Computer-Aided Simulation of the Energetic and Exergetic Efficiency of a Two-Stage Cascade Cooling Cycle

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Abstract

Refrigeration systems are of great importance in industry and are widely used in food preservation or air conditioning. Studies have been carried out in recent years in search of parameters to optimise the cycle by reducing losses, both to increase the COP and to reduce energy consumption, so that process improvements have been made from which different cycles are derived, such as the gas cooling cycle with regeneration, absorption cooling, gas liquefaction or multi-stage compression cooling systems. This document presents a combined refrigeration system consisting of two vapour compression refrigeration cycles linked by a heat exchanger that not only reduces the work of the compressor but also increases the amount of heat absorbed by the refrigerated space as a result of the cascade stages and improves the COP of a refrigeration system. The study was carried out under the assumption of a stationary flow, and without significant changes in the kinetic and potential energy, i.e. assuming an ideal process, carrying out several case studies with the help of UNISIM, evaluating the effect of varying the low temperature on the exergetic efficiency, the destroyed exergy of the condenser, the net input work and the COP, as well as the influence of the input pressure in the upper cycle compressor on the net input work and the COP.

Keywords: Refrigeration, exergetic efficiency, heat interchanger, compressor, COP.

INTRODUCTION

Considering the importance of refrigeration for food preservation and air conditioning in industry, significant efforts have been made to improve energy efficiency and COP and to reduce the energy consumption of this process [1],[2]. Efficiency can be increased by making certain improvements that result in minimizing losses, in addition to providing important benefits ranging from an increase in COP to an increase in operating temperature intervals, which would allow moderate low temperatures to be worked on [3].

In many researches computer-aided simulation has been considered as a flexible, efficient and user-friendly tool that allows to simulate and optimize such thermodynamic processes as ProSimPlus® [4][5], Aspen Plus™[6],[7],[8]. Similarly, the Aspen HYSYSYS® software was used to evaluate under a thermodynamic approach the potential application of ionic liquids to absorption cooling cycles using the [9]. Applying this methodology, 7200 systems were evaluated, consisting of 900 independent laboratories and 8 refrigerants, representative of the refrigerants available on the market, which would otherwise be unfeasible due to the lack of experimental data [10],[11],[12]. One of the main application areas of thermodynamics is refrigeration, which is the transfer of heat from a lower temperature region to a higher temperature. The devices that produce refrigeration are called refrigerators, and the cycles in which they operate are called refrigeration cycles. The most frequently used refrigeration cycle is by vapor compression, where the refrigerant is evaporated and condensed alternately, and then compressed in the vapor phase. There is also the gas cooling cycle where the refrigerant remains in the gas phase all the time, the absorption cooling cycle where the refrigerant is dissolved in a liquid before being compressed or the cascade cooling cycle which uses more than one cooling cycle [13]. For special applications, or to improve performance, variations of the basic vapor compression refrigeration cycle are used, such as the cascade cycle, which is the configuration in which refrigeration at relatively low temperatures is produced by a series of vapor compression systems, so that a different refrigerant is normally used for each one[14].

Therefore, the main contribution of this work is to propose an analysis of the performance of various case studies in a two-stage cascade refrigeration cycle using computer-aided simulation using UNISIM, allowing the evaluation of the effect of varying low temperature on the exergic efficiency, on the destroyed exergy of the condenser, on the COP and on the network of the system, as well as the evaluation of the effect of varying the input pressure of the high pressure compressor on the network and the system's COP.

METHODOLOGY

The UNISIM software was used for the analysis of the studied cycle, which is an interactive process engineering and simulation program that allows the simulation of chemical plants and oil refineries, and includes tools for estimating the properties and equilibrium of the liquid-vapor phase, heat and material balances, as well as the simulation of many types of equipment in chemical engineering.

Fundamental equations.

For the modeling of the cycle it was assumed that it was with stationary flow, also it was considered for this case study that the changes in the kinetic and potential energies are insignificant so our energy balance was reduced to the equation (1).

$$\sum \dot{Q}_{in} + \sum \dot{W}_{in} + \dot{m}_{in}(h_{in}) = \sum \dot{Q}_{out} + \sum \dot{W}_{out} + \dot{m}_{out}(h_{out}), \quad (1)$$

where Q and W represent the energy transfer in the form of heat and work respectively, and m and h are the mass flow and enthalpy of the current. The stages of the cascade cooling cycle are connected by means of a heat exchanger in the middle which serves as evaporator for the upper cycle and as condenser for the lower cycle. Assuming that the heat exchanger is well insulated and that the kinetic and potential energies are negligible, the heat transfer of the fluid in the lower cycle must be equal to the heat transfer of the fluid in the upper cycle, so that the ratio of the mass flows in each cycle is estimated as shown in equation (2) and (3) below

$$\sum_{in} \dot{m}h = \sum_{out} \dot{m}h, \quad (2)$$

$$\frac{\dot{m}_A}{\dot{m}_B} = \frac{h_2 - h_3}{h_5 - h_8}. \quad (3)$$

In addition, the coefficient of performance (COP) for cascade cooling is given by equation (4) as

$$COP_{R,cascade} = \frac{\dot{Q}_l}{\dot{W}_{net,in}} = \frac{\dot{m}_B(h_1 - h_4)}{\dot{m}_A(h_6 - h_5) + \dot{m}_B(h_2 - h_1)}, \quad (4)$$

where the power input of the cascade cycle, which is the sum of the power inputs of all the compressors, gives equation (5) as

$$\dot{W}_{net,in} = \dot{m}_A(h_6 - h_5) + \dot{m}_B(h_2 - h_1), \quad (5)$$

and the cooling load is by equation (6)

$$\dot{Q}_l = \dot{m}_B(h_1 - h_4). \quad (6)$$

For compressor efficiency, proceed as shown in equation (7)

$$n_C = \frac{(h_{2s} - h_1)}{(h_2 - h_1)}. \quad (7)$$

A $T_0 = 300K$ was assumed for the processes in each component, the destroyed exergy for the condenser wass determined by equation (8)

$$x_{destroyed,condenser} = \dot{m}_H[(h_6 - h_7) - T_0(s_6 - s_7)] - \left(1 - \frac{T_0}{T_{cond}}\right)\dot{Q}_{cond}, \quad (8)$$

for the compressor you have equation (9)

$$x_{destroyed,compressor} = \dot{W}_{comp} + \dot{m}_L[(h_1 - h_2) - T_0(s_1 - s_2)] + \dot{m}_L[(h_5 - h_6) - T_0(s_5 - s_6)], \quad (9)$$

for the heat exchanger you have equation (10)

$$x_{dest,heat\ exchanger} = \dot{m}_L[(h_2 - h_3) - T_0(s_2 - s_3)] + \dot{m}_L[(h_8 - h_5) - T_0(s_8 - s_5)], \quad (10)$$

for the compressor you have equation (11)

$$x_{dest,evaporator} = \left(1 - \frac{T_0}{T_{evap}}\right)\dot{Q}_{evap} + \dot{m}_L[(h_4 - h_1) - T_0(s_4 - s_1)], \quad (11)$$

for the compressor you have equation (12)

$$x_{dest,valve} = \dot{m}_L[(h_3 - h_4) - T_0(s_3 - s_4)] + \dot{m}_L[(h_7 - h_8) - T_0(s_7 - s_8)], \quad (12)$$

hence, the irreversibility of the cycle is described in equation (13)

$$x_{destr,cycle} = \sum x_{destr,component}. \quad (13)$$

RESULTS AND DISCUSSION

In the two stage cascade refrigeration cycle, which consists of two compression refrigeration cycles connected by a heat exchanger, which serves as the evaporator for the upper cycle and as the condenser for the lower cycle, plus two expansion valves, two compressors, a condenser and a evaporator as shown in Figure 1.
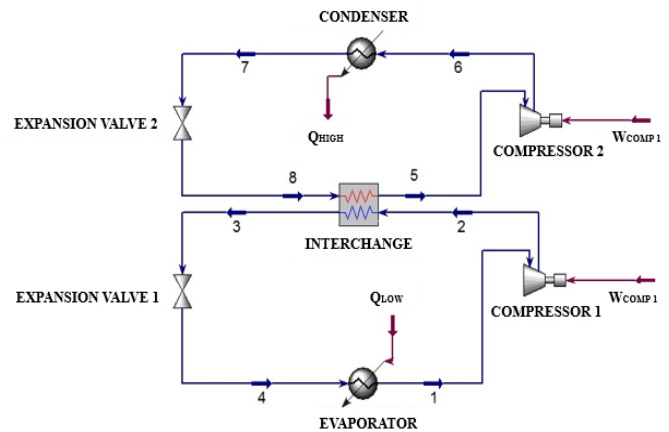

Figure 1. Graphical representation of the two-stage Cascade Cooling Cycle in the Unisim software.

The initial parameters for the cycle analysis were T_1=262.98 °C, P_1=200kPa, P_2=500kPa, T_5=281.93°C, P_5=400kPa, P_7=1200kPa, and T_0=300K, while the mass flow through the upper cycle is m_H=0.212kg/s and the mass flow through the lower cycle is m_L=0.15kg/s. According to these data, the properties of the components of the cycle were obtained and are shown in Table 1.

Table 1. Properties of the Two-Stage Cascade Cooling Cycle

State	Presure (kPa)	Temperature [K]	Enthalpy h [kJ/kg]	Entropy s [kJ/kg-K]	Phase
1	200	262,98	244,4600	0,9377	Saturated Steam
2	500	297,21	268,4600	0,9537	Supersaturated Steam
3	500	288,71	77,4600	0,2937	Saturated Liquid
4	200	262,98	77,4600	0,3027	Mix (x=0,1685)
5	400	281,93	255,5500	0,9269	Saturated Steam
6	1200	327,28	284,5500	0,9547	Supersaturated Steam
7	1200	319,07	120,5500	0,4417	Saturated Liquid
8	400	281,93	120,5500	0,4587	Mix (x=0,2832)

According to the data supplied by the software, the values described in Table 2 are given, which shows the work of the compressors and the evaporator and condenser heatsinks.

Table 2. Compressor work and high and low heat.

Works and heat [kW]	
W Compressor1	3,5735
W Compressor2	6,1068
Q Evaporator, low	25,0370
Q Condenser, high	34,7172

In addition, Table 3 shows the exergy destroyed in each component and the total exergy destroyed, which highlights that 40% of the total exergy destroyed is in the compressor since this is the equipment where energy is supplied to the system and significant irreversibilities occur by heat transfer.

Table 3. Exergy destroyed by component.

Exergy destroyed by component [kW]	
Compressor	2,4204
Condenser	2,1412
Valve	1,4862
Interchange	0,1075
Evaporator	3,3118E-05
X destroyed, Sum	6,1553

In order to have a graphical representation of the process, the T-s diagram of the two-stage Cascade Refrigeration Cycle has been developed. It is shown in Figure 2, where it is emphasized that the operating condition of the working fluid is superheated steam at the compressor outlet, in addition to supercooling and overheating in this process, which is a limit condition for the operation of the system.
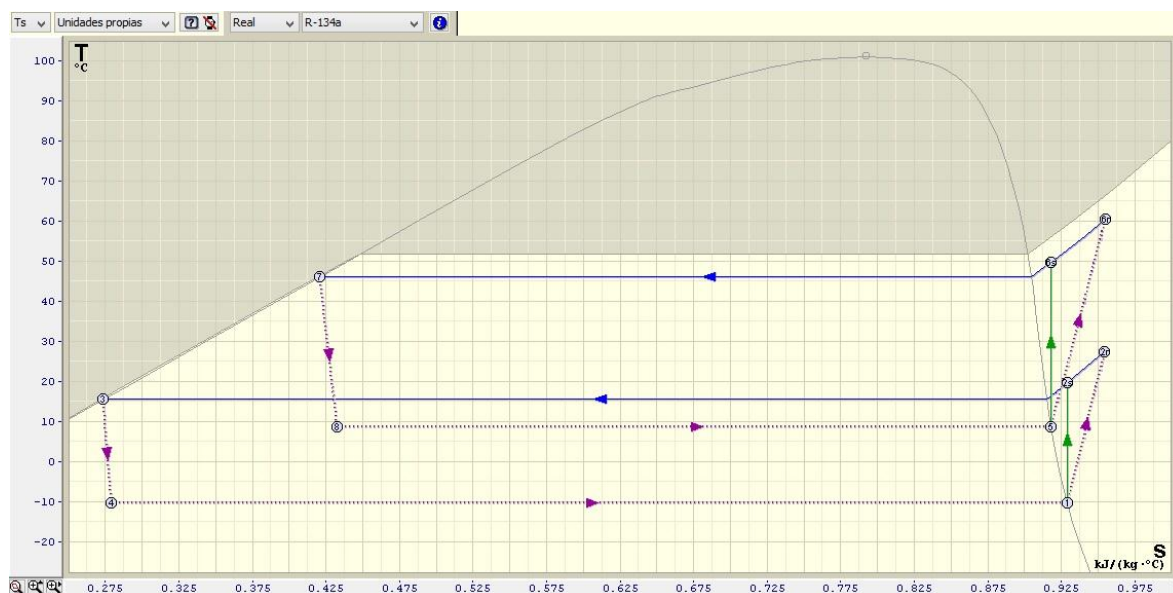

Figure 2. T-s diagram of the two-stage Cascade Cooling Cycle.

Influence of the low temperature on the exergetic efficiency and the destroyed exergy of the condenser.

The behavior of the exergetic efficiency was analyzed when the low temperature varies in the range of 265 K to 295 K, and varying the efficiency of the compressors in the same proportion, it was also studied with the same initial parameters as it influences the destroyed exergy of the condenser, of which Figure 3 was obtained as a result.
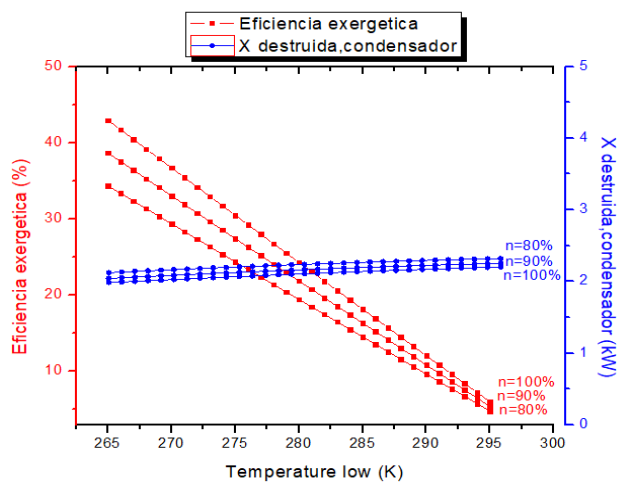
Figure 3. Exergetic efficiency and destroyed exergy of the condenser vs. low temperature (TL).

Figure 4. Net input work and COP vs. low temperature (TL).

As shown in Figure 3, obtained by graphing the results provided by the software, it is observed that they tend to behave linearly which can be explained by the influence of the low temperature, which generates a proportional change in the destroyed exergy of the capacitor and the exergetic efficiency of the cycle. On the other hand, a 10% increase in the low temperature causes a decrease in the exergetic efficiency of the system when working on any of the 3 component efficiencies (80%, 90% and 100%), also when the efficiency of the compressors decreases 100 to 80% in the last test value, it is observed that from that point on, the exergetic efficiency tends to be approximately the same. In turn, this 10% increase in low temperature generates a 5% increase in the destroyed exergy of the condenser. In making a comparison it can be concluded that when the low temperature is increased the exergy destroyed in the condenser increases slightly, but the exergetic efficiency decreases and reaches a certain point at which it tends to be the same for any compressor efficiency, therefore, the low temperature should be controlled so that it does not negatively affect and waste energy in the process.

Influence of the low temperature on the net input work and the COP

The low temperature range was varied from 265 K to 295 K and the compressor efficiency (at the same proportion) to evaluate the performance of the net input work of the system and the COP, resulting in Figure 4.
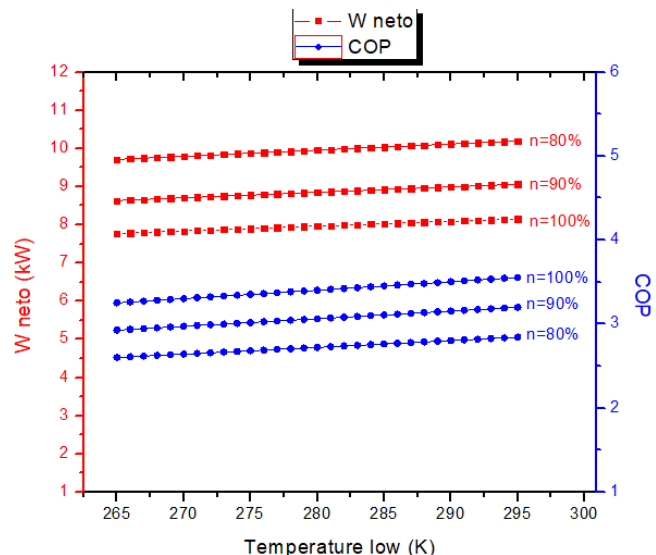
The results show that the results tend to be proportionally linear. A 10% increase in temperature drop causes the net input work to be higher (almost 5% depending on compressor efficiency) and the COP to increase by almost 9%. However, when compressor efficiency drops from 100% to 80% the COP decreases by 20% and the net input work increases by the same percentage for the same low temperature, this is because the COP is a ratio of the heat removal rate in the refrigerated space to the net input work. From which it is concluded that it is not advisable to increase the low temperature because it has a negative effect on the performance of the system, it is better to decrease it as much as possible.

Influence of pressure at compressor inlet 2 (P5) on net input work and COP

The behaviour of the net input work and of the COP was analysed when the input pressure to compressor 2 (P5) was varied in the range of 100 kPa to 1000 kPa and the efficiency of the compressors was varied (in the same proportion), resulting in Figure 5. The result shows an exponential decrease in net inlet work compared to the increase inlet pressure to compressor 2 and the decrease in compressor efficiency, this could be due to the fact that the increase in pressure at the compressor inlet helps the compressor to compress less pressure to reach the desired outlet pressure and therefore requires less inlet work, although the decrease in compressor efficiency from 100% to 80% causes an increase of 20% in the work. At the same time this increase in the inlet pressure of compressor 2 influences the COP which increases exponentially, in the varied pressure range the COP increases by 74%, although in the decrease of compressor efficiency from 100% to 80% it decreases by 20%.
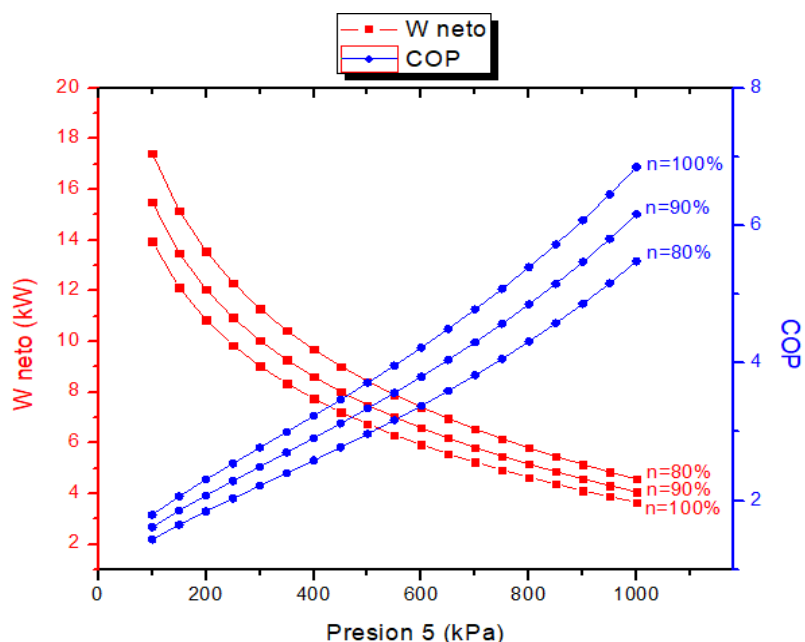
Figure 5. Net Inlet Work and COP vs. Compressor Inlet Pressure 2 (P5)

CONCLUSION

Some industrial applications require moderately low temperatures, and the temperature range involved is too large for a simple vapor compression refrigeration cycle. A large temperature range also means a high pressure level in the cycle and poor performance in the reciprocating compressor. One way to avoid such situations is to have two or more cooling cycles operating in series, such processes being called cascade cooling cycles.

By developing an analysis of the behavior and influence that low temperature has on the exergistic efficiency, the destroyed exergy of the condenser, the net input work and the COP of a cascade refrigeration cycle in which refrigerant 134a is used, it can be concluded that high temperatures had negative results on the exergistic efficiency and cycle performance, because they cause more input work to be required for compressor operation and an increase in the destroyed exergy in the condenser. In addition, the influence of the increase in pressure at the compressor inlet was reflected in the decrease in its required work, which in turn increased the COP because they are inversely proportional. The results obtained through the UNISIM software allow the importance of low temperature and compressor inlet pressure to be reflected in the total performance of the cascade refrigeration cycle.

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