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Thermodynamic analysis of one and two stages absorption chiller powered by a cogeneration plant

Análisis termodinámico de un chiller de absorción de 1 y 2 etapas de una planta de cogeneración

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Abstract

Thermodynamics models of a single and a non-common double stage ammonia-water absorption chiller that use waste heat (from three reciprocating engines of 8.7 MW each one) are developed to analyze the performance of the chiller for different operative conditions. A comparison of a single stage refrigeration system with the two stages proposed system is performed in this paper. The coefficient of performance (COP) obtained for both systems are the same, but the heat flux removed from the cooling media with the two-stage system increase from 1.3MW (single stage) until 1.6 MW due to the heat recovered increased with the second generator. The heat recovered used by the chiller was 3.8 MW, and the utilization factor of the cogeneration plant was 58.11%, and the cooling capacity of the equipment was 1,623 kW. Finally, the estimated economics savings for electric power due to the implementation of the absorption chiller that uses exhaust gases in place of a common refrigeration system by vapor compression with the same cooling capacity was 142,000.00 USD/year.

Keywords: Absorption chiller, cogeneration, heat recovery refrigeration, waste energy, COP.

Resumen

Se han desarrollado modelos termodinámicos de enfriadores de agua por absorción de una etapa y ciclo no común de dos etapas que usan calor de desecho (de motores de combustión interna de 8.7 MW cada uno) para analizar las condiciones de operación de los equipos. Se ha realizado la comparación del sistema de una etapa con el sistema propuesto (2 etapas) en esta investigación. El coeficiente de desempeño (COP) obtenido para ambos sistemas fue el mismo pero el calor removido del espacio refrigerado aumento de 1.3 MW (una etapa) a 1.6 MW (dos etapas) debido a que se recupera más energía residual utilizando un segundo generador. El calor residual aprovechado por el equipo de refrigeración fue de 3.8 MW y el factor de planta del proceso de cogeneración fue de 58.11 % y la capacidad de refrigeración del equipo fue de 1,623 kW. Finalmente, los ahorros económicos estimados por concepto de energía eléctrica que se tienen por implementar el sistema de refrigeración por absorción que utiliza gases de escape como fuente de energía en lugar de un equipo común de refrigeración por compresión de la misma capacidad son 142,000.00 USD/año.

Palabras clave: Absorción, calor recuperado, cogeneración, COP, energía desperdiciada.

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1. Introduction

The vapor compression refrigeration system is widely used in the refrigeration industry. In recent years, the scenario for energy demand in industrial cooling has undergone significant changes, the most significant among them, is the ban on chlorofluorocarbon (CFCs) as cooling agents because of their effect on the ozone layer. Thus, the use of alternative fluids for compression refrigeration systems were studied, and now the manufactures of refrigeration equipment use other types of fluids, for example isobutane (R600a) [1]. The disadvantages of the compression system are high electricity consumption, high operation costs and pollution. The absorption refrigeration systems (ARS), which are powered by heat instead of electricity, could be a good alternative to reduce electricity consumption and obtain economic revenues [2–4].

The absorption refrigeration systems (ARS) were developed to use thermal energy as a cycle input instead of electricity. Two-stage ARS have been developed to improve the operation of these systems [2, 5, 6]. Both systems work with two fluids: the refrigerant and the fluid to absorb the refrigerant in a solution to increase the pressure with a pump, the most common pair of fluids detailed in the bibliography are lithium bromide/water (H_2O as refrigerant and LiBr as transport medium) and ammonia/water (NH_3 as refrigerant and H_2O as transport medium). The ammonia/water systems are used to obtain low evaporation temperatures (for example freezing application) [7]. Hence, the ARS could be a good alternative to reduce the electricity consumption of industrial cooling and obtain economic revenues [2–4].

Colorado and Rivera [6] compared a vapor compression refrigeration system with a hybrid system (compression/absorption) with the first and second laws of thermodynamics. They used R134a and CO_2 as refrigerants of the compression systems and a solution of H_2O/LiBr in the absorption cycle. The hybrid system has a cascade heat exchanger where the condenser compression system is the evaporator of the absorption system. The main objective was to reduce the energy used in the compressor. The research results show that the hybrid system consumes 45% less electricity than the simple compressor cycle. Also, the coefficient of performance (COP) obtained in the hybrid system is higher with R134a.

One of the advantages of ARS is the possibility to use various heat sources as input for the generator. Said et al. [8] designed and built a solar powered absorption chiller with NH_3/H_2O to the Saudi Arabian climatic conditions. The system aim was to produce ice and cool a building's enclosed main hall which presents 10kW of cooling load. During the test, the chiller highest cooling capacity observed was of 10.1 kW, and it produced ice despite the high condensation temperature (45 °C) due to summer climatic conditions. Other authors as Wang *et al.* [9] studied the optimal heat sources for different applications of ARS.

The waste heat of exhaust gases of boilers, combustion engines, turbines can be used as heat source for an absorption system. Du *et al.* [10] built a prototype of a single stage ammonia/water refrigeration system that used waste heat from a diesel engine with an active open heat pipe method, designed to obtain stability of the available heat. The authors designed the heat exchanger to recover waste energy for a specific capacity, and they combined condensation and absorption process in one unit which was cooled by circulated precooled solution. The system had a COP of 0.53, and the cooling capacity was 33.8 kW for a temperature of exhaust gases of 567 °C.

Nowadays, manufacturers such as AGO are building absorption refrigeration systems that can be coupled with a heat exchanger to use solar collector or heat recovery hot water regenerator (HRHWG) with energy of exhaust gases [11]. This kind of systems presents different advantages compared with power generation systems such as: i) increasing energy efficiency with co/tri-generation, ii) reduction of CO_2 emissions, and iii) substitution of high-quality electric power with "low-grade" thermal energy in the form of waste heat. Moreover, among economic reasons are related to kind of systems such as: reduction of operating costs by using waste heat as the power source and minimal use of electricity.

The main objective of this work is to develop a thermodynamic model of a single stage and AGO's (non-conventional) two-stage system of ammonia/water absorption chiller designed and built by AGO [11] to analyze the performance of the chiller for different operation conditions. The absorption chiller uses waste energy recovered from the exhaust gases of reciprocating engines.

1.1. Description of complete cogeneration system

Figure 1 shows a scheme of the cogeneration power plant (CHP). In June 2017 the plant was installed by the "*Unión Energética del Noroeste SA de SV*" company, which sells electricity, and it is subsidiary of "*Negocio Agrícola San Enrique SA de SV*" company. The last one uses chillers to refrigerate its farm products. Both companies are in Agua Prieta in the federal state of Sonora, north-west in Mexico.

The CHP has three reciprocating engines with a total output of 26.1 MW (e) (each engine of 8.7 MW (e)) and 9.8 MW (t) is recovered from the exhaust gases and high-temperature refrigeration circuit. A heat recovery steam generator (HRSG) recovered 6 MW (t) producing vapor and a heat recovery hot water generator (HRHWG) recovered 3.8 MW(t).

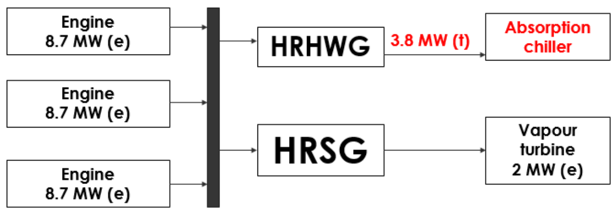

Figure 1. Powered cogeneration plant studied.

The vapor produced with the HRSG is used in a vapor turbine to produce 2 MW (e). The hot water produced in the HRHWG is used as an input of the absorption chillers to generate cold air for the refrigeration of warehouses where agricultural products from the hot Sonora area are conserved.

2. Theoretical Background

2.1. Compression versus absorption

Energy is required to transport heat from a low-temperature to a high-temperature bulb. A fluid refrigerant undergoes a series of thermodynamic transformations. Each refrigerant has a defined and different behavior, and as it is well known, the cycles prevent continuous replenishment of the refrigerant [12]. The main methods of industrial cooling are compression and absorption.

The compression cycle is based on the conventional refrigeration cycle. It transfers heat from the cooling space to the environment using electricity as its input. It works by increasing and lowering the pressure of the refrigerant, changing its saturation temperature. The compression system has a condenser (outdoor unit), an evaporator (inside the cooling space), an expansion valve and a compressor (transports the refrigerant) [13]. Its performance is high considering that the system input is electrical energy and the output is heat removed from the evaporator. These devices work by increasing and lowering the pressure of the refrigerant changing the saturation temperature.

The absorption cycle, uses a pump to increase the pressure of the system and not a compressor, which reduces its electricity consumption. However, it requires an additional heat flux in the generator, which makes the system performance lower than the compression system [3]. This additional heat flux may come from free or residual energy sources such as solar energy or waste heat, which could be economically advantageous [5, 14]. The most common pair of working fluids are: lithium bromide-water (refrigerant) or water- ammonia (refrigerant). This cycle uses only a pump to increase the pressure of the system and for this reason the electric energy used is lower than in the compression cycle. Also, in the absorption system it is necessary to provide a heat flux in the generator, which may come from solar energy or waste energy. The performance of the system is low, but it needs heat as the main input in the system instead of electricity [3].

In air-conditioning of spaces, the most used absorbent-refrigerant pair is lithium-water bromide. The absorption systems (single stage) require an external heat source at a temperature between 80 °C and 120 °C, and have a lower performance. The possibility of using free or residual energy sources is the advantage of absorption versus compression [5, 14].

Finally, another advantage that offers this type of devices compared to compression-type chillers is faint sound level and structure-borne sound propagation (low noise level, no vibrations).

2.2. Description of current absorption chiller products

2.2.1. Water/lithium bromide chillers

Water/lithium bromide is an absorption working fluid that has been used widely since the 1950s when the technology was pioneered by several manufacturers in the United States. This working fluid utilizes water as the refrigerant, and therefore, it is limited to refrigeration temperatures above 0°C. Absorption machines based on water/lithium bromide are typically configured as water chillers for air-conditioning systems in large buildings. They are available in sizes ranging from 35 to 5,000 kW.

The coefficient of performance (COP) of these machines, defined as the refrigeration capacity divided by the driving heat input, typically varies over the range $0.7<COP<1.2$ as a function of the cycle configuration. The most used technology is vapor compression chillers, and the choice between the compression or absorption chillers depends strongly of economic factors [5].

2.2.2. Ammonia/water chillers

Ammonia/water is an absorption pair of fluids that have been used since the late 1800s, at which time it was used for ice production before the introduction of the vapor compression technology. The ammonia is the refrigerant, and water is the transport medium. Thus, the role of water is distinctly different between ammonia/water and the water/lithium bromide. One advantage of ammonia as refrigerant is that it permits to work with lower temperatures because the freezing point of ammonia is -77.7°C. However, the toxicity of ammonia is a factor that has limited its use to well-ventilated areas.

The ability to provide direct gas-fired and air-cooled air conditioning is the primary selling point of water-ammonia chillers technology. Machines are available in capacity ranging from 10-90 kW with COP typically around 0.5-1.5. These units have a niche market because there are few competing gas-fired technologies suitable for many applications. Custom ammonia/water applications in industry for waste heat or renewable energy utilization are an interesting application of this technology [5, 12].

2.3. Single stage system

Figure 2 shows a single stage system (two working pressures and one generation & absorption stages) of absorption chiller that works with ammonia as refrigerant and water as solvent. The refrigerant ammonia allows to get lower evaporator temperatures than other substances and it is available for a wide variety of applications. Therefore it is one of the classic refrigerants [5, 11].

The components of the absorption refrigeration system are:

- **Evaporator:** It removes heat from the cooling media from the chiller (solution of water and glycol). The heat flux removed ($\dot{Q}_{evap}$) evaporates the refrigerant (Ammonia-NH_3) at a low temperature and a low-pressure environment.
- **Dephlegmator:** It is a heat exchanger where a partial condensation of the vapor (rich in ammonia) that is coming from the generator occurs (1). The heat rejection in this process allows to obtain pure ammonia vapor that goes towards the condenser (2V), and the condensate liquid (2L) (rich in water) returns to the rectifier per gravity.

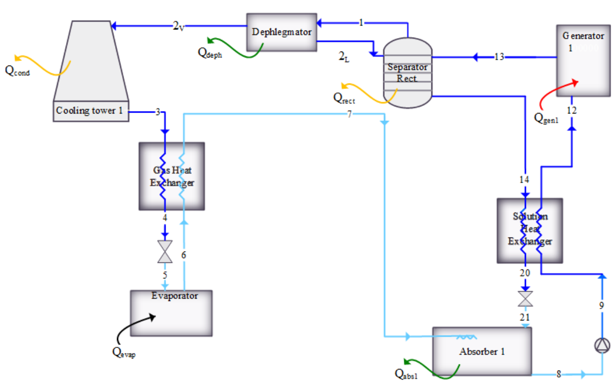

Figure 2. Single stage absorption system.

- **Condenser:** In the input of this component the ammonia vapor (2V) is at high pressure and it until condensate state when it passes through the heat exchanger (3).
- **Gas Heat Exchanger:** It is a heat recovered where the ammonia liquid that comes from the condenser (3) transfer heat to the ammonia vapor that goes to the absorber (6). It helps to subcooling the refrigerant and increase the system COP.
- **Expansion valves:** They reduce the pressure of liquid ammonia (4) and of the poor solution that goes to the absorber (20).
- **Absorber:** A dilute solution of ammonia water (21) absorbs the refrigerant vapor (7). The mixture must be simultaneously cooled down to dilute sufficient ammonia in the solution obtaining a rich solution at the output of this component (8). Therefore, to comply with this process, it is necessary to reject heat flux ($\dot{Q}_{abs}$).
- **Pump:** It transports the rich solution (8) increasing the pressure of the fluid which is an incompressible solution (liquid).
- **Solution Heat Exchanger:** It is a heat recovered where the poor solution (14) transfers heat to the rich solution (9) to be preheated.
- **Generator:** In this component, the rich solution (12) is heated, allowing to obtain vapor with a high concentration of ammonia (13V) and the solution is dilute respect to ammonia.
- **Rectifier:** It is a fractionating column required to condensate the partially vaporized water that is leaving from the generator and produce only very high purity ammonia vapor (1) that goes to the dephlegmator.

The COP of the single stage system is the relation of the heat flux removed in the evaporator respect to the input heat flux to the generator and the power consumed by the pump (Eq. (1)).

$$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{gen1} + \dot{W}_{pump1}} \tag{1}$$

Where:

COP - coefficient of performance
$\dot{Q}_{evap}$ - evaporator heat flux (kW)
$\dot{Q}_{gen1}$ - generator 1 heat flux (kW)
$\dot{W}_{pump1}$ mechanical power (kW)

2.4. Two-stage system (AGO equipment)

A scheme of the system studied in this project is shown in Figure 3. The information was obtained from the single line installation diagram of AGO project [11]. The principal components of the refrigeration system and the auxiliary systems are shown.

The refrigeration system studied is not a common two-stage absorption cycle described in the bibliography [1, 5]. It has two stages of generation and absorption process where the vapor rich in ammonia is leaving only from the Generator 1 (13) to the dephlegmator. The liquid solution leaves from the rectifier (14), it goes to the Generator 2 where heat is added to the solution. As a result, after the separator 2, more quantity of ammonia is absorbed by the state (17). Also, a poor solution (20) is obtained and goes to the absorber 2 and absorber 1 respectively.

The main objective of this two-stage equipment is to recover more energy from the hot water to supply it to the working fluids of the refrigeration system. In this project, a comparison of a thermodynamic model for a single stage absorption refrigeration system (Figure 2) and a double stage system (Figure 3) is performed.

The COP of the double stage system is determined by Eq. (2), and it is the relation of the heat flux removed in the evaporator respect to the heat consumed in generators and the mechanical power required by the pumps.

$$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{gen1} + \dot{Q}_{gen2} + \dot{W}_{pump1} + \dot{W}_{pump2}} \quad (2)$$

Where:

COP - coefficient of performance
$\dot{Q}_{evap}$ - evaporator heat flux (kW)
$\dot{Q}_{gen1}$ - generator 1 heat flux (kW)
$\dot{Q}_{gen2}$ - generator 2 heat flux (kW)
$\dot{W}_{pump1}$ - pump 1 mechanical power (kW)
$\dot{W}_{pump2}$ - pump 2 mechanical power (kW)

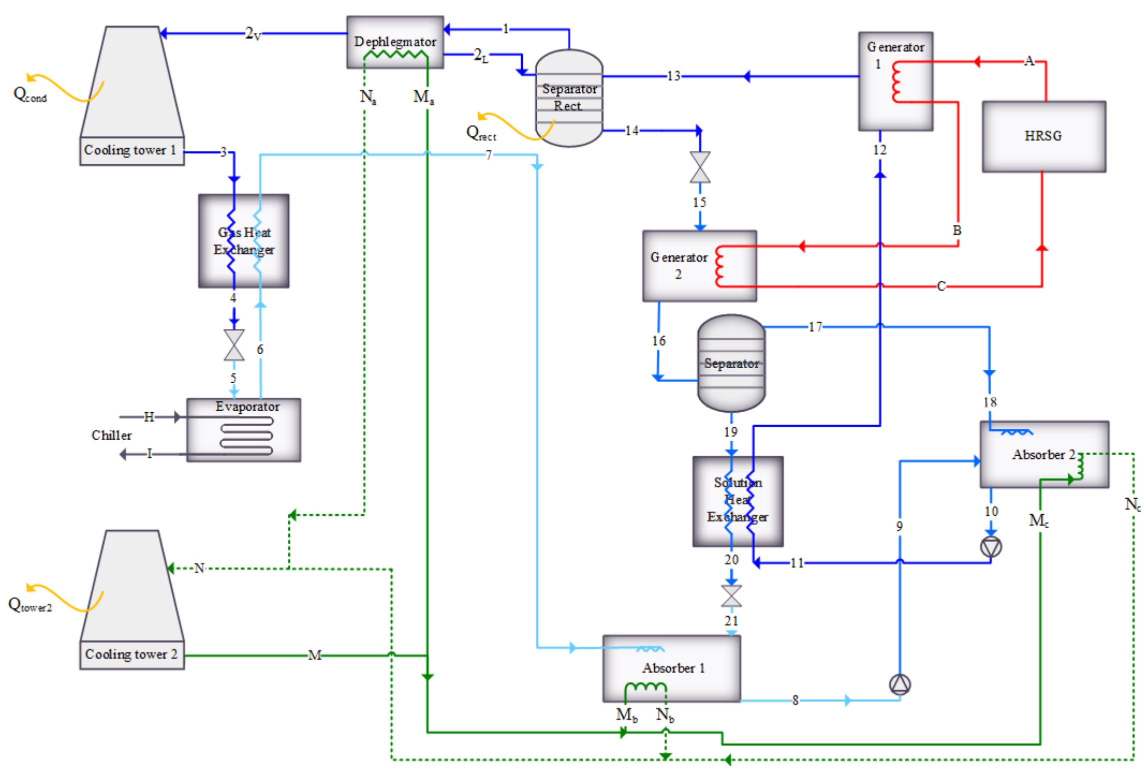

Figure 3. Double stage absorption equipment and auxiliary systems.

2.5. Auxiliary systems

In Figure 3, the auxiliary systems of the absorption system are included. They are interacting with different components of the absorption refrigeration system, and they are described below:

- **HRHWG:** It is a heat exchanger that uses the waste energy from the engine to produce hot water. In this case, the hot water (A) at 110 °C enters the Generator 1, deliveries heat to the ammonia-water solution and leaves (B) at 86 °C. This water continues to the Generator 2 where interchanges heat with the solution, and it goes out (C) at 80 °C. In normal operation conditions, the volumetric flow of water is 120 m_3/h.
- **The evaporator of the chiller:** The heat gained in the evaporator by the refrigerant (NH_3) is removed from a solution (glycol-water). In this system, the glycol enters (H) at -2 °C and it leaves (I) at -8 °C. The simulation developed in this project helps to determine the mass flow rate of glycol solution that can be cooled by this equipment.
- **Cooling tower 1:** In the condenser, the refrigerant needs to reject heat. This heat is rejected in a cooling tower cooled by water and air, with a capacity of 1250 kW.

- **Cooling tower 2:** The heat rejected from different components (absorber 1, absorber 2 and dephlegmator) evaporates the ammonia (M) and the vapor of ammonia (N) goes to the Cooling Tower 2 to condensate and continue the process. The capacity of this cooling tower is 3,750 kW.

The heat flux to be interchanged with the refrigeration system in the HRHWG and the evaporator chiller is calculated directly using the Eq. (3)

$$\dot{Q} = \dot{m} \times c_p \times \Delta T \tag{3}$$

Where:

$\dot{Q}$ - heat exchanger heat flux (kW)
$\dot{m}$ - water mass flow rate (kg/s)
C_p - water specific heat (kJ · kg/°C)
ΔT - temperature difference (°C)

The heat flux removed by the cooling towers are input data for the model of the refrigeration system developed in this project. The heat capacities of the cooling towers are used as a parameter for this model, and the thermodynamic and heat transfer process can be analyzed in detail in future work.

3. Model development

3.1. Thermodynamic analysis

The model was developed in EES (Engineering Equation Solver) [15] due to it has an external routine to know the ammonia water solution (liquid) and gas mixture properties. To address the thermodynamic study of the system presented in Figure 2 and Figure 3, we developed an appropriate methodology for each component as follows:

- **Generator:** The total mass, ammonia mass, and energy balances are expressed respectively in Eq. (4), Eq. (5), and Eq. (6).

$$\dot{m}_{12} = \dot{m}_{13V} + \dot{m}_{13L} \tag{4}$$

$$\dot{m}_{12} \times x_{12} = \dot{m}_{13V} \times x_{13v} + \dot{m}_{13L} \times h_{13V} \tag{5}$$

$$\dot{m}_{12} \times h_{12} + \dot{Q}_{gen1} = \dot{m}_{13V} \times h_{13V} + \dot{m}_{13L} \times h_{13V} \tag{6}$$

Where:

$\dot{m}_{12}$ - gas mixture mass flow rate (kg/s)
$\dot{m}_{13V}$ - gas mixture mass flow rate (kg/s)
$\dot{m}_{13L}$ - solution (liquid) mass flow rate (kg/s)
x_{12} - total ammonia mass fraction to generator 1
x_{13V} - ammonia steam mass fraction
h_{12} - gas mixture specific enthalpy of generator 1 (kJ/kg)
$\dot{Q}_{gen1}$ - generator 1 heat flux (kW)
h_{13V} - ammonia specific enthalpy of generator 1 (kJ/kg)

From this section, it is important to note that the amount of heat supplied to the ammonia-water solution flowing through the generator is equal to the heat delivered by the HRHWG defined by Eq. (7).

$$\dot{Q}gen1_{HRHWG} = \dot{m}_A \times c_p \times (T_A - T_B) \tag{7}$$

Where:

$\dot{Q}gen1_{HRHWG}$ - HRHWG heat flux (kW)
$\dot{m}_A$ - water (HRHWG) mass flow rate (kg/s)
C_p - water specific heat (kJ·kg/°C)
T_A - generator 1 inlet temperature (°C)
T_B - generator 1 outlet temperature (°C)

The mass and energy balances for generator 2 were performed under the same criteria.

- **Rectifier:** Mass and energy balances are developed, with the particularity that rectifier dissipates heat to the environment in response to its participation in the condensation of vaporized ammonia-water solution from the generator. This element performs the separation of the ammonia-water solution liquid and gaseous phases flowing through it. The rectifier is a fractionating column needed to condensate the partially vaporized water leaving from the generator and obtaining only ammonia-vapor of very high purity (1) which goes to the Dephlegmator.

- **Dephlegmator:** Heat is transferred to the auxiliary ammonia system to ensure that the refrigerant returns as vapor to the cooling tower 2, and the heat flux rejected is known with the Eq. (8).

$$\dot{Q}_{deph} = \dot{m}_{Ma} \times (h_{Na} - h_{Ma}) \tag{8}$$

Where:

$\dot{Q}_{deph}$ - dephlegmator heat flux (kW)
$\dot{m}_{Ma}$ - ammonia mass flow of dephlegmator rate (kg/s)
h_{Na} - outlet ammonia specific enthalpy (kJ/kg)
h_{Ma} - inlet ammonia specific enthalpy inlet (kJ/kg)

- **Condenser:** The heat removed from the ammonia vapor at the inlet is released into the atmosphere in the cooling tower.

- **Gas Heat Exchanger:** In the energy balance, to obtain the heat recovered by the ammonia vapor going to the absorber 1, the heat capacities of the incoming fluids to the heat exchanger are first determined using the ammonia's specific heat properties. Then, the minimum heat capacity is identified, as well as the largest temperature difference. In this way, the maximum heat flux that can be transferred will be equal the product of the minimum heat capacity and the largest temperature difference. Finally, the effectiveness of the heat exchanger is given in Eq. (9).

$$\varepsilon = \frac{\dot{Q}recov_{GHX}}{\dot{Q}_{max}} \tag{9}$$

Where:

ε - effectiveness of heat Exchanger
$\dot{Q}recov_{GHX}$ - gas heat exchanger flux (kW)
$\dot{Q}_{max}$ - maximum heat flux of GHX (kW)

The energy balance for solution heat exchanger was performed under the same criteria, where the only difference is that in SHX the heat is interchanged between the rich and poor solution.

- **Expansion valves:** For these components, mass flow and constant enthalpy are assumed.

- **Evaporator:** The heat lost by the glycol-water mixture (cooling media) is the same that the heat gained by the refrigerant (ammonia) circulated through the evaporator. The heat flux in the evaporator is calculated with the Eq. (10).

$$\dot{Q}_{evap} = \dot{m}_{glycol} \times c_{p-propylene} \times (T_H - T_I \tag{10}$$

Where:

$\dot{Q}_{evap}$ - chiller heat flux (kW)
$\dot{m}_{glycol}$ - glycol mass flow (kg/s)
T_H - inlet chiller temperature (°C)
T_I - outlet chiller temperature (°C)
$C_{p-propylene}$ - glycol-water specific heat (kJ · kg/°C)

- **Absorber:** The heat lost by the ammonia-water mixture in the absorber 1 is taken up by the auxiliary refrigerant system mentioned in the section corresponding to the dephlegmator and is the same process in the absorber 2. To know the heat flux lost in the absorbed the Eq. (11) is used.

$$\dot{Q}_{abs1} = \dot{m}_{Mb} \times (h_{Nb} - h_{Mb}) \tag{11}$$

Where:

$\dot{Q}_{abs1}$ - absorber 1 heat flux (kW)
$\dot{m}_{Mb}$ - ammonia mass flow rate of absorber 1 (kg/s)
h_{Nb} - ammonia specific enthalpy of absorber 1 outlet (kJ/kg)
h_{Mab} - ammonia specific enthalpy of absorber 1 inlet (kJ/kg)

The dephlegmator and both absorbers provide the heat used by the auxiliary refrigerant system.

- **Pump:** To calculate the energy required by the pump, we start with the formula that describes the power absorbed by a pump, where the specific volume and pressure difference are considered. This equals the work of a pump as a change of enthalpy as is shown in Eq. (12). The principle is the same for pump 2.

$$\dot{W}_{pump1} = \dot{m}_8 \times (h_9 - h_8) \tag{12}$$

Where:

$\dot{W}_{pump1}$ - pump 1 mechanical power (kW)
$\dot{m}_8$ - solution mass flow rate (kg/s)
h_9 - solution specific enthalpy pump 1 outlet (kJ/kg)
h_8 - solution specific enthalpy pump 1 inlet (kJ/kg)

The refrigeration system analyzed presents two absorption and generation stages that differ from the common cycle used by the industry (one stage). For this reason, two kinds of cycles were simulated in EES [15] with the same operational conditions defined as input data.

3.2. Model input data

The input data was defined based on operational design conditions. The high pressure of the equipment is 1,700 kPa, ammonia temperature in the evaporator is -12 °C. Consequently the system low pressure corresponds to saturation pressure (267.9 kPa).

The HRHWG deliveries 120 m^3/h of water to generators at 108 °C and the generation system decreases the water temperature down to 80 °C.

The chiller refrigerant (water and glycol) decreases its temperature by 6 °C in the evaporator.

The thermodynamic analysis has been developed focusing on the ammonia-water solution and gas mixture concentration. Common values of rich solution concentration is between 0.35-0.45 [14, 16]. The concentration of the solution at the input of the generator 2 corresponds to a medium concentration (x_{medium}), and it has been defined as a function of the cooling towers capacity (1,250 kW and 3,750 kW). The concentration of the medium solution is 0.36 as is shown in Figure 4.

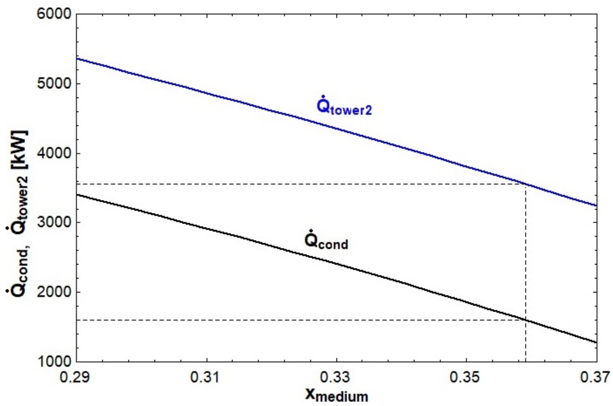

Figure 4. Condenser and Cooling tower 2 heat flux vs. concentration of medium solution.

3.3. Utilization factor of the CHP

The utilization factor of a cogeneration plant is the ratio of the power outputs (thermal energy to the chiller and electrical) respect to the total heat input as is show in Eq. (13).

$$\epsilon_u = \frac{\dot{Q}_{gen1} + \dot{Q}_{gen2} + \dot{W}_{net}}{\dot{Q}_{input}} \tag{13}$$

Where:

ϵ_u - CHP utilization factor
$\dot{Q}_{gen1}$ - generator 1 heat flux (kW)
$\dot{Q}_{gen2}$ - generator 2 heat flux (kW)
$\dot{Q}_{input}$ - CHP total heat input (kW)

The $\dot{Q}_{gen}$ is the thermal energy recovered to use as process heat in the chiller. The $\dot{W}_{net}$ is the net power obtained in gas and vapor turbines and $\dot{Q}_{input}$ is total heat input that corresponds to the heat energy consumed by the reciprocating engines.

3.4. Economics savings respect to a vapor compression refrigeration system.

The reduction of the electric power consumption with the ARS respect to a vapor compression refrigeration system allows obtaining economic savings. The savings have been calculated comparing with a vapor-compression refrigerator that has the same cooling capacity.

The cooling system works 24 hours per day, but according to its main characteristics, it can be assumed that the utilization factor is lower than one because the compressor starts working just when the temperature rises above a certain limit; therefore, as experimental field data, it can be set at 50%.

4. Results and analysis

4.1. Model results

The two-stage cycle lets to recover in a better way the energy from HRHWG water. For this reason, the solution temperature at the principal generator inlet increases and is possible to obtain a richer solution flow with the same available heat from the water.

Table 1. Comparison of one stage versus two stage cycles

	Units	Two stages	One stage
Heat recovered at GEN 1	[kW]	2,954	2,954
Heat recovered at GEN 2	[kW]	800	—
Solution temperature Generator 1 inlet	[°C]	75	65
Evaporator capacity	[kW]	1,6	1,3
COP	—	0.44	0.44

Adding a second absorption and generation stage to the simple absorption cycle increase 23% of refrigeration capacity as is shown in Table 1. The COP is similar for both systems (one and two-stage) but considering that the heat input of the two-stage refrigeration system was recovered from waste energy, it is more advantageous to include one stage more of generation and absorption to increase the cooling capacity.

The concentration of the solution at generator 2's input is the most influential variable in the model. When it presents lower values than 0.36 the COP system increase because of ammonia flow from the generator 2 increases, as is shown in Figure 5, and consequently the evaporator capacity enhances keeping constant the heat required at the ammonia generators.

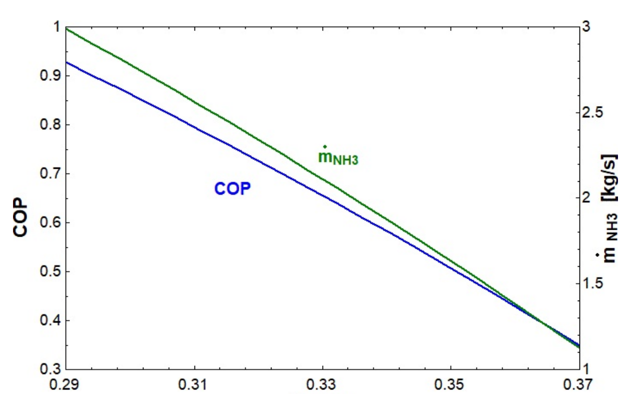

Figure 5. COP and mass flow rate of refrigerant respect to concentration of medium solution.

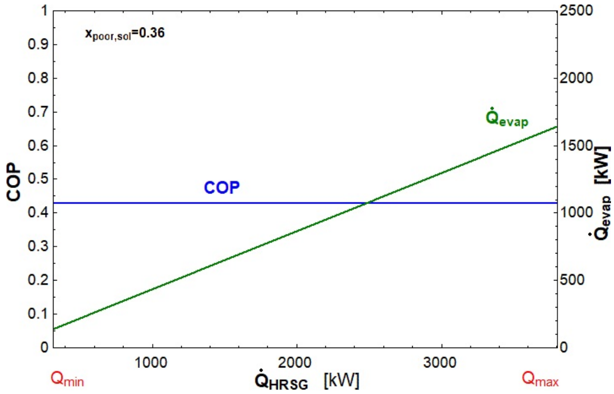

Figure 6. COP and evaporator heat flux respect to the heat recovery in the CHP.

The heat availability is a fundamental issue, and it depends of the engine operating conditions. If the engine is working at maximum load (maximum HRSG and HRHWG capacity), the generator 2 produces a greater ammonia flow and the refrigeration system reaches the maximum capacity (1,600 kW) and it presents 0.43 COP. The COP keeps constant despite heat recovered decreases and the evaporator capacity is reduced as is shown in Figure 6.

The high pressure of the system is an essential operational condition, and it is defined by the geography where the equipment is installed due to the altitude. If the absorption system operates at sea level the evaporator capacity increase and the COP enhance because the heat required by the ammonia generators remains constant as is shown in Figure 7. If the high pressure is higher, the cooling tower capacity (required to dissipate heat from the separators, absorbers, and dephlegmator) increases. For this reason, the cooling tower capacity must to be design based on in situ operational conditions.

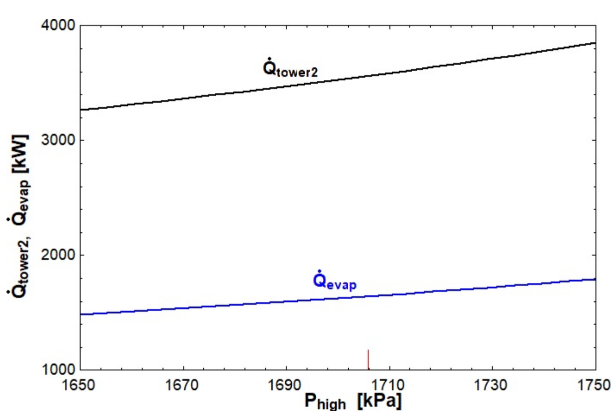

Figure 7. Heat flux in the cooling tower 2 and evaporator respect to high pressure system.

The heat exchangers effectiveness was considered as a constant value of 0.8 [17]. The solution heat exchanger lets to recover heat inside the cycle and increase the solution ammonia-water temperature before it enters to the principal generator. The effectiveness of SHX is directly proportional to the evaporator heat flux and the COP of the system, as is shown in Figure 8. Higher effectiveness SHX represents high investment cost, and it must be evaluated considering that the COP and the cooling capacity increase too.

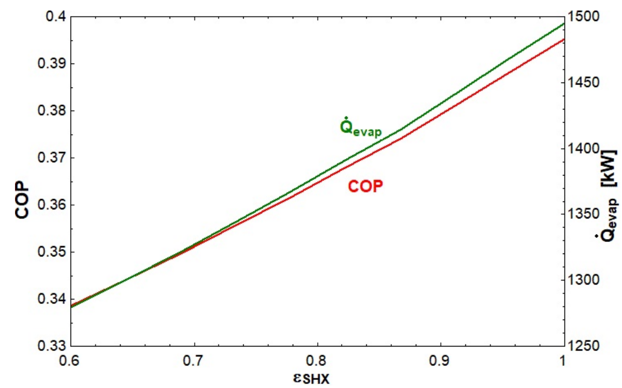

Figure 8. COP and heat flux in the evaporator respect to SHX effectiveness.

4.2. Utilization factor of CHP

The $\dot{Q}_{gen}$ is the heat recovered to use waste energy in the chiller and it is 3.8 MW (t) (obtained by the model). The $\dot{W}_{net}$ is 28.1 MW (e) and corresponds to the net power of the reciprocating engines (8.7 MW (e) per engine) and the vapor turbine (2 MW(e)). The $\dot{Q}_{input}$ is calculated in Eq. (14) with the data sheet information of the Roll Royce engines that reports 7,572 kJ/kWh of specific energy consumption. The utilization factor for the process is calculated with the Eq. (13) and the result is shown in Eq. (15). The input heat flux of CHP is known with kW(f) where f means fuel energy.

$$\dot{Q}_{input} = 3 \times 8,700kw(e)\left(\frac{7,572kJ(f)}{1kWh(e)}\right) \times \frac{1h}{3600s}$$
$$\dot{Q}_{input} = 54,897kW(f) \tag{14}$$

$$\epsilon_u = \frac{3,8MW + (2 + 8,7 \times 3)MW}{54,897MW} = 0,5811 \tag{15}$$

The utilization factor of the cogeneration process is 58.11%, and it means that the process is using a high percentage of the fuel energy that is consumed by the reciprocating engines.

4.3. Estimation of economics savings

The cooling capacity of the two-stage absorption refrigeration (AGO) system obtained with our model is 1,623 kW (t). The electricity that would consume a common vapor compression chiller to obtain the same cooling capacity is calculated in this section to know the economics savings.

The market offers many kinds of machines performing refrigeration with vapor compression cycle. The average performance (COP) is 4.5 [18].

Therefore, it is possible to calculate the mechanical power required by the compressor ($\dot{W}_{comp}$) with Eq. (16).

$$\dot{W}_{comp} = \frac{\dot{Q}_{evap}}{COP} = \frac{1,623kW}{4,5} = 360,67kW(e) \quad (16)$$

The utilization factor for a compression chiller was defined as 50% in the section 3.4 and considering 24 hour per day and 365 days per year, the yearly electricity that would consume by the compression system is 1'579,734.60 kWh/year.

Considering an average energy unit cost (obtained from monthly electricity cost of the two to last year for an industrial rate for Agua Prieta / Mexico [19]) of 0.09 USD/kWh, the avoided electricity cost by the cogeneration system would be 142,000.00 USD/year approximately.

However, considering the possible savings and fixing a payback time of 5 years, the initial investment could be around 710,000.00 USD. It means that the investment is profitable. To obtain more accurate results, it is necessary to acquire specific data.

The efficient cogeneration has several fiscal incentives and preference energy dispatch in different countries (for example Mexico) due to the contribution of technology with the preservation of the environment.

5. Conclusions

It is convenient to use a two-stage absorption system since they allow to exploit better the heat discharged by the engines. Also, the COP of both systems (single and double stage) are similar.

The concentration of the solution at the exit of the generator 1 is the variable assumed to be most influent in the results of the cycle. For this reason, this variable has been chosen according to the capacity of the installed cooling towers. Moreover, it has been observed that if the concentration is close to 0.3 the system COP is close to 1 while it goes down for higher values of concentration.

The COP keeps constant independently from the amount of heat flux entering the system. If the generator is supplied with a higher amount of heat, the mass flow rate of ammonia going to the evaporator increases and consequently refrigeration capacity of the chiller goes up.

The cogeneration process using exhaust gases as heat source for an absorption refrigeration chiller can provide environmental conservation and economic revenues.

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