THERMODYNAMIC ANALYSIS OF TWO-STAGE VAPOUR COMPRESSION REFRIGERATION SYSTEM INTEGRATED WITH AN ABSORPTION SYSTEM (LiBr-H_2O)

A Thesis submitted towards the partial fulfilment of the requirement for the award of the degree of

Master of Technology
In
Thermal Engineering

Submitted by

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2K12/THR/08

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JULY 2014

CERTIFICATE

This is to certify that the thesis report entitled, "**THERMODYNAMIC ANALYSIS OF TWO-STAGE VAPOUR COMPRESSION REFRIGERATION SYSTEM INTEGRATED WITH AN ABSORPTION SYSTEM (LiBr-H_2O) "** being submitted by **Chandan Kumar** to the *Department of Mechanical Engineering, Delhi Technological University* in the partial fulfilment of the requirement for award of Master of Technology degree in **Thermal Engineering** is a record of bona fide work carried out by him under the supervision and guidance of Asst. Prof. Naushad Ahmad Ansari and Dr. Akhilesh Arora. The matter embodied in this report has not been submitted for the award of any other degree.

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DECLARATION

I hereby declare that all information in this document has been obtained and presented in accordance with academic rules and ethical conduct. This report is my own, unaided work. I have fully cited and referenced all material and result that are not original to this work. It is being submitted for the degree of Master of Technology in Engineering at the Delhi Technological University. It has not been submitted before for any degree or examination in any other university.

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ABSTRACT

This work comprises energetic and exergetic analysis of two thermodynamic cycles: First one is a conventional two stage cooling system by steam compression of ammonia and the other is named integrated refrigeration system. The conventional system that has been used as reference in this work is largely used for cooling purpose in industrial sector. The integrated refrigeration system is similar to the conventional system, but it uses the intermediate cooling, between the stages of high and low pressure, by cold water in closed circuit. The cold water is supplied by LiBr-H2O absorption system integrated to the conventional compression system. The heat input is obtained from waste material. Therefore, the energy supplied to the integrated refrigeration system is assumed of zero cost. Numeric simulation is used to compare the behaviour of both cycles. The results obtained in this comparison show that the integrated refrigeration system operates with a reduction of up to 9.70 % in COP. However, the integrated refrigeration system presented an increase of up to 11.89% in exergetic efficiency and 34.46% in refrigerating capacity in relation to the conventional system. These results, together with decrease in operational cost which will be valuable in further study, will make very attractive the use of the integrated refrigeration system.

Keywords: compression-absorption system; LiBr-H2O, exergy.

ACKNOWLEDGEMENTS

I would like to express my sincere gratitude to my project supervisor, Assistant professor **Mr. Naushad Ahmad Ansari**, for her supervision, invaluable guidance, motivation and support throughout during the project. I have benefitted immensely from his wealth of knowlwdge.

I would also like to thank **Dr. Akhilesh Arora,** (Head of The Department of Mechanical and Automation Engg. IGDTUW, Delhi), for his precious suggestion and support.

I am indebted to **Prof. Naveen Kumar,** Head of Department of Mechanical Engineering, Delhi Technological University for his support and encouragement in carrying out this project.

I am also grateful to **Prof. S. K. Garg** Pro Vice- Chancellor, Delhi Technological University for providing the research environment in the institute.

My gratitude is extended to my colleagues and friends who have not been mentioned here personally in making this project a success.

Last but not the least, I take this opportunity to express my deepest thanks to my parent, without their support , love and encouragement, it would not have been possible to pursue M.Tech degree studies. I sincerely thanks them.

Chandan Kumar

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NOMENCLATURE

ARS	Absorption refrigeration system
COP	Coefficient of performance
CRS	Compression refrigeration system
CRSmod	CRS modified system
e	Specific exergy ($kJ.kg^{-1}$)
E	Exergy ($kJ.s^{-1}$)
EV	Expansion valve
h	Specific enthalpy ($kJ.kg^{-1}$)
IRS	Integrated refrigeration system
$\dot{m}$	Mass flow rate($kg^{-1}s.$)
P	Pressure (kPa)
$\dot{Q}$	Rate of heat transfer ($kj^{-1}s.$)
S	Specific entropy ($kJ.kg^{-1}.K^{-1}$)
T	Temperature (K or ºC)
v	Specific volume (m^3kg^{-1})
∀	Volume flow rate(m^3s^{-1})
$\dot{W}$	Power input($kj\text{-}s^{-1}$)
W_e	Electric power(kW)
x	Mass fraction LiBr-water

Subscript

o	Ambient
C	Condenser
E	Evaporator
F	Fraction solution
G	Generator
I	Inter-stage
L	Liquid phase
S	Strong solution
V	Vapour phase
CV	Control volumes
W	Weak solution

Superscripts

0	Standard state value
n	Polytropic coefficient
CH	Chemical component
PH	Physical component

Greek symbols

Δ	Difference
ε	Exergetic efficiency
η_c	Compressor efficiency
η_b	Pump efficiency

(1). INTRODUCTION

Energetic and exergetic thermodynamic analysis have been widely applied to analyze the performance of cooling plants. The measure taken in general in industrial cooling is the compression by stages, when the difference is very high between evaporator and condenser. This allows, through an intermediate cooling, reducing the effort of compressor so resulting an increase in the frigorific performance of the plant. This happens, in general, by injection of a liquid between the stages, where part of the refrigerant fluid, which could be used for increasing the frigorific capacity, is applied to cool the discharge by the compressor in the first stage.

The target of the study is to do an energetic and exergetic analysis to offer a change in this usual method, by proposing a system of in absorption system integrated to the compression refrigeration system (CRS), for intermediate cooling, and increasing the frigorific capacity of refrigeration

For the compression refrigeration system (CRS) and for the integrated refrigeration system (IRS) we study the effects of evaporation temperature (T_E), condensation temperature (T_C),intermediate pressure(P_i), degree of overheating in the temperature of suction ($T_{SUCTION}$), in the power of the compressors ($\dot{W}$), in the heat transfer rates, ($\dot{Q}$), in the coefficient of performance (COP) and in the exergetic efficiency (ε).

Before going deep into the matter we discuss basic elements in this project in brief.

(1.1). **Vapour-compression refrigeration**

Vapour-compression refrigeration is one of the most important refrigeration cycles and is the most widely used method for air-conditioning of buildings and industrial cooling system. It is also used in domestic refrigerators and automobile, large-scale storage of foods and fish, refrigerated vehicles and rail cars, and for other commercial and industrial services. Oil refineries, chemical processing plants, and natural gas processing plants are among the many kinds of industrial plants that often employ large vapour-compression refrigeration systems. Refrigeration can be defined as lowering the temperature of an enclosed region by abstracting heat from that region and carrying it elsewhere.

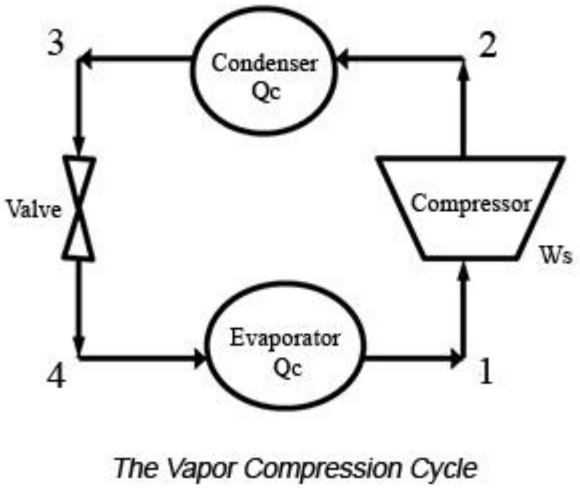

Fig. (1) Simple VC system

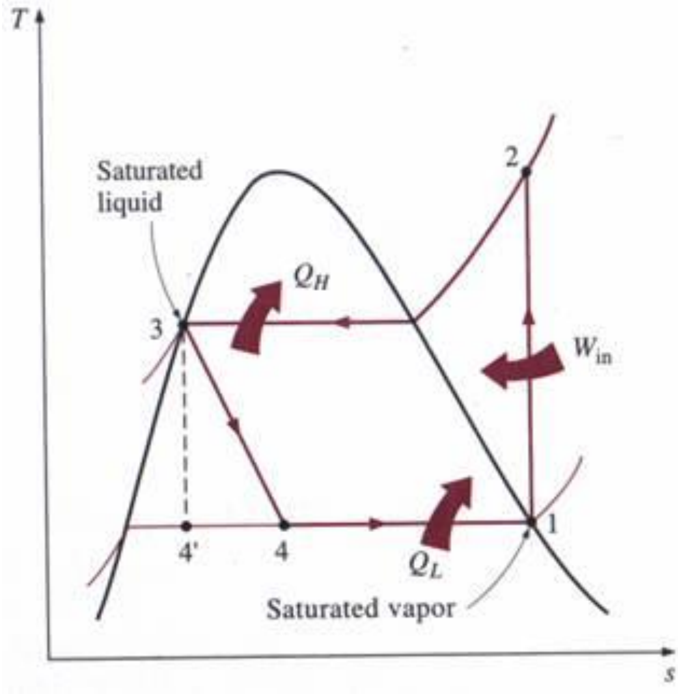

Fig. (2). T-S diagram of simple VC cycle

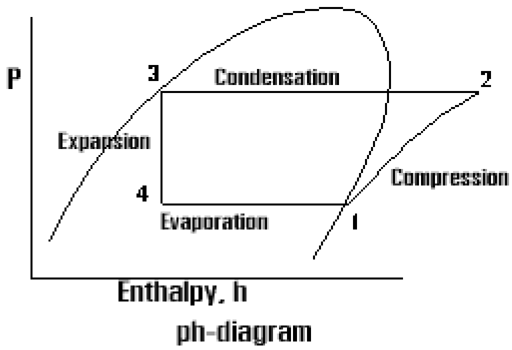

Fig.(3). P-h diagram of simple VC cycle

The vapour-compression uses a circulating liquid refrigerant as the medium which absorbs heat from the space to be cooled and subsequently rejects that heat to the outside space. Figure 1 shows a typical, single-stage vapour-compression system. Such systems have four basic components: a compressor, a condenser, a thermal expansion valve (also called a throttling devise or metering device), and an evaporator. Circulating refrigerant enters the compressor in the thermodynamic state known as a saturated vapour and is compressed to a higher pressure; as a result, it has higher temperature as well. The hot and highly compressed vapour is then in the thermodynamic state known as a superheated vapour and it is at a temperature and pressure at which it can be condensed with either cooling water or cooling air. That hot vapour then flows through a condenser where it is cooled and condensed into a liquid by flowing through a coil or tubes with cool water or cool air flowing across the coil or tubes. This is where the circulating refrigerant releases heat from the system and the released heat is carried away by either the water or the air (whichever may be the case).

The condensed liquid refrigerant, in the thermodynamic state known as a saturated liquid, next flows through an expansion devise where it undergoes a sudden reduction in pressure. That pressure reduction results in the adiabatic flash evaporation of a part of the liquid refrigerant. The auto-refrigeration effect of the adiabatic flash evaporation lowers the temperature of the liquid and vapour refrigerant mixture to where it is colder than the temperature of the enclosed space, which is to be refrigerated.

The cold mixture then flows through the coil or tubes in the evaporator. A fan circulates the warm air in the enclosed space across the coil or tubes carrying the cold refrigerant liquid and vapour mixture. This warm air evaporates the liquid part of the cold refrigerant mixture. At the same time, the circulating air is cooled and thus lowers the temperature of the enclosed space to the desired temperature. The evaporator is where the circulating refrigerant absorbs and removes heat which is subsequently rejected in the condenser and transferred elsewhere by the water or air used in the condenser.

To complete the refrigeration cycle, the refrigerant vapour from the evaporator is again a saturated vapour and is routed back into the compress

(A). Components of vapour compression system

A simple VC cycle consists of four basic components:-

(a). Compressor.

(b). Condenser.

(c). Throttling device.

(d). Evaporator.

(a). Compressor

The purpose of compressor is to raise the pressure of low pressure suction gases and deliver it at the high pressure. Most common forms of compressors are discussed below briefly.

Reciprocating compressor

A reciprocating compressor or piston compressor is called positive-displacement compressor that uses pistons driven by a crankshaft to raise the pressure of inlet gases. The intake gas enters the suction manifold, then flows into the cylinder where it gets compressed by the piston driven in a reciprocating motion through a crankshaft, and is then discharged. It is widely used in oil refineries, gas pipelines, chemical plants, natural gas processing plants and refrigeration plants. One special application is the blowing of plastic bottles made of polyethylene terephthalate (PET).

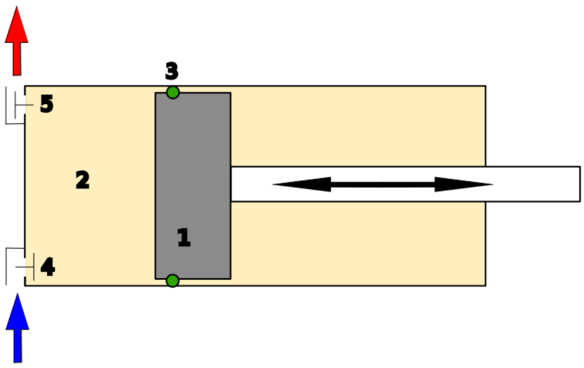

Fig. (4). Reciprocating Compressor

Centrifugal compressor

Centrifugal compressors, sometimes called radial compressors, are a sub-type of dynamic axis-symmetric work-absorbing turbo machinery. The idealized compressive dynamic turbo-machine obtains a pressure rise by adding kinetic energy/velocity to a continuous flow of fluid through the rotor or impeller. This kinetic energy is then converted to an increase in potential energy/static pressure by slowing the flow through a diffuser. The pressure rise in impeller is in most cases almost equal to the rise in the diffuser section. A simple centrifugal compressor has four components: inlet, impeller/rotor, diffuser, and collector.

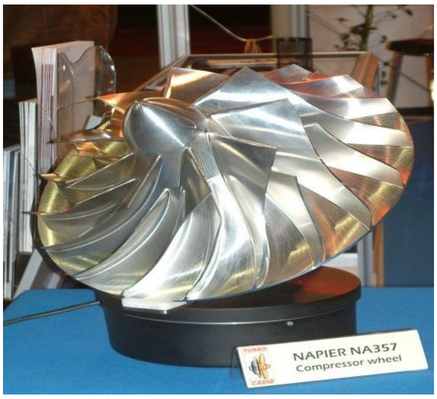

Fig. (5). Centrifugal Compressor

Axial compressor

An axial compressor is a machine that can continuously pressurise gases. It is a rotating, airfoil-based compressor in which the gas or working fluid principally flows parallel to the axis of rotation. This differs from other rotating compressors such as centrifugal compressors, axi-centrifugal compressors and mixed-flow compressors where the fluid flow will include a "radial component" through the compressor. The energy level of the fluid increases as it flows through the compressor due to the action of the rotor blades which exert a torque on the fluid. The stationary blades slow the fluid, converting the circumferential component of flow into pressure

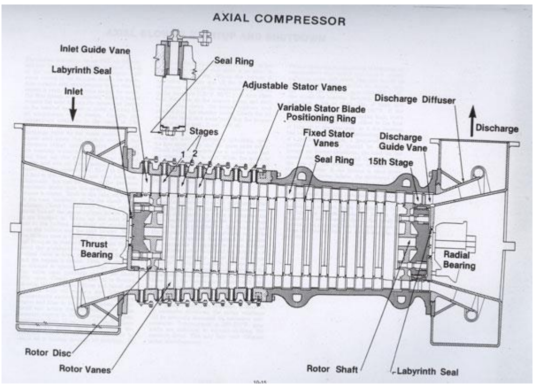

Fig. (6). Axial Compressor

(b). Condenser

In systems that involve heat transfer, a condenser is a device or unit which is used to condense a substance from its gaseous state to its liquid state, typically by cooling it at constant pressure. In doing so, the latent heat is released by the substance, and transferred to the condenser coolant. Condensers are necessarily heat exchangers which have various designs and made in many sizes ranging from little small (hand-held) to very large industrial scale units used in huge plants. For example, a refrigerator uses a condenser to release heat extracted from the evaporator unit to the outside air. Condensers are used in air conditioning, industrial chemical processes such as distillation, thermal power plants and other heat-exchange systems. Use of cooling water or atmospheric air as the coolant is common in many condensers. Most common forms of condensers used are discussed below briefly.

Surface condenser

A surface condenser is a very commonly used term for a water-cooled shell and tube heat exchanger installed on the exhaust steam from a steam turbine in thermal power stations. These condensers are basically heat exchangers which convert steam from its gaseous state to its liquid state at a pressure lower than atmospheric pressure. Where there is a shortage of cooling water supply, an air-cooled condenser is often used. An air-cooled condenser is significantly more costly and cannot achieve as low a steam turbine exhaust pressure as in the case of water-cooled surface condenser. Surface condensers can also be used in applications and industries other than the condensing of steam turbine exhaust in power plants.

Fig. (7). Surface Condenser

Shell and tube condenser

A shell and tube heat exchanger is an important class of heat exchanger designs. It is the most common kind of heat exchanger in petroleum industries and other large chemical processes, and is very suited for higher-pressure applications. As its name implies, this type of heat exchanger consists of a shell (a large pressure vessel) with a bundle of tubes inside it. One fluid flows through the tubes, and another fluid runs over the tubes (through the shell) to exchange heat between the two fluids. The set of tubes is called a tube bundle, and may be composed of several types of tubes: plain, longitudinally finned, etc.

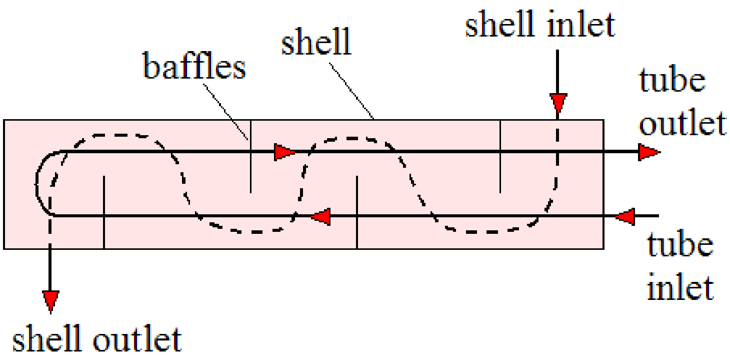

Fig. (8). Shell and Tube heat exchanger

(c). Throttling Devices

Throttling Devices or the Expansion valves used in the Refrigeration and Air Conditioning Systems Throttling devise is another important component of all the refrigeration systems and air conditioning systems apart from the compressor, condenser and the evaporator. The refrigerant leaves the compressor at very high pressure and temperature and enters into the condenser. After leaving the condenser the refrigerant is at medium temperature and high pressure and then it enters the throttling devise. In the throttling valve the pressure and the temperature of the refrigerant is reduced drastically and quickly. Thus it is the throttling valve where the temperature of the refrigerant is decreased and it is then able to produce the refrigeration effect in the evaporator of the refrigerator or the cooling coil of the air conditioner. The throttling valve also controls the amount of the refrigerant that should enter into the evaporator depending on the load on refrigeration.

Working of Throttling Devices Work-

The throttling devise can be the valve or the copper tubing that permits the flow of the refrigerant through very small opening also called as orifice. The throttling devices permits restricted flow of the refrigerants. The throttling devices are also called as the expansion valves because when the refrigerant flows through them the pressure of the refrigerant decreases or it expands. When refrigerant flows through the small opening, its pressure decreases due to friction. The amount of the refrigerant passing through the throttling valve depends on the extent of opening of the orifice. It also depends on the difference of pressure across two sides of the throttling valve, which are condenser and the evaporator.

Evaporator

When the liquid refrigerant reaches the evaporator its pressure has been reduced, releasing its heat content and making it much cooler than the fan air flowing around it. This results in the refrigerant to absorb heat from the warm air and reach its low boiling point quickly. The refrigerant then vaporizes, absorbing the maximum amount of heat. This heat is then carried by the refrigerant from the evaporator as a low-pressure gas through a hose or line to the low side of the compressor, where the whole refrigeration cycle is repeated.

The evaporator absorbs heat from the area that has to be cooled. The required temperature of cooling of the area will determine if refrigeration or air conditioning is desired. For example, food preservation generally requires low refrigeration temperatures, ranging from 40°F (4°C) to below 0°F (-18°C).

A higher temperature is required for human comfort. A larger area is cooled, which requires that large volumes of air be passed through the evaporator coil for heat exchange. A blower becomes a necessary part of the evaporator in the air conditioning system. The blower fans must not only draw heat-laden air into the evaporator, but must also force this air over the evaporator fins and coils where it surrenders its heat to the refrigerant and then forces the cooled air out of the evaporator into the space being cooled.

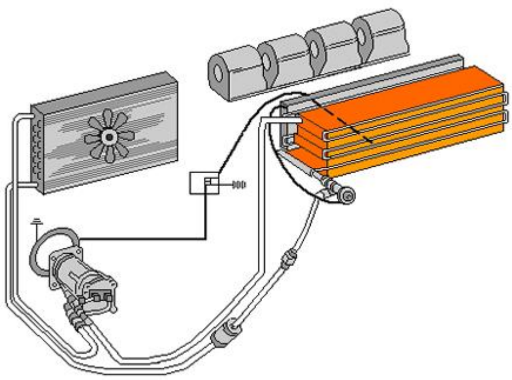

Fig. (9). Evaporator

(1.2). MULTISTAGE VAPOUR COMPRESSION SYSTEM

Multistage vapour compression refrigeration system can be studied under the following heads:

(a). Multistage Compression System

Whenever a gas is compressed from a given pressure to a higher pressure in one stage, it takes more work for compressing the same mass and needs a larger swept volume of the compressor than when more than one stage is used. If the compression ratio is very large, then the compressor may stop delivery of the fluid. Single stage compressors are therefore designed with the compression ratio around 7 to 9. However, for high capacity refrigeration systems, even at this pressure ratio, multistage compression is used.

Multistage compression results in a higher volumetric efficiency. This means that the compressor capacity has increased as compared to a single stage compressor having the same stroke and clearance volume.

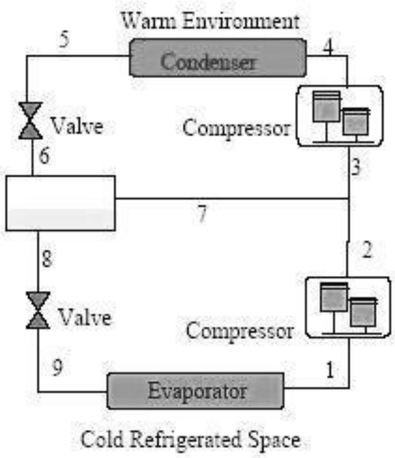

Figure (10) Schematic Representation of Multistage Compression System

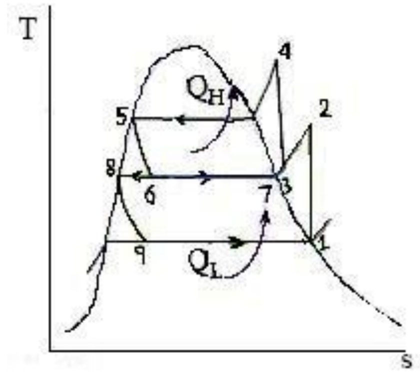

Figure (11) T-*s* Plot of Multistage Compression System

The working of a multistage compression system is discussed with the help of the following T-*s* diagram (fig 11). The refrigerant from the evaporator at state 1 is compressed by the low pressure (LP) compressor to the flash intercooler. The vapour is de-superheated from state 2 to state 3. It combines with the vapour separated by throttling from state 5-6 and this increased mass of vapour is now delivered to the high pressure (HP) compressor. The compressed vapour from the HP compressor is delivered at state 4. This vapour is now condensed in the condenser before being throttled. The liquid at the intermediate pressure pi is again throttled to the evaporator pressure. The heat transfer to the evaporator causes the vaporization of the refrigerant, thus completing the cycle.

(b). Multistage Evaporator System

In multistage evaporator system, the refrigerant is used to provide cooling effect in more than one refrigeration space. This is used when we have to maintain two or more refrigerated spaces at different temperatures and the size of the refrigerated space is such that the setting up of individual refrigeration systems for each one of them is not justified. The schematic representation of this type of system is given in Figure (12) along with the corresponding T – S diagram in Figure (13).

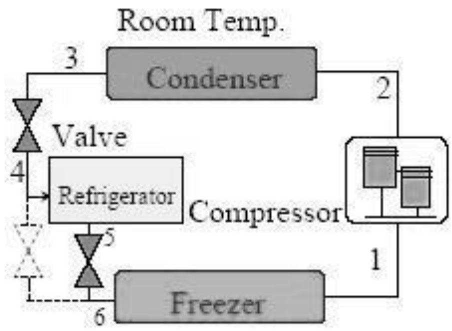

Figure (12) Schematic Representation of Multistage Evaporator System

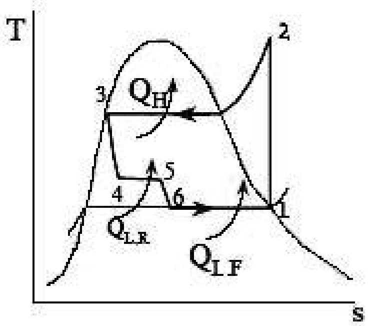

Figure (13) T-*s* plot of Multistage Evaporator System

The refrigerant is first throttled from state 3-4 and then passes into the first evaporator space. The refrigerant coming out the evaporator is again throttled to a lower pressure and then enters the next evaporator (freezer in figure). The refrigerant coming out of the freezer is then compressed in the compressor from state 1 to state 2. The rest of the cycle is similar to a simple vapour compression refrigeration cycle.

(c). Cascade Refrigeration System

Figure (14) is a schematic representation of a cascade refrigeration system. A cascade refrigeration system can be considered to be equivalent to two independent vapour-compression systems linked together in such a way that the evaporator of the high-temperature system becomes the condenser of the low-temperature system. However, the working media of the

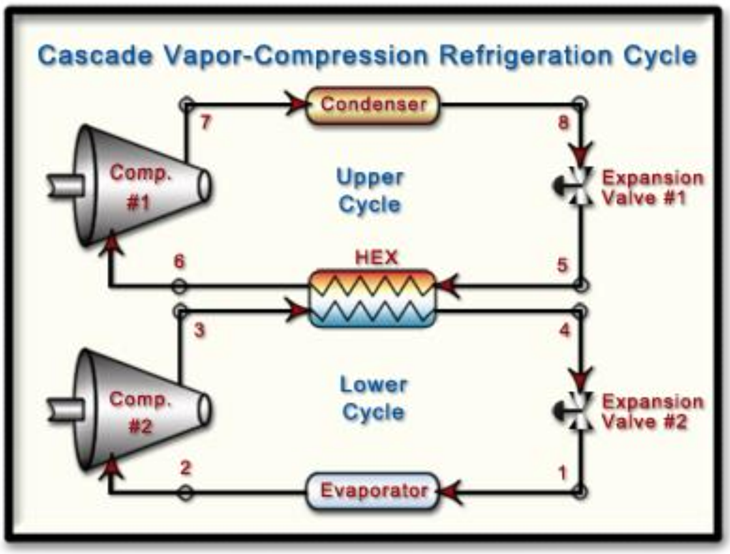

Figure (14) Cascade refrigeration system

two systems are separated from each other. This therefore, allows the use of different refrigerants working at different temperature ranges to achieve the desired effect, which would otherwise, need to be achieved by a single refrigerant working at a bigger operating pressure range. Thus, we can make use of suitable refrigerants at the higher and lower pressure ranges to derive maximum benefit.

(1.3). Simple Vapour Absorption Refrigeration System

In view of shortage of energy production and fast increasing energy consumption, there is a need to minimize the use of energy and conserve it in all possible ways. Energy conservation (i.e., energy saved is more desirable than energy produced) is becoming a slogan of the present decade and new methods to save energy, otherwise being wasted, are being explored. Recovering energy from waste heat and/or utilizing it for system efficiency improvement is fast becoming a common scientific temper and industrial practice now days. The present energy crisis has forced the scientists and engineers all over the world to adopt energy conservation measures in various industries. Reduction of the electric power and thermal energy consumption are not only desirable but unavoidable in view of fast and competitive industrial growth throughout the world. Refrigeration systems form a vital component for the industrial growth and affect the energy problems of the country at large. Therefore, it is desirable to provide a base for energy conservation and energy recovery from Vapour Absorption System. Although, the investigations undertaken in this work are of applied research nature but certainly can create a base for further R & D activities in the direction of energy conservation and heat recovery options for refrigeration systems and the analysis can be extended further to other Refrigeration and Air Conditioning Systems. In recent years, research has been devoted to improvement of Absorption Refrigeration Systems (ARSs).

What is Absorption Refrigeration System?

The vapour absorption refrigeration system comprises of all the processes in the vapour compression refrigeration system like compression, condensation, expansion and evaporation. In the vapour absorption system the refrigerant used is ammonia, water or lithium bromide. The refrigerant gets condensed in the condenser and it gets evaporated in the evaporator. The refrigerant produces cooling effect in the evaporator and releases the heat to the atmosphere via the condenser. .

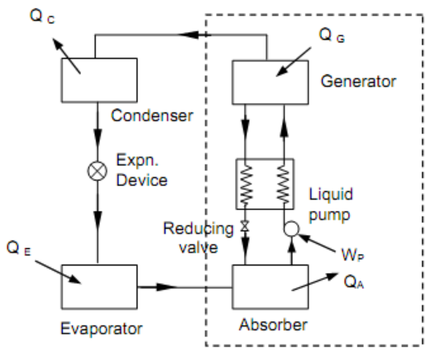

Fig. (15). Simple Absorption System

Simple Absorption System and how it Works?

(a) Condenser: Just like in the traditional condenser of the vapour compression cycle, the refrigerant enters the condenser at high pressure and temperature and gets condensed. The condenser is of water cooled type.

(b) Expansion valve or restriction: When the refrigerant passes through the expansion valve, its pressure and temperature reduces suddenly. This refrigerant (ammonia in this case) then enters the evaporator.

(c) Evaporator: The refrigerant at very low pressure and temperature enters the evaporator and produces the cooling effect. In the vapour compression cycle this refrigerant is sucked by the compressor, but in the vapour absorption cycle, this refrigerant flows to the absorber that acts as the suction part of the refrigeration cycle.

(d) Absorber: The absorber is a sort of vessel consisting of water that acts as the absorbent, and the previous absorbed refrigerant. Thus the absorber consists of the weak solution of the refrigerant (ammonia in this case) and absorbent (water in this case). When ammonia from the evaporator enters the absorber, it is absorbed by the absorbent due to which the pressure inside the absorber reduces further leading to more flow of the refrigerant from the evaporator to the absorber. At high temperature water absorbs lesser ammonia, hence it is cooled by the external coolant to increase it ammonia absorption capacity

(e) Pump: When the absorbent absorbs the refrigerant strong solution of refrigerant-absorbent (ammonia-water) is formed. This solution is pumped by the pump at high pressure to the generator. Thus pump increases the pressure of the solution to about 10bar.

(f) Generator: The refrigerant-ammonia solution in the generator is heated by the external source of heat. This is can be steam, hot water or any other suitable source. Due to heating the temperature of the solution increases. The refrigerant in the solution gets vaporized and it leaves the solution at high pressure. The high pressure and the high temperature refrigerant then enters the condenser, where it is cooled by the coolant, and it then enters the expansion valve and then finally into the evaporator where it produces the cooling effect. This refrigerant is then again absorbed by the weak solution in the absorber.

When the vaporized refrigerant leaves the generator weak solution is left in it. This solution enters the pressure reducing valve and then back to the absorber, where it is ready to absorb fresh refrigerant. In this way, the refrigerant keeps on repeating the cycle.

The pressure of the refrigerant is increased in the generator, hence it is considered to be equivalent to the compression part of the compressor.

Refrigerants

(A). Refrigerant development – a brief history

Table 1 summarizes the different refrigerants used in the last 150 years. Many of the refrigerants used during the early periods did not survive, mainly due to their toxicity.

Ammonia, however, continues to be a refrigerant of choice for food freezing applications even today in spite of its toxicity, mainly due to its excellent thermodynamic and thermal properties. Carbon dioxide used in the early days of refrigeration is again being considered as a refrigerant in spite of its high operating pressures Hydrocarbons used in the early part of the last century were quickly discontinued because of their flammability. However, hydrocarbons have made a successful comeback and are being used extensively in small domestic refrigerators and freezers in recent years. The discovery of CFCs in the late twenties revolutionized the refrigeration industry. Both CFCs and hydrochlorofluorocarbons(HCFCs) are non-toxic, possess excellent thermodynamic properties, and are non-flammable. Both CFCs and HCFCs dominated the refrigeration industry for nearly 70 years till the Montreal Protocol imposed a ban due to their contribution to ozone depletion.

TABLE-1

Period	**Refrigerants**
1800–1900	Ethyl alcohol, methyl amine, ethyl amine, methyl chloride, ethyl chloride, sulphur dioxide, carbon dioxide, ammonia
1900–1930	Ethyl bromide, carbon tetrachloride, water, propane, isobutene, gasoline, methylene chloride
1931–1990	Chlorofluorocarbons, hydrochlorofluorocarbons, ammonia, water
1990–2010	Hydrofluorocarbons, ammonia, isobutene, propane, carbon dioxide, water
Immediate future	Hydrofluorooelifins, hydrofluorocarbons, hydrocarbons, carbon dioxide, water

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In the last two decades, hydrofluorocarbons (HFCs), which possess zero Ozone Depletion Potential (ODP), have gradually replaced CFCs. Very recently; global warming due to emission of various gases into the atmosphere has been the issue being dealt with by the Kyoto Protocol. HFCs which have high Global Warming Potential (GWP) are also being banned in spite of the fact that they are ozone friendly. Hydrofluorooelifins (HFOs), which have very low GWP and invented very recently are expected to replace HFCs in many applications. A detailed discussion on the different refrigerants is given below.

(B). Desirable Properties of Refrigerants

The following are the important requirements of a refrigerant:

• Low GWP and zero ODP from a sustainable environment point of view.

• Non-toxic from human and animal safety point of view.

• Compatibility with the materials used in refrigeration systems (metals, polymers, lubricating oils, etc.) from design and operation points of view.

• Desirable thermodynamic and thermo physical properties from the operating efficiency point of view.

As already mentioned, the requirements of low GWP and ODP are relatively new as a consequence of Kyoto and Montreal Protocols respectively

Thermodynamic and Thermo physical Properties

The ideal refrigerant should have the following thermodynamic and thermo physical properties:

(a) Low condensing pressure to allow the use of lightweight materials for heat exchangers, compressors, piping, etc.

(b) Suction pressure above atmosphere for ease of leak detection and to prevent air and moisture ingress into the system.

(c) Low compression ratio to give high volumetric efficiency and low power consumption.

(d) High latent heat of vaporisation for large refrigerating effect or a small mass flow rate for a given cooling load.

(e) Small specific volume for large mass flow rate per unit volume of compression (specifically for use with positive displacement compression systems).

(f) Moderate temperature rise during compression to reduce the risk of compressor overheating and to avoid chemical reaction between refrigerant oil and other materials.

(g) Low liquid specific heat capacity to increase liquid sub-cooling prior to expansion and to minimize flash gas.

(h) High vapour specific heat capacity to reduce vapour superheat at suction.

(i) High thermal conductivity of both liquid and vapour to improve heat transfer.

(j) Low viscosity of both vapour and liquid to reduce pressure

Environmental and Safety Properties

In addition to thermodynamic and thermo physical properties, possible environmental impacts, safety, toxicity and flammability are also to be considered. In fact, environmental friendliness of the working fluid today is a major consideration. It is essential that all newly considered fluids should possess zero ODP, and low GWP. Decomposition and environmental assimilation of released substances may require many days to years to take effect. These effects are widespread and may affect locations far away from the source. Hence, these issues are to be tackled on a global basis.

On the other hand, toxicity effects are generally localised and immediate. The margin of safety from the viewpoint of toxicity depends on the degree of concentration and the duration of exposure to cause harmful effects. An ideal refrigerant should be non-toxic to humans, animals and plant life even at low concentrations and with long time exposure.

The main concern with the use of hydrocarbon refrigerants is their flammability especially in case of leaks. Small- and medium-sized refrigeration systems use ‘hermetic’

compressors(compressor and motor placed inside a single sealed shell). Large systems, on the other hand, use open-type compressors that are difficult to seal completely. Similarly, a small amount of leakage also can occur from other components such as valves, measuring and safety equipment, etc. Refrigerant leaks are therefore difficult to avoid in large systems. The degree of hazard expected by the use of a refrigerant depends on factors such as the quantity of
refrigerant used, volume of the space into which the refrigerant leaks, type of occupancy of the room, risk of naked flame or electric sparks.

Flammable refrigerants (isobutene, mixture of propane and isobutene, etc.) have been adopted widely for use in home refrigerators, including in India. Most low GWP refrigerants are flammable and their use is becoming the norm due to regulations such as the F-gas regulations in Europe which require refrigerants with GWP less than 150 to be used in new car models from this year. R1234yf, the low GWP refrigerant of choice, in the new car models has been accepted in spite of its flammability. Standards already exist on the type of safety features to be mandatorily used in plant rooms, the amount of refrigerant charge permitted to be used in plant rooms of given size (volume), etc. In cases where it is not recommended to use large amounts of hydrocarbon refrigerants (supermarkets, concert halls, auditoria, etc.), hydrocarbon refrigerants can serve as primary refrigerants in the plant room,
While secondary refrigerants such as propyleneglyocol or ethyleneglycol can be used inside the building enclosure. With the increasing concern for global warming, it is likely that the refrigeration industry will have to learn to adopt flammable refrigerants even in large applications.

Another essential characteristic is the chemical stability within the refrigeration system so that the refrigerant does not decompose when it is repeatedly compressed/expanded and condensed/evaporated. It should not react with the lubricant or system construction materials to form harmful compounds. However, once emitted, a refrigerant should not be so stable that it persists indefinitely in the atmosphere to reach the upper layers of atmosphere. The ideal refrigerant should be totally stable while in use but decompose quickly into safe substances when released. These contradictory requirements essentially limit the number of fluids that can be used as refrigerants.

***(C).* Properties of Specific Working Fluids**

The basic thermodynamic and environmental properties of different fluids that have been used or can be used as refrigerants in vapour compression refrigeration and air conditioning equipment are discussed in this section.

Chloroflurocarbons (CFC)

These are fully halogenated fluids that have high ODP and were found to be the most responsible for the creation of ozone hole. Use of formerly popular CFCs such as R12 and R11 in new equipment was banned by the Montreal Protocol. While R12 recovered from old systems may still be available, new lots of CFCs are no longer being produced.

Hydrochlorofluorocarbons (HCFC)

Unlike fully-halogenated CFCs, which contain only carbon and halogen atoms, in the case of partially-halogenated HCFCs, not all hydrogen atoms are replaced by halogen atoms. The remaining hydrogen atoms facilitate partial breakdown of the compounds in the troposphere. For this reason these compounds are less harmful to the stratospheric ozone layer, though they still have the some potential to damage the ozone layer. However, since they are known to cause global warming, HCFCs are no longer used in the industrialized countries of the West. Phase-out of HCFCs (mainly HCFC22, which is still widely used in India) is being accelerated.

Hydrofluorocarbons (HFC)

Hydrofluorocarbons contain fluorine but no chlorine or bromine in the molecule, so that their ODP is zero. Some examples of HFCs areR23, R32, R125, 134a, 143a and 152a. A problem with HFCs is that they are chemically stable and can accumulate in the atmosphere contributing to the global warming. Hence, HFCs need to be eventually replaced.

Hydrofluorooelifins (HFO)

These also belong to a class of HFCs, but are derived from unsaturated hydrocarbon molecules such as propylene. HFOs are relatively unstable, have a small atmospheric lifetime and therefore a small GWP. R1234yf and R1234ze are two HFO refrigerants invented recently. R1234yf has been widely accepted for use in cars by the automobile industry because of its very low GWP of 4. As soon as it becomes commercially available, R1234yf is expected to replace R134a, which is currently being widely used in air-conditioning plants, automobile air conditioners, domestic refrigerators, etc. There are also attempts to find mixtures of R1234yf and other HFCs such as R32 for use in other applications such as domestic air conditioners since mixtures containing R1234yf will have low GWP, typically less than 1000.

Fluoroiodocarbons (FIC)

These are a group of chemicals containing fluorine, iodine and carbon such as, trifluoromethyl iodide (CF3I), perfluoroethyl iodide (C2F5I) and perfluoropropyl iodide (C3F7I). The FICs are reported to have zero ODP and negligible GWP due to their very short life periods. These can also be used in blends. A blend of CF3I and HFC 152a (51/49mole percent) was run in a refrigerator without oil change for over 1,500 hours without apparent ill effects. Measurements showed that the energy efficiency and capacity were equal to or slightly better than CFC 12.

Hydrocarbons

Several hydrocarbons have excellent thermodynamic properties and can be used as refrigerants. Though alkenes, ketones, alcohols and ethers can be used, alkanes are the most preferred group. As already mentioned, the main concern is that most of the hydrocarbons are flammable. Here, one should note that in certain industrial applications hydrocarbons have been used as refrigerants since the beginning of the 20th century. Hydrocarbons, for instance, are used in pure or mixture forms as refrigerants in petrochemical plants and in gas liquefaction plants. In LNG plants, mixtures of methane and n-pentane are in common use.

With adequate safety precautions flammability will not pose a major problem in the usage of hydrocarbons. Home refrigerators have been sold in tens of millions worldwide, including India, during the last twenty years. The ODP of hydrocarbons is zero, while their GWP is very small. While hydrocarbons have found acceptance in Europe, these were scarcely used in USA, mainly due to commercial reasons. However the Environment Protection Agency (EPA, USA) issued ruling on December 14, 2011, approving propane, isobutene, and R-441A for use in small domestic and commercial refrigeration appliances. R-441A is a patented blend of ethane (3.1%), propane (54.8%), n-butane (36.1%) and isobutane (6.0%). This mixture is also considerably more energy efficient than HFC134a.Thus, hydrocarbons will now be allowed in domestic household refrigerators, freezers and window air conditioners in the USA.

Hydrocarbon refrigerants possess full chemical compatibility with nearly all lubricants commonly used in refrigeration systems. Good miscibility is maintained with most lubricants under all operating conditions. Due to the particularly high solubility with mineral oils, it may be necessary to use a lubricant with lower solubility or increased viscosity to compensate for possible thinning under situations where high solubility could occur. Almost all common elastomer and plastic refrigeration materials used as ‘O’ rings, valve seats, seals and gaskets are compatible with hydrocarbon refrigerants. These include neoprenes, vitons, nitrile rubbers, PTFE and nylon. Evaporators and condensers using hydrocarbons tend to be virtually of the same design and size as those used for conventional fluorocarbon refrigerants that operate at similar pressures. Heat transfer coefficients tend to be higher for most hydrocarbons. Most compressor types are also suitable for use with hydrocarbon refrigerants. Compared to CFCs or HFCs, a much smaller refrigerant charge is normally required when hydrocarbons are used.

Natural Inorganic Fluids

Ammonia is an environmentally safe but toxic working fluid which is attracting renewed attention. It possesses the most advantageous thermodynamic and thermo- physical properties needed for refrigeration. Ammonia-based compression systems, mainly for low temperature applications, are well developed. These are generally suited for industrial surroundings where sufficient knowledge and facilities exist to handle chemical leaks. There are proposals to extend its use into areas occupied by common public (e.g., comfort air conditioning, cooling of display cases in food shops, heat pumps, etc.). But this requires careful planning and design to avoid panic and accidents in case of leaks.

Water has many desirable characteristics for cooling applications such as: thermal and chemical stability, neither toxic nor flammable, high COP and high heat transfer coefficients. Disadvantages of water include sub-atmospheric pressure operation, large specific compressor displacement, limitations of evaporation temperatures above 0°C and problems of lubrication.

Air has been used commercially for aircraft cooling since a long time. In spite of the low COP, this is being used because of the operating conditions (e.g., availability of compressed air and ram effect) and stringent specifications (e.g., low weight, small size, absolute safety, zero toxicity, etc.) which are exclusive to aircrafts. In the light of the new situation created due to the ban on synthetic refrigerants, possible use of air for on-ground applications is being considered actively. It should be noted here that the technology with air as refrigerant will be totally different from that with other working fluids due to the fact that air does not undergo phase change (condensation/evaporation) at the temperature levels encountered in conventional refrigeration applications.

Use of carbon-dioxide as refrigerant started in the early years of refrigeration. It is environmentally benign. Being the by product of many energy conversion processes, it is cheap and easily available. Its use as a refrigerant can reduce its release to the atmosphere, thereby making a positive contribution to the environment. Very high operating pressure is a drawback. Because of its low critical point, most of the thermodynamic cycle operates in the single phase region. Since CO2 enters the expansion valve as a superheated vapour, it results in a large energy loss during the throttling process. Carbon dioxide is an excellent refrigerant when both heating and cooling are desired. Also, it is not preferable for use in tropical countries such as India due to the high ambient temperatures which result in high condensing pressures.

(2). LITRATURE REVIEW

Energetic and exergetic thermodynamic analysis have been widely applied to analyze the performance of cooling plants.

In 1985 Kotas described the phenomenon of energetic losses in various processes and its quantification in detailed analysis of heat and mass transfer.

In 1995 Pátek and Komflar gave a set of equations which described the proprieties of vapour-liquid balance for ammonia-water. Those equations were used for comparison with the data given in the literature work and good results were found, thus, it was a great help for analysis of these systems.

In 1996 Bejan et al. studied the application of the 1st and 2nd law of thermodynamics, particularly, the concept of exergy in the thermal system and in the qualitative evaluation of efficiencies.

In 1998 Nikolaids and Probert used the exergy method to analyse a two stage compression refrigeration system. The condensation temperature ranged from 298K to 308K, and evaporation temperature ranged from 228K to 238K. They found irreversibility rate of the plant taking into account of the effects of changes the condenser and evaporator temperature.

In 1999 Göktun analysed the performance of an irreversible heat engine which combines a vapour compression (VCR) and absorption refrigerator (AR), where the condenser of the AR unit discharged heat to the evaporator of the VCR unit. The heat released by condenser of the VCR is, then, transferred to the external medium for its heating.

In 2007 Kaynakli and Kilic gave a detailed analysis of an absorption cooling cycle functioning with lithium bromide/water. They also analysed the effect of operational temperatures, coefficient of performance (COP), effectiveness of the heat exchanger.

In 2006 Kairouani and Nehdi studied, aiming to develop a new cycle for refrigeration, in which geothermal energy is applied in a simple vapour compression system, and to analyse the thermodynamics of the cycle and its practical viability. This refrigeration system constitutes of a vapour compression system combined with an absorption system.

In 2007 Cabello et al. analysed the effect of variation of different parameters in a plant of vapour compression of a single stage. These included changes in temperature of condensation and evaporation and the degree of overheating at the inlet of the compressor.

The measure taken in general in industrial cooling is the compression by stages, when the difference is very high between evaporator and condenser. This allows, through an intermediate cooling, reducing the effort of compressor so resulting an increase in the frigorific performance of the plant. This happens, in general, by injection of a liquid between the stages, where part of the refrigerant fluid, which could be used for increasing the frigorific capacity, is applied to cool the discharge by the compressor in the first stage.

The target of the study is to do an energetic and exergetic analysis to offer a change in this usual method, by proposing a system of in absorption system integrated to the compression refrigeration system (CRS), for intermediate cooling, and increasing the frigorific capacity of refrigeration

For the compression refrigeration system (CRS) and for the integrated refrigeration system (IRS) we study the effects of evaporation temperature (T_E), condensation temperature (T_C),intermediate pressure(P_i), degree of overheating in the temperature of suction ($T_{SUCTION}$), in the power of the compressors ($\dot{W}$), in the heat transfer rates, ($\dot{Q}$), in the coefficient of performance (COP) and in the exergetic efficiency (ε).

(3). Conventional and Integrated refrigeration system

Figure.16 shows the compression refrigeration system (CRS) in use in the industrial plant with evaporators temperature of low and intermediate pressure, respectively T_{E1}=−40ºC and T_{E2}= −10º C and condensation temperature T_C= 35º C.

The evolution of NH_3 flowing in CRS initiates in the state of overheated vapour, when it leaves the compressor of high (state. 13). Ammonia is, then, liquefied in the condenser, converting into a saturated liquid (state. 14); after that, this flow divides into three different streams. The first $\dot{m}_{a1}$ passes through a expansion valve, where it is expanded in isenthalpic manner , submitted to pressure corresponding to the lowest temperature T_{E1} (state 16). This stream passes through the intercooler, and, as it flows down to the evaporator 1 absorbing heat, making it a saturated vapour (state 17) and, after that, it goes through compressor 1 leaving as overheated vapour (state 18). The second stream $\dot{m}_{a2}$ aims to inject liquid ammonia to cool partially the overheated vapour (state 18) in the exit compressor 1, this is due to the reduction of temperature of ammonia corresponding to the intermediate pressure (state 25), obtained through isenthalpic expansion in a expansion valve. As a result, there is a mixture $\dot{m}_{a1}+\dot{m}_{a2}$ overheated (state 22). Third stream,$\dot{m}_{a3}$ by crossing another expansion valve, where an isenthalpic expansion occurs, and this one is submitted to an intermediate pressure corresponding to T_{E2} (state 20). Continuing its flow, it passes through the intercooler, and by flowing down in evaporator 2 it absorbs heat, getting converted into saturated vapour (state 21).Next, it is mixed with $\dot{m}_{a1}+\dot{m}_{a2}$ overheated (state22) resulting in a mass flow rate $\dot{m} = \dot{m}_{a1}+\dot{m}_{a2}+\dot{m}_{a3}$ (state23) which passes through the compressor 2 returning to (state 13) to complete the cycle.

It is worth noting that overheating increases the specific volume of the fluid at the inlet to the compressor, thus, increasing, the work of compression. In this refrigeration cycle, the greater the overheating in the suction of compressor 2, the larger is the work needed. Here cooling by injection of liquid $\dot{m}_{a2}$ between the compression stages reduces this work, but with a reduction in the frigorific capacity, which could have been used by the evaporators, instead.

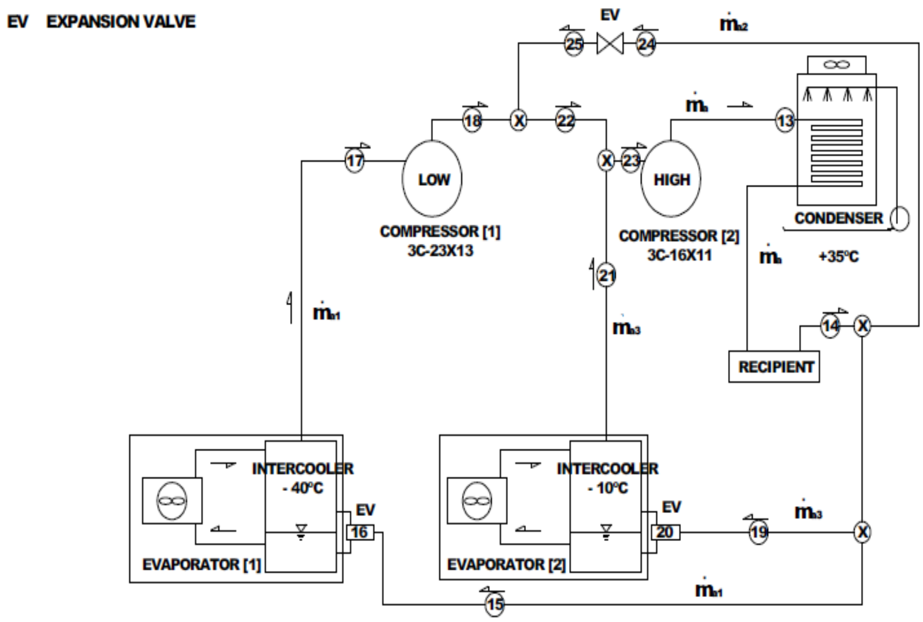

Figure 16 Schematic drawing of CRS used in industry

Figure.17 shows an integrated refrigeration system (IRS). It is similar to the conventional system, although it consists an intermediate cooling as the only difference, cold water in closed circuit, between the stages of high and low pressure compressor. Cold water is kept cool by a cooling system of absorption LiBr-H2O, integrated to the conventional system.

It consists of two systems, which can be observed in highlight in Fig.17, where the temperatures of evaporator, condenser and the assumptions for energy balance of CRSmod are the same used for CRS and for ARS.

It must be noted, that the working fluid in ARS is binary mixture of LiBr and water, and its main source of energy is the heat transferred to the generator, coming from the industrial waste material. ARS consists of the following components: a condenser, an evaporator, an absorber, two expansion valves and the pump of solution LiBr-H2O.

A brief analysis of ARS can be initiated from the pump (state 1) where flowing of the binary solution LiBr-H2O of ARS receives a strong solution from the absorber, $\dot{m}_s$, liquid to low pressure and has its pressure risen, until the pressure of the heat exchanger (state 2). There, a heat transfer between currents occur, where the strong solution $\dot{m}_s$ is heated (state 3), while the weak solution $\dot{m}_w$, hot, originally from the generator has its temperature reduced (state 5), as it returns to the absorber (state 6), after its passage through the expansion valve.

Liquid solution $\dot{m}_s$ (state 3) enters generator. The solution which leaves the generator with mass flow rate $\dot{m}_f$ (state 7) presents a fraction of 0.1 % water. It passes by the condenser, leaving as a saturated liquid (state 8); then, the solution $\dot{m}_f$ crosses an expansion valve in an isenthalpic expansion, reducing the pressure until it is obtained a temperature of saturation T_E (state 9). The solution crossing the evaporator, absorbs heat from the water in recirculation with the intermediate cooler of CRSmod, coming out as $\dot{m}_f$ as saturated vapour of low pressure (state 10). Completing the refrigeration cycle in evolution, the saturated vapour $\dot{m}_f$ of the evaporator is then absorbed by the liquid solution of the absorber expelling heat for the external medium, and returning to the initial state 1.

(A).Thermodynamic Considerations of the Conventional and Integrated Refrigeration System.

The mathematical models applied in the simulation of the systems under study were obtained dividing the systems considered (Figs.16 and 17) in the thermodynamic analysis, in various control volumes and applying to these the balance of mass, energy and exergy. In this analysis, the following considerations are made:

(a). The components of the systems are analyzed in a permanent regime.

(b). The effects of variation of kinetic energy and potential are not taken into account.

(c). Expansion through the valves is a throttle process.

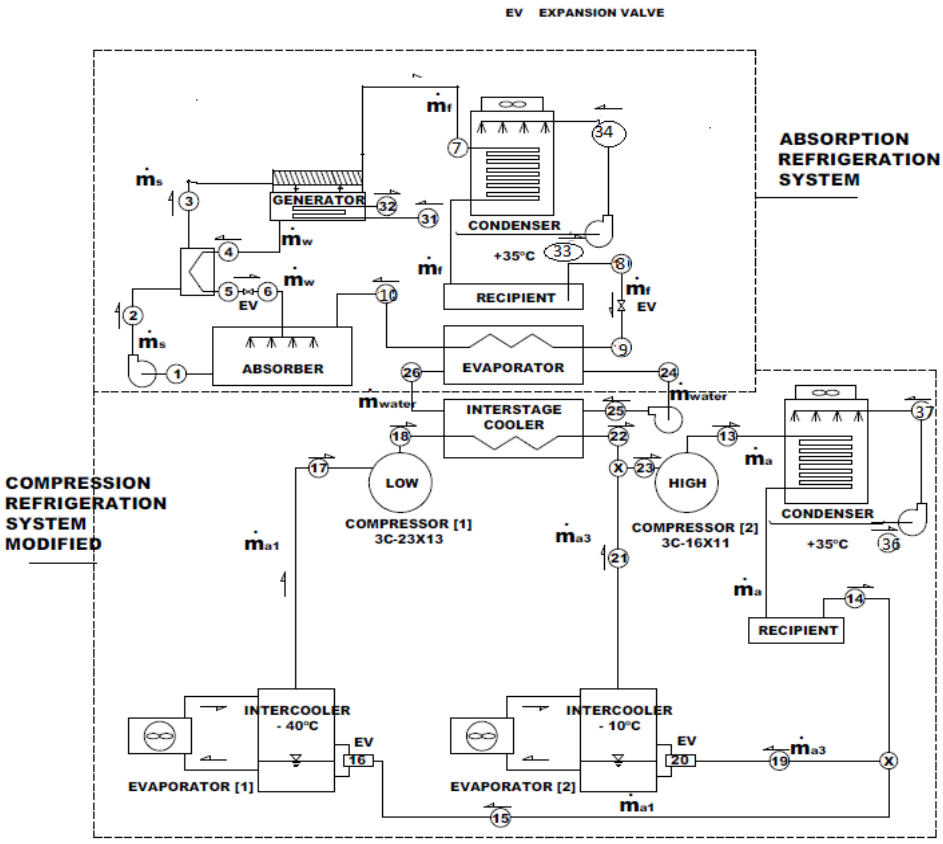

Figure 17 Schematic drawing of IRS.

(d). The working fluid is saturated liquid in the exit of condensers, saturated vapour in the entrance of the compressor of low and overheated vapour at 7^o C in the compressor of high pressure.

(e). Room temperature and pressure for the calculation of exergy are respectively T_0=293K and p_0= 1bar.

(f).In ARS the temperatures in the condenser, evaporator, absorber and generator are constant and uniform along the components.

(g).Pressures in the condenser and evaporator of ARS are balance pressures corresponding to the temperatures and concentrations of the condenser and evaporator respectively.

(h). The strong solution x_s in the exit of absorber and the weak solution x_w in the exit of the generator are balanced with their respective temperatures and pressures and the concentration of refrigerant vapour in the exit of the generator is equal to x_f=0.001

The regime of work in the refrigeration plant was obtained through instrumentation, during the operation of the system. Such data are presented in Table 1 and will be a source for the following guidelines of calculation for CRS and IRS.

In Table 1, the refrigerant fluid in each cluster (14, 15, 19 & 24) and (20 & 25), presented the same state, pressure and temperature, but with different mass flow rates..

Table 2 presents the operational data of the absorption refrigeration system (ARS). The temperatures of the solutions are determined numerically.

Table 1 - Data for CRS

Power input for the low-pressure compressor: 32.95 kW

Power input for the high-pressure compressor : 61.23 kW

Ambient temperature: 25ºC

Ambient pressure: 101 kPa

T_{in} and T_{out} cooling water: 25ºC and 30ºC

Refrigerant: NH_3

Point	Fluid state	T [ºC]	P [kPa]

13	overheated vapour	112.5	1351
14,15,19,24	Saturated liquid	35	1351
16	Liquid + vapour	-40	71.66
17	Saturated vapour	-40	71.66
18	overheated vapour	48.9	290.8
20,25	Liquid + vapour	-10	290.8
21	Saturated vapour	-10	290.8
23	overheated vapour	7	290.8

This installation presents data with the following considerations:

1- The frigorific capacity of ARS is the same as the heat removed from the intermediate cooler of CRSmod obtained by thermal balance of the latter.

2- They were attributed practical values for the range of the process, concentrations, temperatures and pressures applied and verified in COP which would meet the needs of the operation.

Table 2 - Data for ARS

$\dot{Q}_{resf}$ Heat transfer rate by the intercooler:8.918kw

T_0 Ambient temperature: 25ºC

P_0 Ambient pressure: 101 kPa

T_{in} ,T_{out} (Fig.2) cooling water: 25ºC and 30ºC

T_{in} ,T_{out} (Fig.2) heating water: 96ºC and 90ºC

Fluid : LiBr-H2O

Δx= strong solution - weak solution: 0.05

Point	**State**	**T[ºC]**	**x g_{LiBr}/g_{sol}**	P[kPa]	solution
1	Liquid	33.8	0.55	0.823	strong
2	Liquid	33.8	0.55	7.91	Strong
3	Liquid	63.1	0.55	7.91	Strong
4	Liquid	85.7	0.59	7.91	Weak
5	Liquid	52.5	0.59	7.91	Weak
6	Liq.+vap.	43.3	0.59	0.823	Weak
7	Vapour	76	0.0073	7.91	Solution
8	Liquid	41.3	0.0073	7.91	Solution
9	Liq.+vap.	4.2	0.0073	0.823	Solution
10	Vapour	4.2	0.0073	0.823	Solution

(B). Functions for CRS

The balance of mass and energy for each volume of control are presented, where the numeric rates represent the states of the working fluid in the cycle of Figure 16:

Condenser Unit - The condenser cools and condenses the overheated vapor coming from the compressor of high. This operation is carried out transferring the heat of the refrigerant fluid to the medium. Water is used as refrigerant fluid.

$$\dot{Q}_c = \dot{m}_a(h_{13}-h_{14}) \quad (1)$$

Evaporator Unit 1 – It is a heat exchanger where the evaporation of the refrigerant liquid takes place to the lowest pressure and temperature of the compression system.

$$\dot{Q}_{E1} = \dot{m}_{a1}(h_{17}-h_{16}) \tag{2}$$

Evaporator Unit 2 – Evaporator 2 is responsible by the removal of heat, temperature and intermediate pressure of the compression system.

$$\dot{Q}_{E2} = \dot{m}_{a3}(h_{21}-h_{20}) \tag{3}$$

Compressor Unit 1 – The amount of energy provided to the system in the time unit in compression represents the mechanic power of the first stage.

$$\dot{W}_{1} = \dot{m}_{a1}(h_{18}-h_{17}) \tag{4}$$

The mechanic power required from the compressor 1 is obtained, using the specifications of the manufacturer for the compressor model applied (MADEF 3C-16x11), according to the procedure described by Pohlmann (1964), who makes the connection to the variance of power for different operational conditions for temperature and pressure.

Compressor Unit 2 – represents the mechanic power in the 2nd stage.

$$\dot{W}_{2} = \dot{m}_{a2}(h_{13}-h_{23}) \tag{5}$$

Likewise, the mechanic power required by compressor 2 is obtained from the specifications of the manufacturer for the compressor model applied (MADEF 3C-23x13).
Mixtures - The refrigerant fluid compressed by the low pressure (state 18) is rarefied by injecting a liquid (state 25) resulting in state 22. The fluid in this state is mixed with the vapour in state (21) and, by the end, in state (23) it is aspired by compressor 2.

$$\dot{m}_a = \dot{m}_{a1} + \dot{m}_{a2} + \dot{m}_{a3} \tag{6}$$

$$(\dot{m}_{a1} + \dot{m}_{a2})\, h_{22} = \dot{m}_{a1} h_{18} + \dot{m}_{a2} h_{25} \tag{7}$$

$$\dot{m}_a h_{23} = \dot{m}_{a3} h_{21} + (\dot{m}_{a1} + \dot{m}_{a2})\, h_{22} \tag{8}$$

With the pressures of the condenser, evaporators and entrance and exit temperatures of the compressors, obtained through reading in the instrumentation existing in the frigorific plant, it is determined the following properties: temperatures of saturation, enthalpies, entropies and specific volumes. The mass flow rates $\dot{m}_{a1}$ and $\dot{m}_a$ are determined with the equations (4) and (5). As to equations (6, 7 and 8) the mass flow rates $\dot{m}_{a2}$ and $\dot{m}_{a3}$ are determined, and the enthalpy h_{22} . The rates of heat in the evaporators 1 and 2, respectively $\dot{Q}_{E1}$ and $\dot{Q}_{E2}$, and the heat rate in the condenser, $\dot{Q}_c$ is obtained by equations (1, 2 and 3), respectively.

The polytropic coefficient of the 1st stage (n_1) is obtained by Eq. (9) and about the condition obtained from the reading of pressures and temperatures obtained in the frigorific plant by the compressor 1. This polytropic coefficient, n_1 , is then considered constant for all the numeric simulations carried out in this work. An identical procedure is made for the calculation of the polytropic coefficient for the compressor in the 2nd stage, n_2 through eq. (10).

$$(n_1-1)/n_1 = (\log T_{18}/T_{17})/(\log p_{18}/p_{17}) \tag{9}$$

$$(n_2-1)/n_2 = (\log T_{13}/T_{23})/(\log p_{13}/p_{23}) \tag{10}$$

In the numeric simulation of the compression refrigeration system (CRS), they are admitted for the working fluid different values for the inlet and outlet pressures for the compressors, as well as for the inlet temperatures in the compressors. The outlet temperatures of the compressors were obtained by equations (9 e 10), with the assumption that the polytrophic coefficients might be constant. With these pressures and temperatures, they are determined the other properties.

The coefficient of performance (COP) matches the frigorific capacity with the mechanical power provided to the system

$$COP_{CRS} = (\dot{Q}_{E1} + \dot{Q}_{E2}) / (\dot{W}_1 + \dot{W}_2) \quad (11)$$

Figure 18 presents diagram p-h of ammonia for CRS, where the characteristics of the refrigerating in any state are represented by the continuing lines.

In this diagram it can be observed that the inlet and outlet states of the refrigerant fluid ammonia, provided by readings (pressure and temperature) in the compressors of high pressure (states 23 and 13) and low pressure (states 17 and 18) installed in the refrigerating plant.

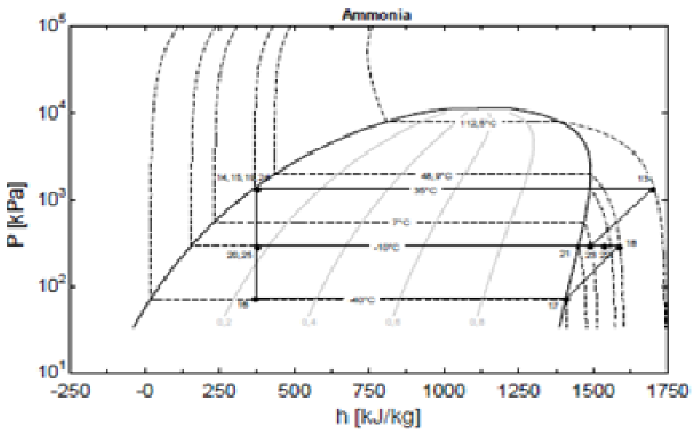

Figure18 Diagram p-h: ammonia of CRS

(C). Energetic functions for IRS

The methodology applied in the thermodynamic analysis of IRS allows to considerate the CRSmod and the ARS as two subsystems which, for a matter of simplicity, will be analyzed separately.

(D). Energetic Functions of CRSmod

As for CRS, eq. (1) - (5), (9) and (10) are valid under the assumptions mentioned in the previous section. For the mixture which is admitted in the compressor of high pressure we have: mixture

$$\dot{m}_{a} = \dot{m}_{a1} + \dot{m}_{a3} \tag{12}$$

$$\dot{m}_{a}h_{23} = \dot{m}_{a3}h_{21} + \dot{m}_{a1}h_{22} \tag{13}$$

Unit-cooler-

$$\dot{Q}_{resf} = \dot{m}_{a1}(h_{18}-h_{22}) \tag{14}$$

With the pressures and temperatures of the working fluid in the entrance and exit of the compressors they are determined the properties: saturation temperatures, enthalpy and specific volumes.

Provided the mechanical powers and with the eq. (4) and (5) they are determined the flows $\dot{m}_{a1}$ & $\dot{m}_{a3}$ and with the eq. (12) and (13) the flow $\dot{m}_{a}$ and enthalpy h_{23} are determined. The heat changes of the evaporators (1) & (2) respectively $\dot{Q}_{E1}$ & $\dot{Q}_{E2}$ and the heat rates in the condenser $\dot{Q}_{C}$ are obtained by eqn.(1-3).

(E). Energetic Functions for ARS

For the thermodynamic analysis of the absorption system are applied the principles of mass conservation, species and energy of each component of the system. Each component is treated as a control volume with inlet and outlet of flows, and with interactions of heat and work transfer, where $\dot{m}$ is the mass flow rate of the solution LiBr-H2O, x is the concentration of mass of LiBr in the solution, and h is the enthalpy of the solution LiBr-H2O, corresponding to the inlet and outlet state of each component. The sub index of the solution "f"(fraction), "s"(strong), "w"(weak), "L"(liquid), & "v"(vapour) mean a species of concentration "x" of the solution LiBr-H2O.

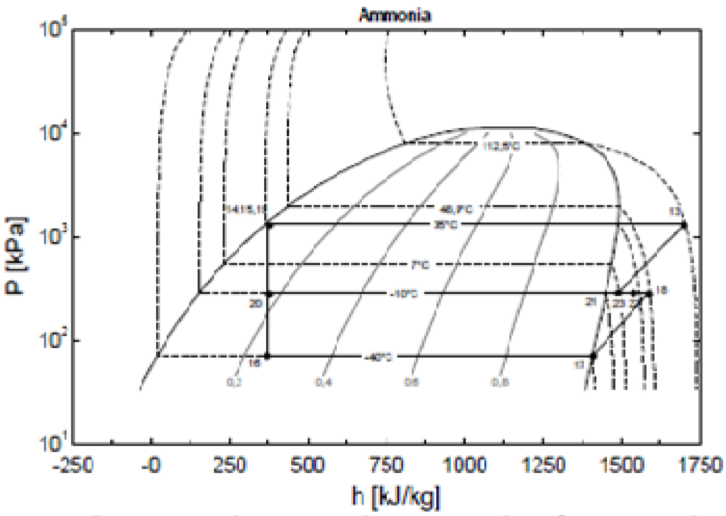

Figure 19 Diagram p-h: ammonia of CRSmod

Unit – Condenser

$$\dot{Q}_C = \dot{m}_f(h_8 - h_7) \tag{15}$$

Unit – Evaporator

$$\dot{Q}_E = \dot{Q}_{resf} \tag{16}$$

$$\dot{Q}_E = \dot{m}_f(h_{10} - h_9) \tag{17}$$

Unit – Absorber

$$\dot{m}_s = \dot{m}_w + \dot{m}_f \tag{18}$$

$$\dot{m}_s x_1 = \dot{m}_w x_6 + \dot{m}_f x_{10} \tag{19}$$

$$\dot{Q}_a = \dot{m}_f h_{10} + \dot{m}_w h_6 - \dot{m}_s h_1 \tag{20}$$

Unit - Solution pump

$$h_2 = h_1 + v_1(p_2 - p_1) \tag{21}$$

$$\dot{W}_b = \dot{m}_s(h_2 - h_1) \tag{22}$$

Unit – Heat exchanger

$$\dot{m}_s x_2 + \dot{m}_w x_4 = \dot{m}_s x_3 + \dot{m}_w x_5 \tag{23}$$

$$\dot{m}_s h_2 + \dot{m}_w h_4 = \dot{m}_s h_3 + \dot{m}_w h_5 \tag{24}$$

Unit - Generator

$$\dot{Q}_G = \dot{m}_w h_4 + \dot{m}_f h_7 - \dot{m}_s h_2 \tag{25}$$

Coefficient of performance of IRS

$$COP = (\dot{Q}_{E1}+\dot{Q}_{E2})/(\dot{W}_1+\dot{W}_2+\dot{W}_b+\dot{Q}_G) \tag{26}$$

(F). Exergetic functions of CRS

The exergetic analysis is obtained by the combination of the first and second law of thermodynamic, where Kotas (1985) states that exergy is the standard of quality for energy, and it is equal to the maximum useful work which can be obtained by a given form of energy, using the environment (P_0,T_0) as reference.

The main connections to be made in the thermodynamic exergetic analysis of flow are the balance of mass and exergy in permanent regime, not taking into consideration the variation of kinetic and potential energy, and according to Bejan et al. (1996) the physical energy can be calculated in mass basis (e^{PH}) as in:

$$e^{PH}=(h-h_0)-T_0(s-s_0) \tag{27}$$

Where h_0 and T_0 are properties assessed in temperature T_0 and environmental pressure P_0.

In this conception, as a product we have the energy rates of the evaporators 1 and 2, as fuel to the electric power provided to the compressors 1 and 2.

Unit –Evaporator 1

$$\dot{E}_{17} - \dot{E}_{16} = \dot{m}_a(\dot{e}_{17}-\dot{e}_{16}) \tag{28}$$

Unit –Evaporator 2

$$\dot{E}_{21} - \dot{E}_{20} = \dot{m}_{a3}(\dot{e}_{21}-\dot{e}_{20}) \tag{29}$$

Unit – Compressor 1st stage

$$\dot{E}_{18} - \dot{E}_{17} = \dot{m}_{a1}(\dot{e}_{18}-\dot{e}_{17}) \quad (30)$$

$$\dot{W}_{1e} = \dot{m}_{a1}(h_{18}-h_{17})/\eta_C \quad (31)$$

Unit – Compressor 2nd stage

$$\dot{E}_{13} - \dot{E}_{23} = \dot{m}_{a}(\dot{e}_{13}-\dot{e}_{23}) \quad (32)$$

$$\dot{W}_{2e} = \dot{m}_{a}(h_{13}-h_{23})/\eta_C \quad (33)$$

Where $\dot{W}_{1e}$ & $\dot{W}_{2e}$ are the electric powers provided to the mechanic compressors of the 1st & 2nd stage, and the mean average performance (η_C=0.504) corresponds to the product of thermal performance
(η_t=0.7), mechanic (η_m=0.8) and electric (η_e=0.9).

In order to simplify the simulation and analysis of heat transfer among the system and its environments, the contribution of the work in water pump and the losses by friction in the system were neglected. Thus, the exergetic efficiency ε which is the correlation between the product and the fuel summarizes itself. (Fig. 1):

$$\varepsilon_{CRS}=[(\dot{E}_{17}- \dot{E}_{16})+ (\dot{E}_{21}- \dot{E}_{20})]/(\dot{W}_{1e}-\dot{W}_{2e}) \quad (34)$$

(G). Exergetic functions IRS

In the exergetic analysis they were applied the balance of mass and exergy in permanent regime, not taking into account the variation of kinetic and power energy.

In this conception, we have as a product the energy rates of the evaporators 1 and 2, as fuel: the electric power provided to the compressors, the electric power provided to the pump, the solution ammonia-water, the energy rate provided to the generators. As the loss, the process of cold water consumption of IRS coming from the exterior medium is taken into account.

(H). Exergetic functions of CRSmod

Unit – Evaporator 1

$\dot{E}_{17} - \dot{E}_{16} = \dot{m}_{a1}(\dot{e}_{17} - \dot{e}_{16})$ (35)

Unit – Evaporator 2

$\dot{E}_{21} - \dot{E}_{20} = \dot{m}_{a3}(\dot{e}_{21} - \dot{e}_{20})$ (36)

Unit – Compressor 1st stage

$\dot{W}_{1e} = \dot{m}_{a1}(h_{18} - h_{17}) / \eta_C$ (37)

Unit - Compressor 2nd stage

$\dot{W}_{2e} = \dot{m}_{a}(h_{13} - h_{23}) / \eta_C$ (38)

(I). Exergy of ARS

It is taken from the operational condition of absorption refrigeration system according to Palacios-Bereche(2007b) and listed in table-5.

(4). RESULTS AND DISCUSSION

Table 3 shows for each state, the values "$\dot{m}$, h, and e" in each position of interest of CRS, in the operational conditions shown in Fig. 1. The pressures and temperatures corresponding to the states shown in Tables 3 and 5, are found respectively in Tables 1 and 2.

Table 3 - State-point properties of NH_3

State	h[kj/kg]	$\dot{m}$[kg/s]	e[kj/kg]
13	1704	0.1459	864.3
14	366	0.1459	798.7
15	366	0.09134	798.7
16	366	0.09134	736.7
17	1408	0.09134	446.9
18	1589	0.09134	631.3
19	366	0.04849	798.7
20	366	0.04849	778.3
21	1450	0.04849	634.7
22	1514	0.09738	629.6
23	1493	0.1459	630.6

Table 4 shows the results of the energetic and exergetic balances, of (COP) and exergetic efficiency (ε) of CRS.

Table 4 - Results of the thermodynamic balances of COP and ε of CRS

Energetic			Exergetic	
$\dot{Q}_{E1}$	$\dot{Q}_{E2}$ [kw]	$\dot{Q}_{C}$ [kw]	$\Delta\dot{E}_{E1}$ [kw]	$\Delta\dot{E}_{E2}$ [kw]
95.15	52.57	195.2	26.47	6.96
$\dot{W}_1$[kw]	$\dot{W}_2$[kw]	COP_{CRS}	$\dot{W}_{1e}$[kw]	$\dot{W}_{2e}$[kw]
16.61	30.86	3.112	32.95	61.23
		ε_{CRS}=0.355		

Table 5 shows the properties of state and mass flow of IRS.

Comparing the results of Tables 3 and 4, obtained both for the same operational conditions (T_{E1}= −40ºC , T_{E2}=−10ºC and T_C=35ºC) it is observed that for IRS values lower than 9.7 % in COP regarding to CRS. However, in the efficiency(ε) IRS presented an increase of 11.88% regarding to CRS. This increase in the efficiency is due to an increase in the exergetic frigorific capacity (ΔE_1+ ΔE_2) of 34.46% regarding to CRS which accounts for applying IRS. Through the simulation of several operational conditions it was investigated the influence of the temperatures in obtaining exchanged heat of the components, power of compressors, performance (COP) and efficiency (ε) of the facilities. All of these results were presented in graphs.

Table 5 - State-point properties and mass flow rate of IRS

Mixture LiBr+H_2O			
State	h[kj/s]	$\dot{m}$[kg/s]	e[kj/s]
1	84.7	0.0556	502.88
2	84.8	0.0556	502.88
3	144.2	0.0556	507.16

4	205.6	0.0518	575.05
5	141.4	0.0518	566.82
6	141.4	0.0518	565.20
7	2641.6	0.00382	176.71
8	173	0.00382	49.92
9	173	0.00382	39.38
10	2508.2	0.00382	-129.57

Fluid:NH_3

State	h[kj/s]	$\dot{m}$[kg/s]	e[kj/s]
13	1672	0.1893	857.6
14	366	0.1893	798.7
15	366	0.1216	798.7
16	366	0.1216	736.7
17	1408	0.1216	446.9
18	1589	0.1216	631.3
19	366	0.06775	798.7
20	366	0.06775	778.3
21	1450	0.06775	634.7
22	1516	0.06775	629.5
23	1493	0.1893	630.6

Table 6 shows the results of the energetic and exergetic balances, of (COP) and efficiency of IRS.

Table 6 - Results of the thermodynamic balances of COP and ε of IRS

	Energetic	
$\dot{Q}_{Resf}$[kw]	$\dot{Q}_G$[kw]	$\dot{Q}_{Ca}$[kw]
8.918	12.72	9.43
	$\dot{Q}_A$[kw]	$\dot{W}_b$[kw]
	12.35	0.055
$\dot{Q}_{E1}$[kw]	$\dot{Q}_{E2}$[kw]	$\dot{Q}_C$[kw]
126.6	73.45	437.8
$\dot{W}_1$[kw]	$\dot{W}_2$[kw]	COP_{IRS}
22.1	34.07	2.81
	Exergetic	
ΔE_{E1}[kw]	ΔE_{E2}[kw]	
35.23	9.72	
$\dot{W}_{1e}$[kw]	$\dot{W}_{1e}$[kw]	
43.66	32.95	
	ε_{IRS}=0.3972	

Figure 20, for CRS and IRS, presents the variation of refrigerating capacities in function of temperature T_{E1} of the refrigerating fluid. It is observed that as the temperature varies (T_{E1}), it results, for CRS as well as for IRS, in the same tendencies of variation of heat rates.

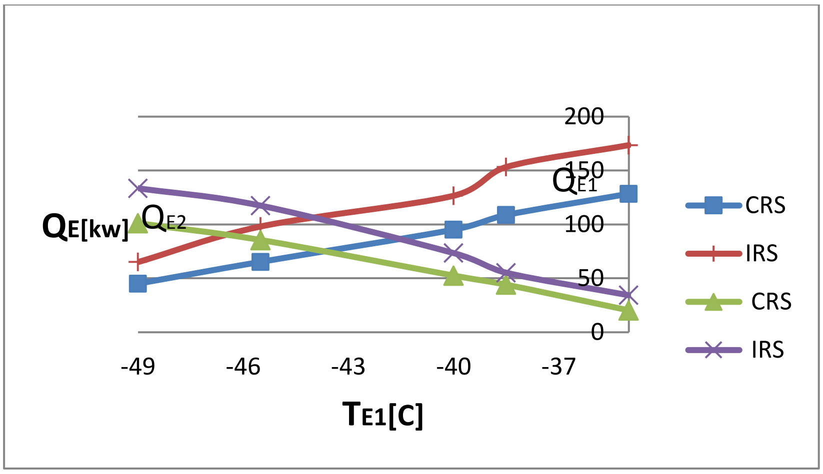

Figure 20.Variation of refrigerating capacity with T_{E1}

Fig.6, shows the variation of performance (COP) and efficiency (ε) in terms of: condensation temperature, T_C , [Fig. 21(a),(b)]; temperature of the evaporator 1, T_{E1} [Fig. 21(c),(d)] and the evaporator 2, T_{E2} [Fig. 21(e),(f)].

In Fig. 21(a) & (b), it is observed that the increase of T_C reduced the performance, and the efficiency of the two systems analyzed. This is due to an increase in the mechanic power of the compressors and the reduction for the heat rate removed from the evaporators. It is also noted that the lower values of performance for COP_{IRS} COP_{CRS} , and higher values of efficiency ($\varepsilon_{IRS} > \varepsilon_{CRS}$).

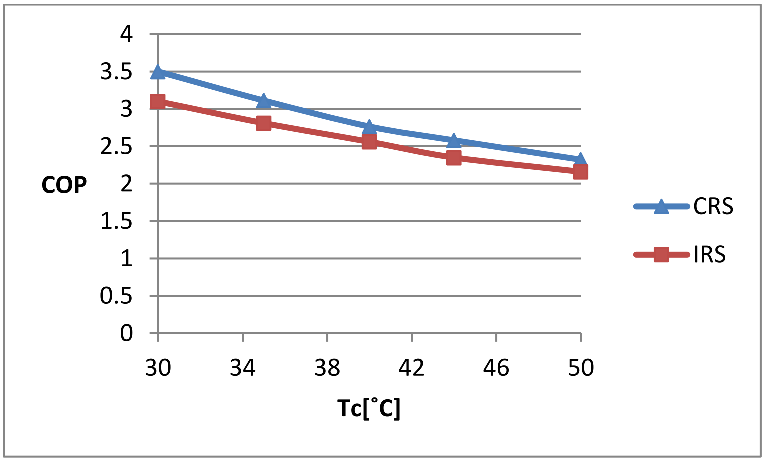

Figure.21(a)

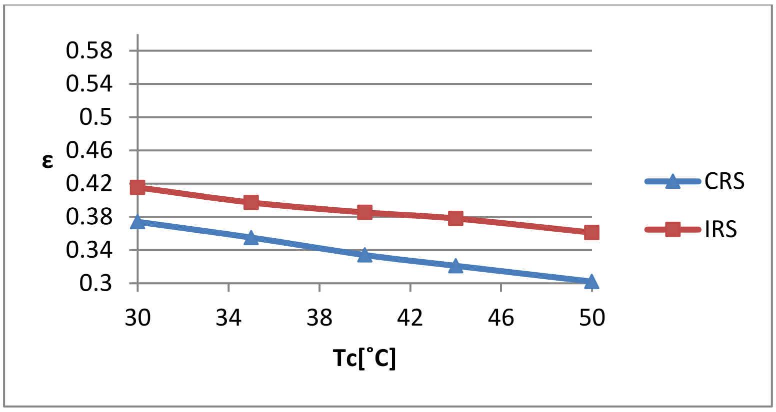

Figure.21(b)

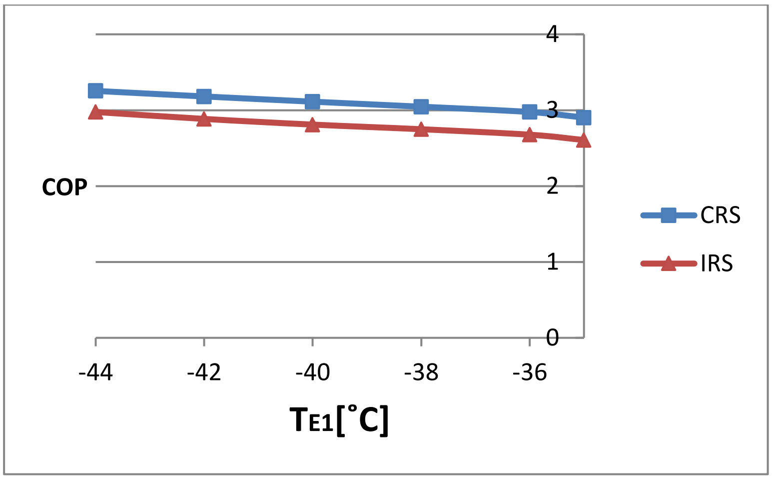

Figure.21(c)

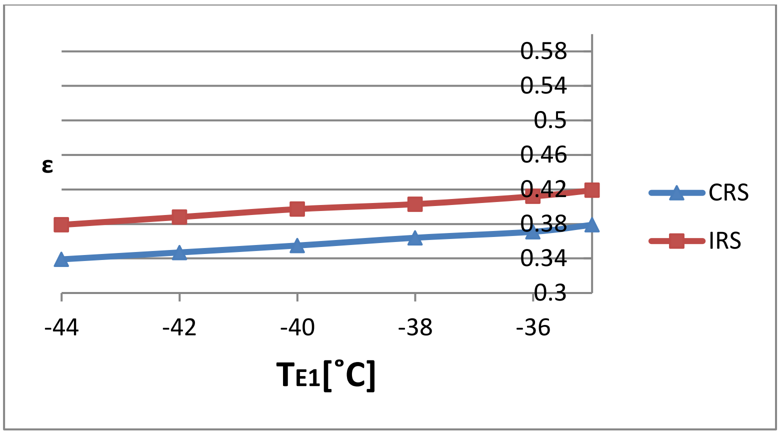

Figure.21(d)

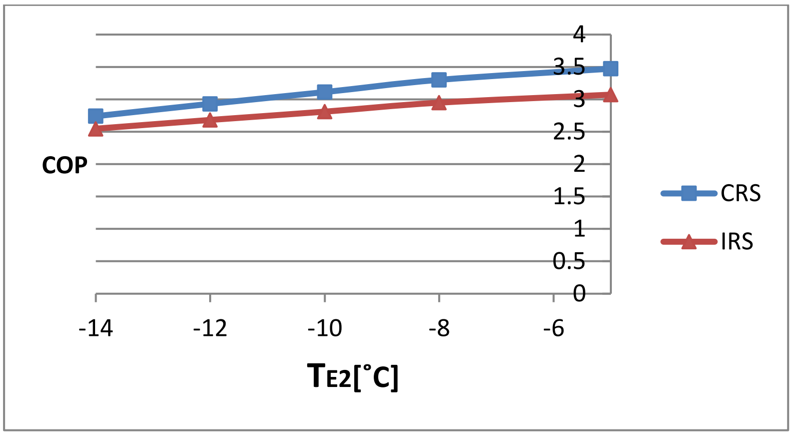

Figure.21(e)

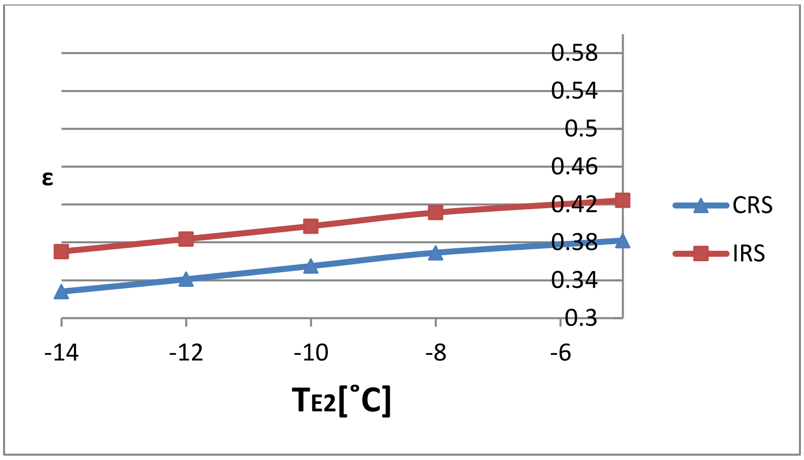

Figure.21(f)

Figure 21- Variation of the performance (COP) and efficiency (ε) in terms of: (a),(b) condensation temperature, T_C , (c),(d) temperature of the evaporator 1, T_{E1}, and (e), (f) the evaporator2 T_{E2}.

Fig. 21(c)& (d), shows a reduction of COP for systems CRS and IRS and an increase of efficiencies ε_{CRS} and ε_{IRS} , as it increases T_{E1} of the evaporator 1, maintaining T_{E2} = -10ºC. For the two systems CRS and IRS, the power of the compressors and the heat exchanged by the evaporators of the two systems increase, and as the relative increase of the compressors is larger it occurs a reduction in COP. As for the efficiencies, they increase due to an increase of the product (frigorific capacity) obtained of the exergetic balance.

Fig. 21-(e) & (f), shows that as T_{E2} of evaporator 2 increases there is a rise of COP and efficiency ε_{CRS} and ε_{IRS} of the systems CRS and IRS. In these two systems the power of the compressors and the total heat rate of the evaporators rise, but in order to raise the value of COP. The efficiencies of both the systems rise owing to the growth of the product (frigorific capacity) obtained from the exergetic balance, even occurring a raise in the heat rate of the generator $\dot{Q}_G$ in IRS.

(5). CONCLUSION

This work gives energetic and exergetic thermodynamic analysis of two refrigeration systems, which are, Conventional refrigeration system (CRS) and Integrated refrigeration system (IRS).

It carries out a comparison between the two systems. The main outcome is that IRS gives efficiency 11.89% higher than CRS. This increase in efficiency is due to increase in the exergetic refrigerating capacity ($\Delta E1 + \Delta E2$) by 34.46 % regarding to CRS. With the help of simulation of several operational conditions, it investigates the influence of temperatures on heat transfer in different components, power of the compressors, performance (COP) and efficiency (ε) of the systems, Here results prove that COP of IRS are lower than that of CRS, but, the exergetic efficiency of the IRS system is higher than that of CRS system.

In the energetic analysis, IRS has greater refrigerating capacity by 35.42% regarding to the conventional system. It is of great advantage that thermal energy provided is of low cost makes the IRS more attractive.

These are very important results in the context of thermodynamic optimization and of thermo-economic assessment of the refrigerating systems, where the lower operational cost which will be useful in future study, makes the application of the integrated refrigeration system worthy.

FUTURE SUGGESTIONS

(a). Different refrigerants can be tested in CRSmod cycle so that we can get best performing combination.

(b). Effect of using more than two compressors can be testified.

(c). A detailed cost analysis may be done to find out effectiveness in terms of cost.

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