.74	33.02	35.1
%error	0.87	1.4	1.3	1.5	1.2	1.4	1.2	0.8	1.2	0.6	1.6	1.1	0.9

Table 10: Comparison data of the outlet proposed model desiccant concentration results and the corresponding experimental data

The hybrid liquid desiccant system is composed mainly of: Dehumidifier, Regenerator, conventional vapor compression cooling system (compressor, evaporator, condenser, and expansion valve), and heat

exchangers. Dehumidifier is used to remove the moisture of the inlet air by bringing into contact with sprinkled liquid desiccant. It consists of: circulation pump, a strong desiccant tank, a weak desiccant tank, and a main blower. The strong desiccant is sprayed in a counter-flow direction and is brought in contact with blower air stream through packing material; strong desiccant absorbs humidity from air and converts to weak desiccant. Packing materials is the place where mass transfer occurs between falling film of the liquid desiccant and inlet air. Hence, the selection of packing materials will undoubtedly exert influence upon the performance of the dehumidification unit. Regenerator is used to regenerate the weak (diluted) solution flowing from dehumidification unit to an acceptable concentration. The regenerator is made up of a counter-flow packed bed regeneration tower similar to dehumidifier, a circulation pump, a strong desiccant tank, a weak desiccant tank, and a main blower. The weak desiccant is sprayed in a counter-flow direction and is brought in contact with hot air stream through packing material.

Conventional vapor compression cooling system is used to handle the sensible load in the processed air only, or with some latent load. Heat exchangers are used to make a heat transfer between the weak and strong desiccant and between the external heat source and the weak desiccant to be regenerated, also between strong desiccant and cooling water.

The study will be on the liquid desiccant dehumidification and regeneration processes, using lithium chloride (LiCl) as a liquid desiccant.

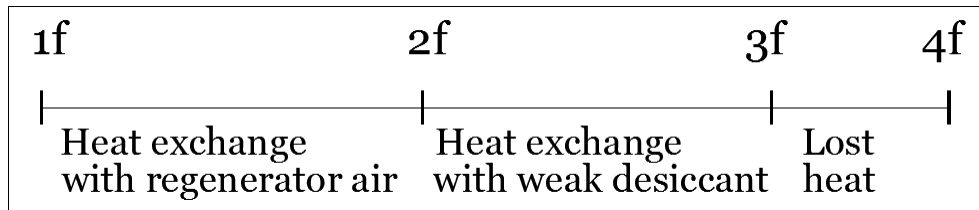## Exergy analysis of the hot exhaust and the above mentioned system

Figure 28: Heat exchange with the hot flow profile

Regenerator
Energy balance

$$\dot{m}_{ar}(h_9 - h_{10}) + \dot{m}_w h_4 - \dot{m}_s h_5 = 0$$

Exergy balance

$$\dot{m}_{ar}(ex_9 - ex_{10}) + \dot{m}_w ex_4 - \dot{m}_s ex_5 - \dot{Ex}_{d,reg} = 0$$

By substituting the energy balance equation in the exergy balance equation; then the irreversibility rate can be written as:

$$\dot{I}_{reg} = \dot{Ex}_{d,reg} = T_o(\dot{m}_{ar}(s_{10} - s_9) + \dot{m}_s s_5 - \dot{m}_w s_4)$$

Dehumidifier
Water vapor is transferred from air to liquid desiccant, accompanied by heat transfer
- Energy balance

$$\dot{m}_{ad}(h_0 - h_8) + \dot{m}_s h_7 - \dot{m}_w h_1 = 0$$

- Exergy balance

$$\dot{m}_{ad}(ex_0 - ex_8) + \dot{m}_s ex_7 - \dot{m}_w ex_1 - \dot{Ex}_{d,deh} = 0$$

By substituting the energy balance equation in the exergy balance equation; then the irreversibility rate can be written as:

$$\dot{I}_{deh} = \dot{Ex}_{d,deh} = T_o(\dot{m}_{ad}(s_8 - s_0) + \dot{m}_w s_1 - \dot{m}_s s_7)$$

Cooling water heat exchanger
- Energy balance

$$\dot{m}_{cw}(h_{ci} - h_{co}) + \dot{m}_s(h_6 - h_7) = 0$$

- Exergy balance

$$\dot{m}_{cw}(ex_{ci} - ex_{co}) + \dot{m}_s(ex_6 - ex_7) - \dot{Ex}_{d,CHWE} = 0$$

By substituting the energy balance equation in the exergy balance equation; then the irreversibility rate can be written as:

$$\dot{I}_{CWHE} = \dot{Ex}_{d,CWHE} = T_o\big(\dot{m}_{cw}(s_{co} - s_{ci}) + \dot{m}_s(s_7 - s_6)\big)$$

Solution heat exchanger
- Energy balance

$$\dot{m}_w(h_2 - h_3) + \dot{m}_s(h_5 - h_6) = 0$$

- Exergy balance

$$\dot{m}_w(ex_2 - ex_3) + \dot{m}_s(ex_5 - ex_6) - \dot{Ex}_{d,SHX} = 0$$

By substituting the energy balance equation in the exergy balance equation; then the irreversibility rate can be written as:

$$\dot{I}_{SHX} = \dot{Ex}_{d,SHX} = T_o\big(\dot{m}_w(s_3 - s_2) + \dot{m}_s(s_6 - s_5)\big)$$

External Heating of the regenerator inlet air
- Energy balance

$$\dot{m}_f(h_{1f} - h_{2f}) + \dot{m}_{ar}(h_0 - h_9) = 0$$

- Exergy balance

$$\dot{m}_f(ex_{1f} - ex_{2f}) + \dot{m}_{ar}(ex_0 - ex_9) - \dot{Ex}_{d,EHRA} = 0$$

By substituting the energy balance equation in the exergy balance equation; then the irreversibility rate can be written as:

$$\dot{I}_{EHRA} = \dot{Ex}_{d,EHRA} = T_o\left(\dot{m}_f(s_{2f} - s_{1f}) + \dot{m}_s(s_9 - s_0)\right)$$

External Heating of the weak desiccant
- Energy balance

$$\dot{m}_f(h_{2f} - h_{3f}) + \dot{m}_w(h_3 - h_4) = 0$$

- Exergy balance

$$\dot{m}_f(ex_{2f} - ex_{3f}) + \dot{m}_w(ex_3 - ex_4) - \dot{Ex}_{d,EHWD} = 0$$

By substituting the energy balance equation in the exergy balance equation; then the irreversibility rate can be written as:

$$\dot{I}_{EHWD} = \dot{Ex}_{d,EHWD} = T_o\left(\dot{m}_f(s_{3f} - s_{2f}) + \dot{m}_w(s_4 - s_3)\right)$$

Finally the internal exergy destruction of the global system is the sum of the individual exergy destruction at each component:

$$\sum_i \dot{E}x_{d,i}$$

$$\sum_i \dot{E}x_{d,i} = \dot{I}_{reg} + \dot{I}_{deh} + \dot{I}_{CHWE} + \dot{I}_{SHX} + \dot{I}_{EXRA} + \dot{I}_{EHWD}$$

The exergy loss in the hot flow is between 3f and downstream 4f, that is $\dot{E}x_{3f4f}$, and the available exergy in the flow is $\dot{E}x_{a,f}$ which is equal to $\dot{E}x_{1f4f}$, thus we can calculate the global exergy efficiency of the system.

By proposing the set of different air and desiccant flows, set of the different temperatures in the whole system, like the temperature at the points 4, 9, 7, 0,1f, 2f, 3f, 4f and the set of the different heat-exchangers effectiveness's, we can get the temperature and concentration at the remaining points of the whole system by the using of the above mentioned models. Then with the help of REFPROP [9] and [16] we can get the different thermodynamic properties of the different points in the cycle.

Results of the exergy analysis of the desiccant cycle

The table below shows the different properties of the desiccant cycle at almost all the points that can reflect the behavior of the cycle according to some specific operating conditions using LiCl as a desiccant

Point	Temperature (ºC)	X (KgLiCl/Kgsolution)	h (KJ/Kg)	s(KJ/Kg K)
1f	165.0	NA	490.78	6.391404
2f	94.80	NA	414.30	6.206538
3f	30.00	NA	343.23	6.011310
4f	26.00	NA	338.81	5.998051
ci	29.60	NA	124.15	0.431206
co	33.00	NA	138.36	0.477883
0	26	NA	349.48	7.045453
1	32.32	0.109655	101.85	0.477565
2	32.36	0.109655	102.00	0.478032
3	39.11	0.109655	126.46	0.557263
4	40.50	0.109655	131.48	0.573317
5	39.10	0.11	126.30	0.556956
6	32.32	0.11	101.77	0.477432
7	30.28	0.11	94.380	0.453161
8	31.78	NA	305.33	6.90080
9	68	NA	389.03	7.236833
10	86.45	NA	375.50	7.373087

*Ga,d=1.5, Gs,d=8.4, Ga,r=1, heat exchangers effectiveness=0.8 absolute humidity of ambient air=0.018, βd=6.2, βr=-0.671 global efficiency=26%

Table 11: State properties of the desiccant cycle at some specific operating conditions

The pie chart below shows the contribution of the different components in the cycle in the overall irreversibility rate according to the specific conditions in the preceding table
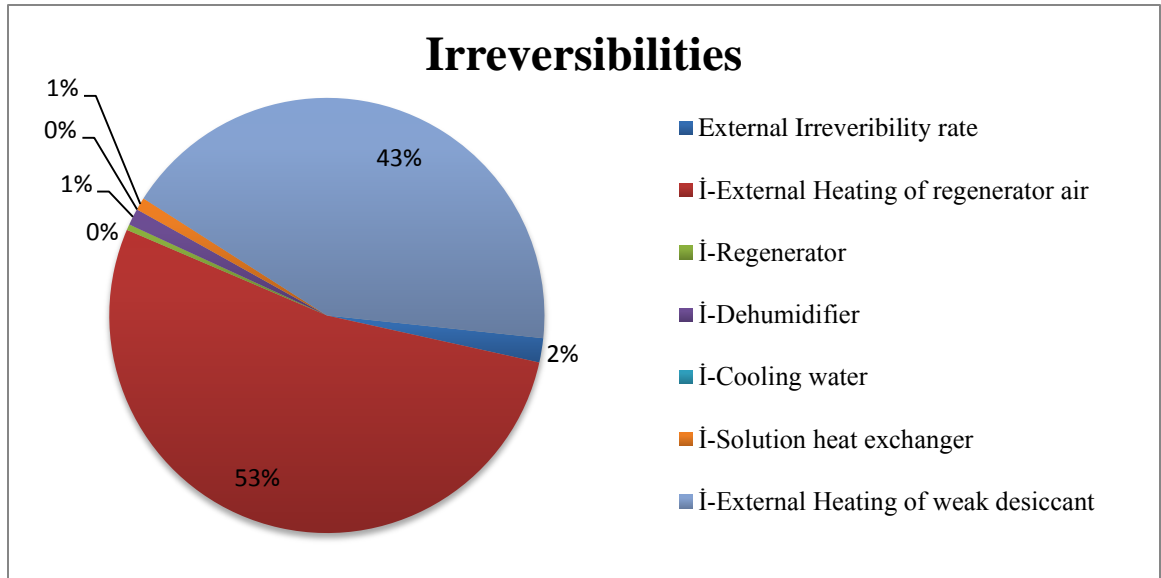
Figure 29: contribution of each component in the overall irreversibility rate

Figure 30 shows the effect of the temperature of the hot flow at the end of the heat exchange with the regenerator air on the global exergy efficiency, this figure shows clearly that the global exergy efficiency increases with the increase of T2f with a slope of 0.052 %/ ^{O}C

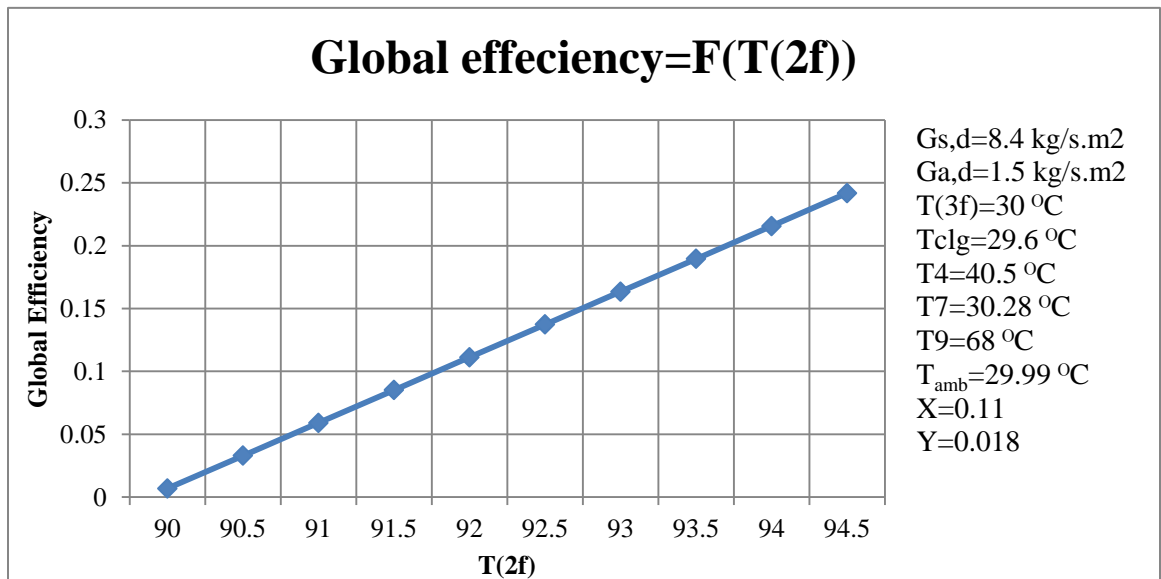Figure 30: Influence of the hot flow temperature at the end of heat exchange with the regenerator inlet air on the global exergy efficiency

Figure 31 shows the effect of the dehumidifier desiccant specific mass flow on the global exergy efficiency, it clearly shows that the global exergy efficiency decreases with the increase of the specific flow by a slope of 0.3509 %/kg.s^{-1}.m^2
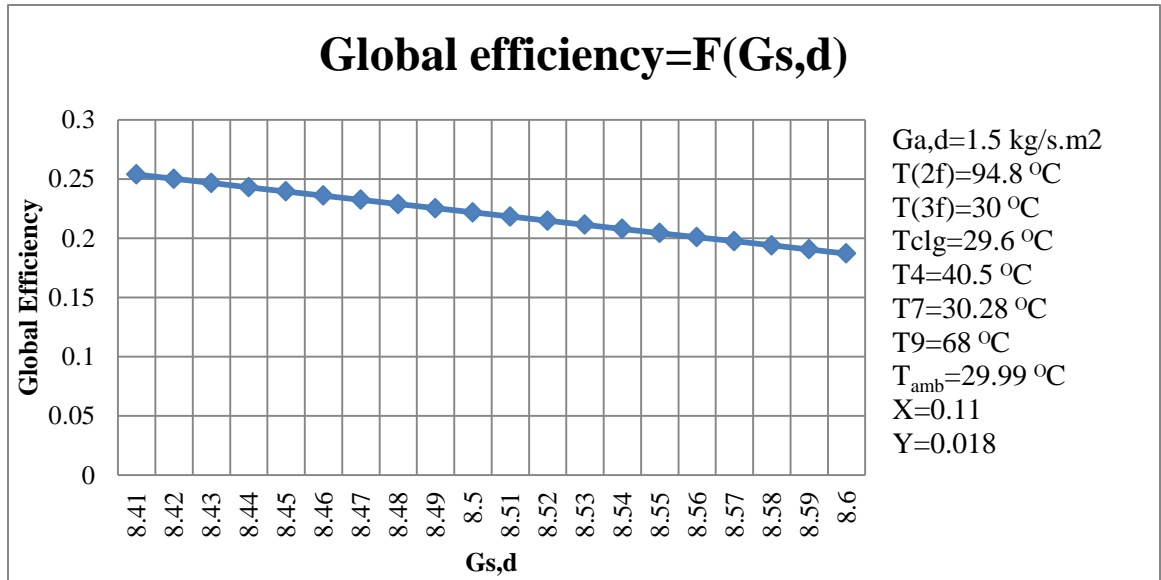
Figure 31: Influence of the dehumidifier desiccant specific flow on the global exergy efficiency

Figure 32 shows the effect of the dehumidifier air specific mass flow on the global exergy efficiency, it clearly shows that the global exergy efficiency increases with the increase of the specific flow by a slope of 0.0201 %/kg.s^{-1}.m^2
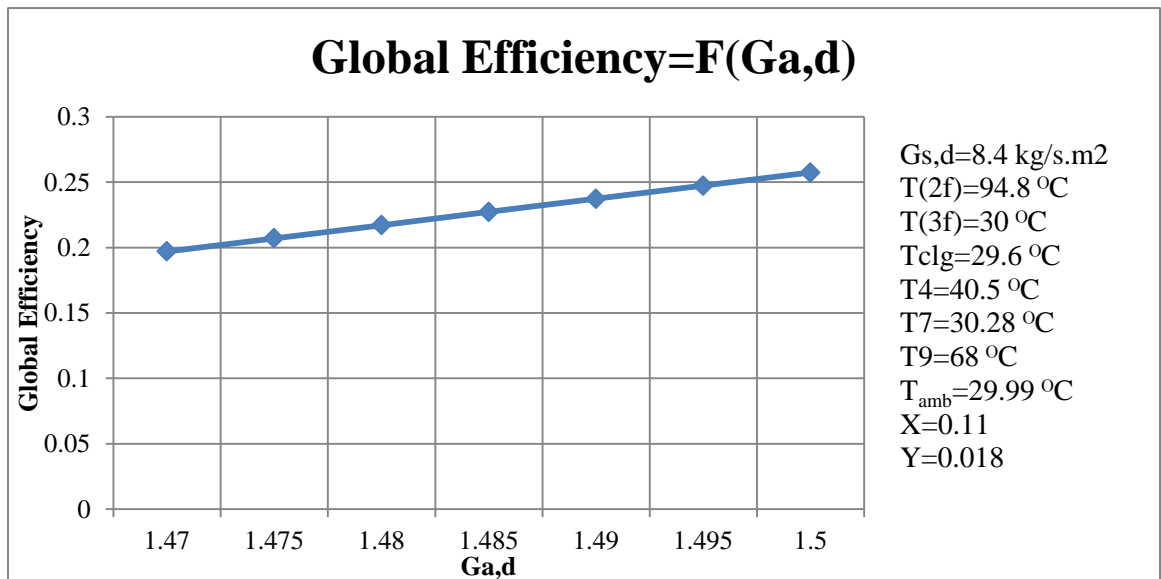

Figure 32: Influence of the dehumidifier air specific flow on the global exergy efficiency

Figure 33 shows the effect of the dehumidifier inlet desiccant concentration on the global exergy efficiency, it clearly shows that the global exergy efficiency increases with the increase of the concentration by a slope of 4.01 %/(kg/kg)
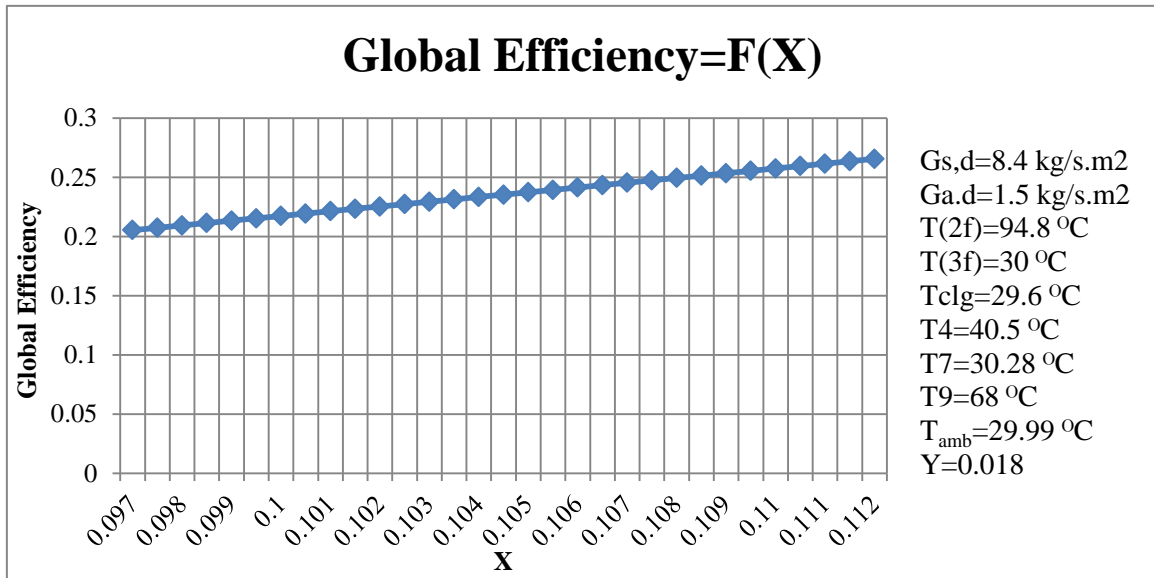
Figure 33: Influence of the dehumidifier inlet desiccant concentration on the global exergy efficiency

Figure 34 shows the effect of the ambient air absolute humidity on the global exergy efficiency, it clearly shows that the global exergy efficiency increases with the increase of the absolute humidity by a slope of 290 %/(kg/kg)
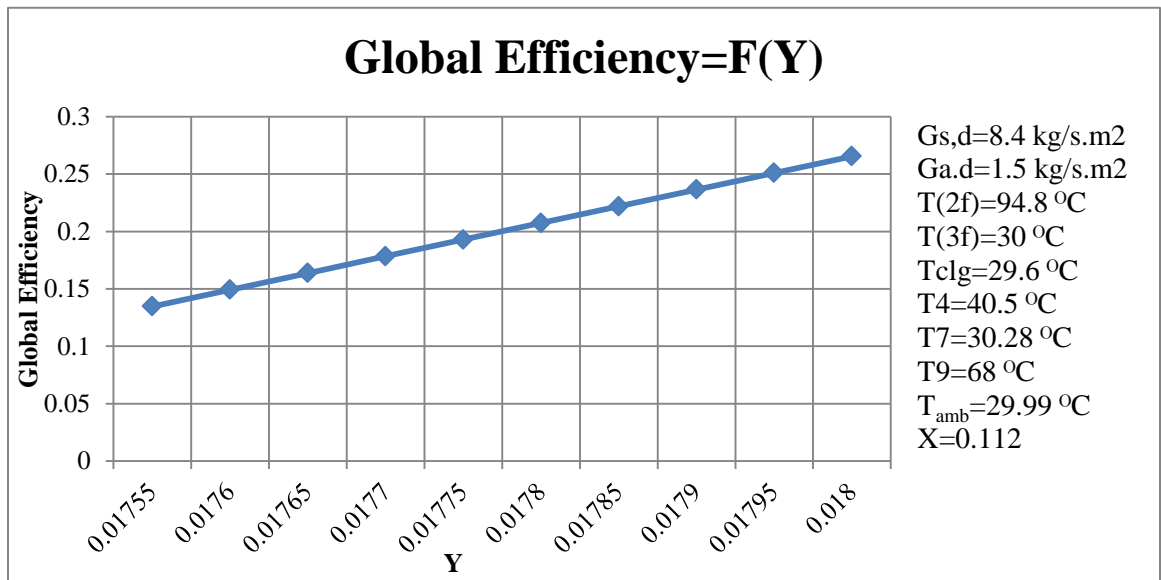

Figure 34: Influence of the ambient air absolute humidity on the global exergy efficiency

Figure 35 shows the effect of the hot flow temperature at the end of heat exchange with the weak desiccant on the global exergy efficiency, it clearly shows that the global exergy efficiency decreases with the increase of the temperature by a slope of 0.03 %/^{O}C
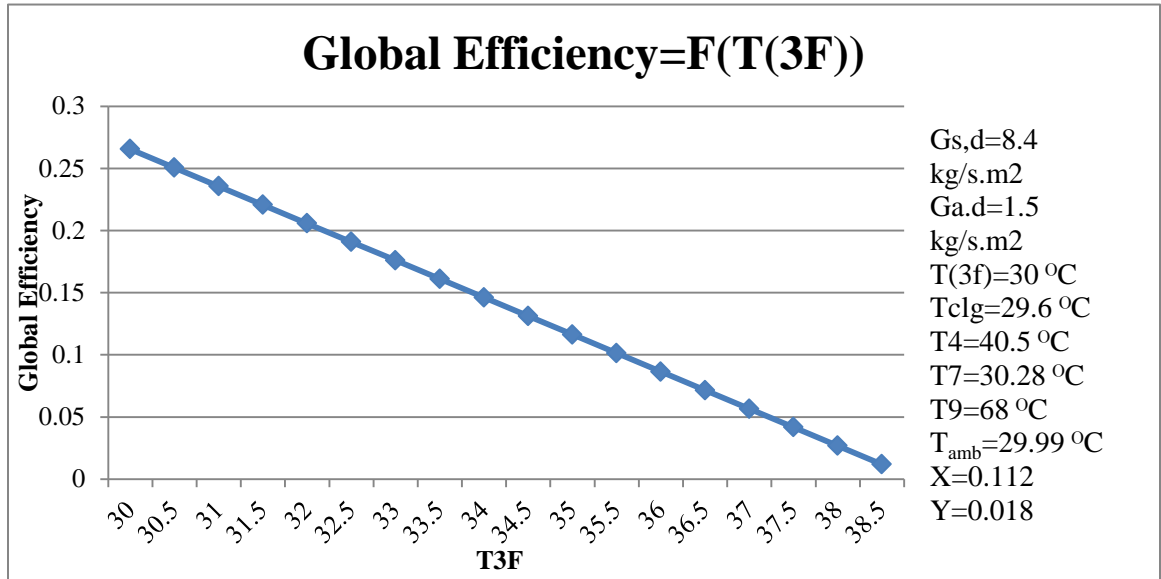
Figure 35: Influence of the hot flow temperature at the end of heat exchange with the weak desiccant on the global exergy efficiency

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