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Thermoeconomic Simulation of Cascaded and Integrated Vapor Compression-Absorption Refrigeration Systems

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Abstract. *The present work is composed by a comparative thermoeconomic analysis between two refrigeration systems: Vapor Compression Cascade Refrigeration System (VCCRS) and Integrated Refrigeration System by Absorption and Vapor Compression (VCACRS). The thermoeconomic analysis compares the systems under energy, exergy, economic and environmental aspects. The development of mathematical models for each of the systems is performed through the EES (Engineering Equation Solver) program. The optimized functions are exergy destruction and total cost rate (sum of cost rates of investment, operation, maintenance and enviromental) by minimizing these functions. The optimization method used is the weighted sum of the objectives, this can be achieved by assigning different weights for each goal, then a new function that represents the linear relationship between all the objectives is found. In present case the two objective functions are exergy destruction and total cost rate. In multiobjective optimization, the process of choosing among optimized solutions involves the definition of an equilibrium point, also called the ideal point. In order to achieve a real solution of the minimum values of the described functions simultaneously one must determine which is the smallest distance from the ideal point to the curve that defines the optimized solutions. In the study the economical advantage of VCCRS in relation to VCACRS was demonstrated. VCACRS has a cost 10.26% lower than VCCRS while VCCRS has a better exergetic efficiency, with its destruction of exergy 38.46% lower than VCACRS.*

Keywords: *Refrigeration, Multiobjective optimization, Thermoeconomic analysis*

1. INTRODUCTION

The use of electrical energy has become essential in the life of modern man. From the most complex to the simplest tasks mankind depends on electric power in residential, services, commercial or industrial sectors. Therefore the search for energy savings and the development of sustainable technologies is a subject of great interest in the scientific community around the world. Electricity generation in Brazil in public service and self-generating plants was $619.7TWh$ in 2016, a slightly higher result of $4TWh$ than in 2015 according to MME (2017).

Brazilian energy matrix is comprised of 43.5 % of renewable energies, 13.5 % higher than world average. When it comes to the electrical matrix, the numbers are even better, with 81.7 % of the electricity consumed in the country coming from renewable sources. The distribution of electrical matrix consists of 68.1 % hydraulic generation, 8.2 % biomass, 9.1 % natural gas, 5.4 % wind energy, 6.6 % coal, oil and its derivatives and 2.6 % of nuclear energy according to MME (2017).

According to Reis (2016) data from the Brazilian Supermarket Association (Abras) reveal that in 2014 energy consumption in the segment was 8.6 GWh, which corresponds to 2.5 % of energy consumption in the entire country. The average consumption per store was $103MWh$, which resulted in an expense of about $R\$3.5$ billion just with the energy bill. In 2015, energy costs exceeded rental expenses and became the second largest expense in supermarkets only behind payroll.

The reduction in operating costs is an important tool for increasing the economical competitiveness in industry. Most supermarket energy costs comes from air conditioning and refrigeration, these systems must be regularly inspected, controlled and in some cases replaced with more efficient ones. Due to their high relevance in energy consumption researchers aim to understand the mechanism of refrigeration systems in search of analysis and improvement of projects. The traditional way of evaluating the performance of a refrigeration system is to carry out standardized experimental tests. Those tests are expensive and take time, which greatly increases the costs of their development. As an alternative to experimental tests the use of mathematical models to simulate the behavior of refrigeration systems are being held.

The use of Integrated Vapor Compression Absorption Cascaded Refrigeration System (VCACRS) can represent an alternative to Vapor Compression Cascaded Refrigeration System (VCCRS), as integrated systems require less electrical energy than the equivalent vapor compression cycle and the absorption cycle enables the refrigeration process using alternative sources of thermal energy. These alternative sources can be from renewable sources such as solar energy or thermal rejects from other processes, considering that the Brazilian energy matrix is not 100 % renewable, the reduction in the consumption of electricity also represents a reduction in gas emissions that accelerate the greenhouse effect.

Research results regarding the performance of refrigeration systems are widely studied according to Tab. 1.

Table 1. Bibliographic studies on refrigeration systems articles results

Author(s)	Year	Results
KAIROUANI E NEHDI	2006	The results showed that the COP of VCACRS are significantly higher than those of VCCRS, and can be increased by 37-54 % under the same operating conditions.
REZAYAN E BEHBAHANINIA	2011	The total annual cost of the optimized system was reduced by 9.34 % in relation to the base case.
DA SILVA et al.	2012	Reduction of electrical energy consumption of 13 to 24 % of the VCCRS in relation to the conventional system.
JAIN et al.	2013	The result was a reduction of 61 % in the consumption of electric energy and an increase of 155 % in the COP of VCACRS in relation to the VCCRS.
MA et al.	2014	2.3 % increase in the COP of the optimized system compared to the base case.
AMINYAVARI et al.	2014	The results showed that the optimum point corresponds to an exergetic efficiency of 45.89 % and a total cost rate of 0.01099 US$ $\cdot s^{-1}$
JAIN et al.	2015	The study showed a reduction of the annual cost of the system of 11.9 % and 22.4 % reduction of investment costs for the base case.
JAIN et al.	2015	After determining the optimized parameters for VCACRS, a comparative study was developed in relation to VCCRS and it was concluded that the energy consumption was reduced by 60% and the COP was increased by 153.6%.
BOYAGHCHI et al.	2016	In a study related to economic aspects of the use of refrigerants for VCACRS the most efficient fluid was R1234ze resulting in a total annual operating cost of US$ 7016 / year.
DIXIT et al.	2017	The study showed that the annual operating cost of VCACRS was 21.6 % lower in relation to the equivalent compression cycle, which was reduced by an additional 18.2 % after the system optimization was performed.
TURGUT E TURGUT	2018	In a study related to the use of refrigerant fluids for VCACRS the analysis, whose objective was the thermodynamic investigation, identified that the system operating with refrigerant R290 in the compression cycle showed the highest efficiency of second law of 0.41, whereas the system operating with R134a was the system that had the worst efficiency of 0.38.

Source: The Author

This work aims to develop a thermal simulation for two refrigeration systems the Vapor Compression Cascaded Refrigeration System (VCCRS) and the Integrated Vapor Compression Absorption Cascaded Refrigeration System (VCACRS).

The specific objectives are:

- Develop a computational modeling of a VCACRS.
- Develop a computational modeling of a VCCRS.
- Investigate the design parameters that influence the systems models.
- Find optimal values of project parameters by minimizing exergy destruction functions and the total annual cost of the system (sum of investment and maintenance, operation and environmental costs) through the optimization of the VCACRS and VCCRS.

- Compare the performance of VCACRS using R744 and the pair $H_2O - LiBr$ with the conventional VCCRS using R744 and R717.

The optimization studies were carried out by varying the operating parameters of the refrigeration systems with the objective of minimizing the annual cost of the system and maximizing their energy efficiency. Another observed trend is the comparison of the constructive configurations of the systems and the refrigerants used in them. Regarding the environmental issue the studies are based on models whose energy sources are renewable sources.

2. THEORETICAL FORMULATION

In this topic mathematical models are presented for the simulation of VCACRS and VCCRS to refrigerate a propylene glycol solution from $-20\ C$ to $-30\ C$ for industrial refrigeration applications.

2.1 Vapor Compression Cascaded Refrigeration System (VCCRS)

Many times industrial applications of vapor compression refrigeration systems require temperatures below $0\ C$ generating a large gap between the temperatures of the zone to be cooled and the zone to which heat is rejected. After leaving the evaporator the refrigerant is compressed to a temperature higher than that of the refrigerated zone, therefore for large temperature ranges, large pressure ranges will be necessary requiring larger compressors which have a higher acquisition cost and consume more energy. In addition single-stage systems have smaller COPs compared to a cascaded system in accordance with Rezayan and Behbahaninia (2011).

One solution to this problem is to perform the refrigeration process into stages, assembling two or more cycles to operate in series. The assembling of the cycles is constructed by connecting the evaporator of one cycle with the condenser of the other one as shown in Fig. 1. The heat discharged in the process 2-3 is used to evaporate the refrigerant fluid from the process 8-5 in a heat exchanger.

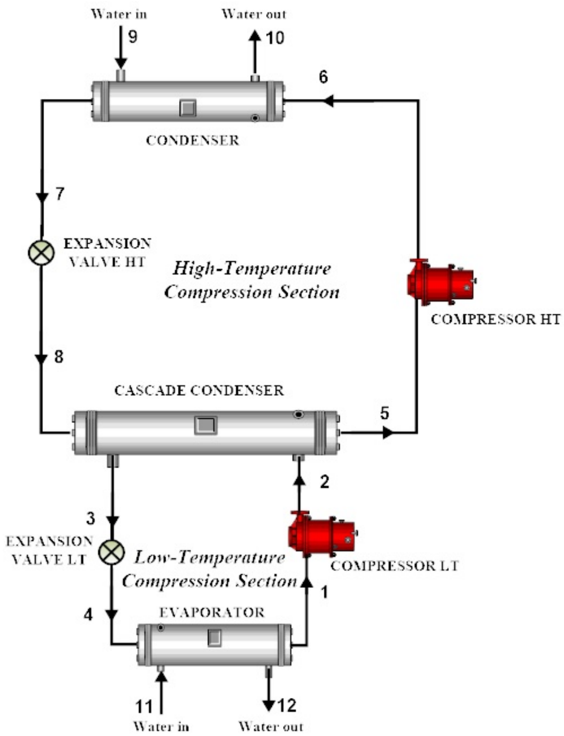

Figure 1. Vapor Compression Cascaded Refrigeration System

Source: The Author

The refrigerant used in the low temperature section of the VCCRS is the R744 as it is a non-toxic, non-flammable, natural refrigerant, those are the mains advantages of using this refrigerant. It has also a positive vapor pressure at temperatures below $-35\ C$, which prevents leaks in pipes according to Rezayan and Behbahaninia (2011). In the high temperature section R717 was selected because it is a CFC-free natural fluid, making it more advantageous in environmental terms according to Rezayan and Behbahaninia (2011).

2.2 Vapor Compression Absorption Cascaded Refrigeration System (VCACRS)

An Integrated Vapor Compression Absorption Cascaded Refrigeration System is the assembling between a vapor compression cycle with an absorption cycle in a cascaded disposal as shown in Fig. 2 according to Jain *et al.* (2016).

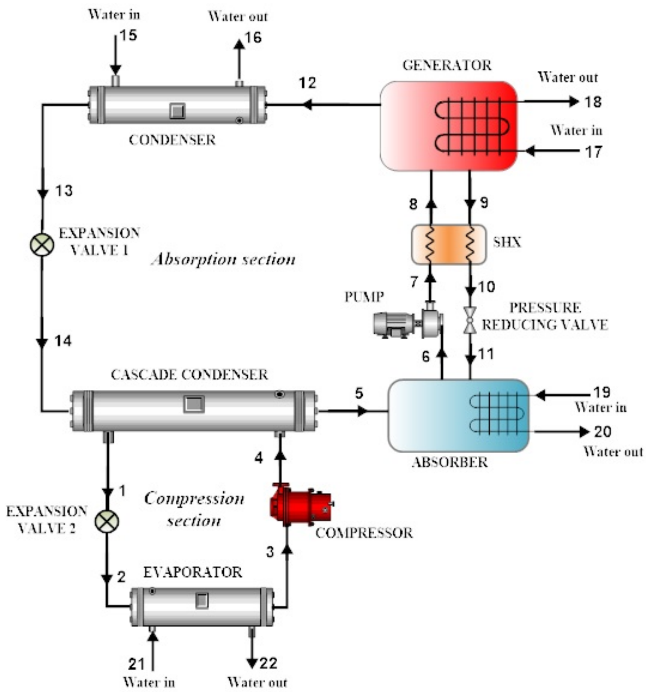

Figure 2. Vapor Compression Absorption Cascaded Refrigeration System

Source: The Author

The absorption system is similar to the vapor compression system but the compressor is replaced by a more complex mechanism composed of one absorber, one pump, one generator, one pressure reducing valve and one solution heat exchanger. After going through this mechanism the refrigerant has a high pressure and follows the same compression cycle, so it is cooled and condensed in the condenser, strangled and passes through the evaporator to receive heat from the fluid to be cooled.

After the saturated vapor leaves the evaporator it enters the absorber where it forms a liquid mixture with the transport medium. The solubility is proportionally inverse to the temperature, heat is removed from the mixture so that the maximum amount of refrigerant can be absorbed. The solution now rich in refrigerant is pumped to the generator where heat is gained by the mixture in order to vaporize the refrigerant and thus separate it from the mixture. The vapor with a high amount of refrigerant returns to the condenser and restarts the cycle. The rest of the mixture which is hot and weak in refrigerant goes through a solution heat exchanger where it transfers part of the heat to the solution that is going to the generator in order to improve the thermal efficiency. It is then strangled until the absorber pressure and the process continues.

2.3 Energetic Analysis

The following premises were adopted to elaborate the models

- Variations of kinetic, potential, nuclear, magnetic and chemical energy were neglected;
- Permanent regime;
- Pressure drop was neglected;
- Saturated liquid at the condenser outlet;
- Saturated vapor at the evaporator outlet;
- Saturated vapor at the output of the cascade heat exchanger in the absorption part

- Isenthalpic expansion valves;
- Heat loss at the compressors was neglected;
- The strong and weak solutions of $LiBr - H_2O$ leaving the absorber and generator are saturated and in balance at given pressures and temperatures;

The development of the model was elaborated according to the equations of the mass and energy balances represented by Eq. 1, Eq. 2 and Eq. 3 according to Cengel and Boles (2007).

$$\Sigma \dot{m}_{in} - \Sigma \dot{m}_{out} = 0 \tag{1}$$

$$\dot{Q} - \dot{W} + \Sigma \dot{m}_{in} h_{in} - \Sigma \dot{m}_{out} h_{out} = 0 \tag{2}$$

$$\Sigma \dot{m}_{out} s_{out} - \Sigma \dot{m}_{in} s_{in} - \frac{\dot{Q}}{T} = \dot{S}_{GEN} \tag{3}$$

The equations for each component of VCACRS are presented in Tab. 2 and are analogous for VCCRS.

Table 2. Energy and Mass Balance for VCACRS

Component	Second law equations
Evaporator	$\dot{Q}_{EVAP} = \dot{m}_2(h_3 - h_2) = \dot{m}_{22}(h_{22} - h_{21})$
Low Temperature Circuit Compressor	$\dot{W}_{LTC} = \dot{m}_2(h_4 - h_3)$
Low Temperature Circuit Expansion Valve	$h_1 = h_2$
Cascade Condenser	$\dot{m}_1(h_1 - h_4) = \dot{m}_5(h_5 - h_{14})$
Condenser	$\dot{Q}_{COND} = \dot{m}_{12}(h_{13} - h_{12}) = \dot{m}_{15}(h_{16} - h_{15})$
High Temperature Circuit Expansion Valve	$h_{13} = h_{14}$
Absorber	$\dot{m}_5 h_5 + \dot{m}_{11} h_{11} = \dot{m}_6 h_6 + \dot{Q}_{ABS}$
Pump	$\dot{W}_P = \frac{\dot{m}_6(P_{12} - P_5)}{\rho_S \eta_B}$
Solution Heat Exchanger	$\dot{m}_9 h_9 - \dot{m}_{10} h_{10} = \dot{m}_8 h_8 - \dot{m}_7 h_7$
Generator	$\dot{m}_{12} h_{12} + \dot{m}_9 h_9 = \dot{m}_8 h_8 + \dot{Q}_{GEN}$
Pressure Reducing Valve	$h_{10} = h_{11}$

Source: The Author

To calculate the isentropic efficiency of the compressors, Eq. 4 was used, according to Sayyaadi and Nejatolahi (2011), P_R represents the compression ratio.

$$\eta_s = 0.85 - 0.046667 P_R \tag{4}$$

The VCCRS performance coefficient (COP) is defined as the ratio between the refrigeration effect and the net work required to achieve that effect according to Eq. 5, Cengel and Boles (2007).

$$COP = \frac{\dot{Q}_{EVAP}}{\dot{W}_{LTC} + \dot{W}_{HTC}} \tag{5}$$

The performance coefficient (COP) for VCACRS is calculated, according to Jain *et al.* (2015) by Eq. Ref eq6.

$$COP = \frac{\dot{Q}_{EVAP}}{\dot{W}_{LTC} + \dot{W}_P + \dot{Q}_{GEN}} \tag{6}$$

To calculate the area of the heat exchangers, Eq. 7 was used according to Bejan and Kraus (2003).

$$\dot{Q} = UA\Delta T_{LM} \tag{7}$$

$$U = \frac{1}{[(\frac{D_o}{D_i})\frac{1}{h_i} + (\frac{D_o}{D_i})F_i + (\frac{D_o}{2k})\ln(\frac{D_o}{D_i}) + F_o + \frac{1}{h_o}]} \tag{8}$$

$$\Delta T_{LM} = \frac{(T_{q,ent} - T_{f,sai}) - (T_{q,sai} - T_{f,ent})}{\ln\frac{(T_{q,ent} - T_{f,sai})}{(T_{q,sai} - T_{f,ent})}} \tag{9}$$

$\dot{Q}$ represents the value of heat absorbed or rejected in the equipment in kW, U represents the global heat transfer coefficient in the equipment, according to Samant (2008) by Eq. 8 given in $\frac{kW}{m^2K}$, A represents the total area of the heat exchange surface in m^2 and ΔT_{LM} represents the difference of the logarithmic mean of temperature between fluids in K, and can be calculated by Eq. 9, according to Bejan and Kraus (2003). The heat exchange area was developed in accordance with Nogueira (2019) and will not be presented in this article.

2.4 Exergetic Analysis

For the exergetic analysis of the systems the Eq. 10 by Gouy-Stodola according to Jain *et al.* (2015) was applied.

$$\dot{E}x_D = T_0 \dot{S}_{GEN} \quad (10)$$

$\dot{E}x_D$ represents the exergy destroyed in kW, T_0 ambient temperature in K and $\dot{S}_{GEN}$ the entropy generation rate in kW/K.

Applying the entropy balance to each component of the systems the exergy destroyed in each equipment is obtained. The results are presented in Tab. 3 and the numerical indices refer to Fig. 2 for VCACRS. The equations are analogous to the VCCRS.

Table 3. Exergy destruction in VCACRS

Component	Exergy destruction
Evaporator	$Ex_{D,EVAP} = T_0[\dot{m}_2(s_3 - s_2) + \dot{m}_{22}(s_{22} - s_{21})]$
Low Temperature Circuit Compressor	$Ex_{D,LTC} = T_0[\dot{m}_3(s_4 - s_3)]$
Low Temperature Circuit Expansion Valve	$Ex_{D,LTEV} = T_0[\dot{m}_1(s_2 - s_1)]$
Cascade Condenser	$Ex_{D,CAS} = T_0[\dot{m}_4(s_1 - s_4) + \dot{m}_{14}(s_5 - s_{14})]$
Condenser	$Ex_{D,COND} = T_0[\dot{m}_{12}(s_{13} - s_{12}) + \dot{m}_{15}(s_{16} - s_{15})]$
High Temperature Circuit Expansion Valve	$Ex_{D,HTEV} = T_0[\dot{m}_5(s_{14} - s_{13})]$
Absorber	$Ex_{D,ABS} = T_0[\dot{m}_6 s_6 - \dot{m}_5 s_5 - \dot{m}_{11} s_{11} - \dot{m}_{19}(s_{20} - s_{19})]$
Pump	$Ex_{D,P} = T_0[\dot{m}_6(s_7 - s_6)]$
Solution Heat Exchanger	$Ex_{D,SHX} = T_0[\dot{m}_7(s_8 - s_7) + \dot{m}_9(s_{10} - s_9)]$
Generator	$Ex_{D,GEN} = T_0[\dot{m}_{12} s_{12} + \dot{m}_9 s_9 - \dot{m}_8 s_8 + \dot{m}_{17}(s_{18} - s_{17})]$
Pressure Reducing Valve	$Ex_{D,PRV} = T_0[\dot{m}_{13}(s_{14} - s_{13})]$

Source: The Author

The total exergy supplied to the system is calculated by Eq. 11 for the VCCRS, according to Cengel and Boles (2007).

$$\dot{E}x_{supply,VCCRS} = \left| \dot{W}_{LTC} + \dot{W}_{HTC} \right| \quad (11)$$

The total exergy supplied to the system is calculated by Eq. 12 and the $T_{mt,GEN}$ is defined as the thermodynamics mean temperature for the generator and can be calculated according to Eq. 13 for the VCACRS, according to (Jain *et al.*, 2015).

$$\dot{E}x_{supply,VCACRS} = \left| \dot{W}_{LTC} + \dot{W}_P + \left(1 - \frac{T_0}{T_{mt,GEN}}\right) \dot{Q}_{GEN} \right| \quad (12)$$

$$T_{mt,GEN} = \frac{(h_{18} - h_{17})}{(s_{18} - s_{17})} \quad (13)$$

The exergy destruction $Ex_{Destruction}$ and the exergetic efficiency of VCACRS is calculated by Eq. 14 and 15.

$$\sum_{i=1}^{n} \dot{E}x_{D,k} \quad (14)$$

$$\eta_{II} = 1 - \frac{\dot{E}_{Destruction}}{\dot{E}_{supply}} \quad (15)$$

The sum of the destroyed exergy is the sum of exergy destruction of each component of the system.

2.5 Economic and Environmental Analysis

In the economic model the equipment purchase cost function developed by Kızılkan *et al.* (2007) was adopted according to Eq. 16. For the compressors purchase cost Eq. 17 developed by Sayyaadi and Nejatolahi (2011) was adopted. The acquisition costs of the expansion valves, pump, refrigerants, pipes and connections were neglected because they represent a cost less than 0.84 % of the total investment cost according to Sanaye and Malekmohammadi (2004). The energy supplied to the generator was considered to come from a renewable energy source so for this equipment any economic operating expenses were neglected.

$$Z = 516.621 A_k + 268.45 \quad (16)$$

$$Z = \frac{573\dot{m}}{0.8996 - \eta_s}(P_R)\ln(P_R) \quad (17)$$

The environmental cost was calculated based on the emission of CO_2 and its impact is calculated according to the carbon cost pricing methodology, also called the carbon rate, which is the amount to be paid for the emission of carbon in the atmosphere to generate electricity by burning fossil fuels. It can be calculated according to Aminyavari *et al.* (2014) by Eq. 18 and Eq. 19.

$$C_{env} = m_{CO_2} C_{CO_2} \tag{18}$$

$$m_{CO_2} = \lambda \dot{W}_t t_{op} \tag{19}$$

The value of $C_{CO_2} = US\$$ 90/ton and $\lambda = 0.968 kg/kWh$ was used. This value is an emission conversion factor of CO_2 given in $\frac{kg}{kWh}$ which represents the amount of gas used to produce $1kWh$ of electricity according to Aminyavari *et al.* (2014). W_t represents the sum of the compressor and pump work in the system in kW and t_{op} the operating time in hours considered 5,000 h according to Jain *et al.* (2016).

The total annual cost of the system is defined as Eq. 20 according to Jain *et al.* (2016).

$$C_T = t_{op}(C_i^{el} \dot{W}_t) + a^c \phi \sum_{k \epsilon Eqs.} Z_k + C_{env} \tag{20}$$

The maintenance factor (ϕ) was considered 1.06 according to Aminyavari *et al.* (2014) and the capital recovery factor (a^c) was calculated by Eq. 21 according to Kızılkan *et al.* (2007). The life time (N) and interest rate (i_R) were considered as 10 years and 15% respectively according to Jain *et al.* (2016). The cost of electricity (C_i^{el}) was considered as 0.075 $US\$/kW$ according to Sayyaadi and Nejatolahi (2011).

$$a^c = \frac{i_R(1+i_R)^N}{(1+i_R)^N - 1} \tag{21}$$

2.6 Optimization

The optimization consists of minimizing the two functions described according to Eq. 14 and Eq. 20.

The optimization method used in the present work is the weighted sum of objectives which consists of transforming the multi-objective problem, determined to be that problem involving the optimization of more than one function, in a mono-objective problem by assigning weights to each objective. When different weights to each objective is assigned a new function is obtained which represents the linear relationship between all objectives according to Eq. 22 Deb (2001). The optimization was performed in the Engineering Equation Solver (EES). The method employed is the genetic algorithms considering a population of 100 individuals, 55 generations and a probability of mutation of 0.01. Thus the Pareto boundary is obtained by varying the weight w from 0 to 1.

Multiobjective optimization is the process of choosing among Pareto-optimal solutions and it involves defining a balance point also called an ideal point. At this point both functions have their minimum values. Such a condition in practice does not exist so the ideal point is unattainable. To achieve a real solution of the minimum values of the functions described simultaneously it is necessary to determine the shortest distance from the ideal point to the curve that defines the Pareto-optimal solutions Jain *et al.* (2016).

$$f(x) = wC_T + (1-w)\dot{E}x_D \tag{22}$$

The weight is represented by w and ranges from 0 to 1.

The thermodynamic input parameters for the base case of VCACRS and VCCRS and the variations of the parameter ranges investigated in the simulation are presented in Tab. 4.

Table 4. Parameters and ranges for the base case

Parameter	Value	Range
Cooling capacity of evaporator [$\dot{Q}_{EVAP}$]	65.0 kW	-
Environmental temperature [T_{env}]	25.0 C	-
Evaporator refrigerant outlet temperature [$T_{out,evap}$]	-30.0 C	-
Evaporation temperature in cascade condenser [$T_{cas,evap}$]	10.0 C	-15.0 C to 15.0 C
Degree of overlap [ΔT_{cas}]	3.3 C	3.0 C to 12.0 C
Condenser temperature [T_{COND}]	35.0 C	35.0 C to 45.0 C
Evaporation temperature [T_{EVAP}]	-45.0 C	-45.0 C to -32.0 C
Condenser temperature in cascade condenser [$T_{cas,cond}$]	13.3 C	-
Generator temperature [T_{GEN}]	85.0 C	81.0 C to 89.0 C
Absorber temperature [T_{ABS}]	40.0 C	36.0 C to 45.0 C
Effectiveness of solution heat exchanger [$\epsilon_{T,SHX}$]	0.6	0.6 to 0.8
Electrical efficiency of pump [η_P]	0.9	-

Source: The Author

3. RESULTS

The simulation performed in the EES program obtained the results for the ideal operating point of each system as shown in Fig. 3 and in Tab. 5.

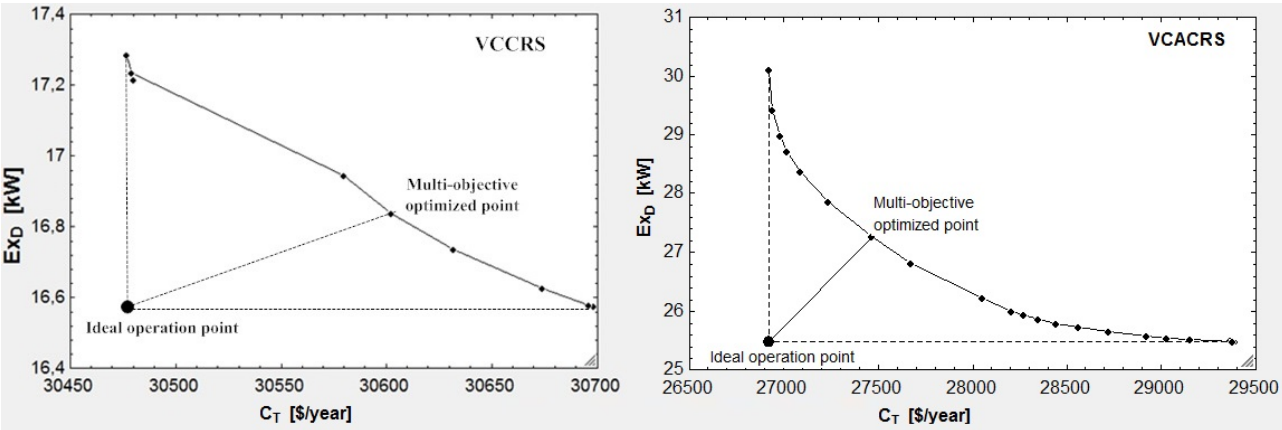

Figure 3. Ideal operation point

Source: The Author

Table 5. Multi-objective optimized point

Parameter	VCCRS	VCACRS
Total annual cost $[C_T]$	30602 $US\$/year$	27462 $US\$/year$
Exergy destruction $[Ex_D]$	16.8 kW	27.3 kW
Evaporation temperature $[T_{EVAP}]$	-32.0 C	-32.0 C
Condensing temperature $[T_{COND}]$	36.2 C	37.4 C
Degree of overlap $[\Delta T_{cas}]$	3.2 C	3.0 C
Evaporation temperature in cascade condenser $[T_{cas,evap}]$	-13.0 C	5.0 C
Generator temperature $[T_{GEN}]$	86.9 C	-
Absorber temperature $[T_{ABS}]$	36.9 C	-
Effectiveness of solution heat exchanger $[\epsilon_{T,SHX}]$	0.8	-

Source: The Author

Figure 3 shows the optimized solutions for both systems. It can be seen that the lowest economic value and the highest value of destruction of exergy are represented at the point closest to the ordinate axis. While the point closest to the abscissa axis represents the highest economic value and the lowest exergy destruction value.

When comparing the optimized results the total annual product cost (C_T) from VCACRS was 10.26% lower than VCCCRS which represents a difference of 3140 $US\$/year$. The cost function consists of operating costs, investment and maintenance costs and finally environmental costs. In Fig. 4 these results are shown.

When analyzing Fig. 4 it is evident that VCACRS is more advantageous in terms of operational and environmental costs. This can be noticed because the VCCRS that is composed by two compressors is expected to consume more electrical energy and these costs functions are directly related to electrical consumption of the system. Regarding the investment and maintenance costs the VCCRS proved to be economic better since VCACRS has more equipments that has a greater sum of area which is directly related to these costs. The greatest impact on the annual cost of the system is the investment and maintenance cost for VCACRS and the environmental cost for VCCRS. For the VCCRS operating costs, environmental costs and investment and maintenance costs represent 39.61%, 46.02% and 14.37% of the total system cost respectively. For VCACRS operating costs, environmental costs and investment and maintenance costs represent 30.00%, 34.85% and 35.15% of the total system cost respectively.

In response to the exergy destruction (Ex_D) the VCCRS proved to be more advantageous in relation to VCACRS resulting in a value 38.46% lower of exergy destruction as shown in Fig. 5.

The higher value of destroyed exergy from VCACRS in relation to VCCRS occurs mainly due to the contributions of destroyed exergy in the low temperature compressor, cascade heat exchanger, low temperature expansion valve, absorber and generator.

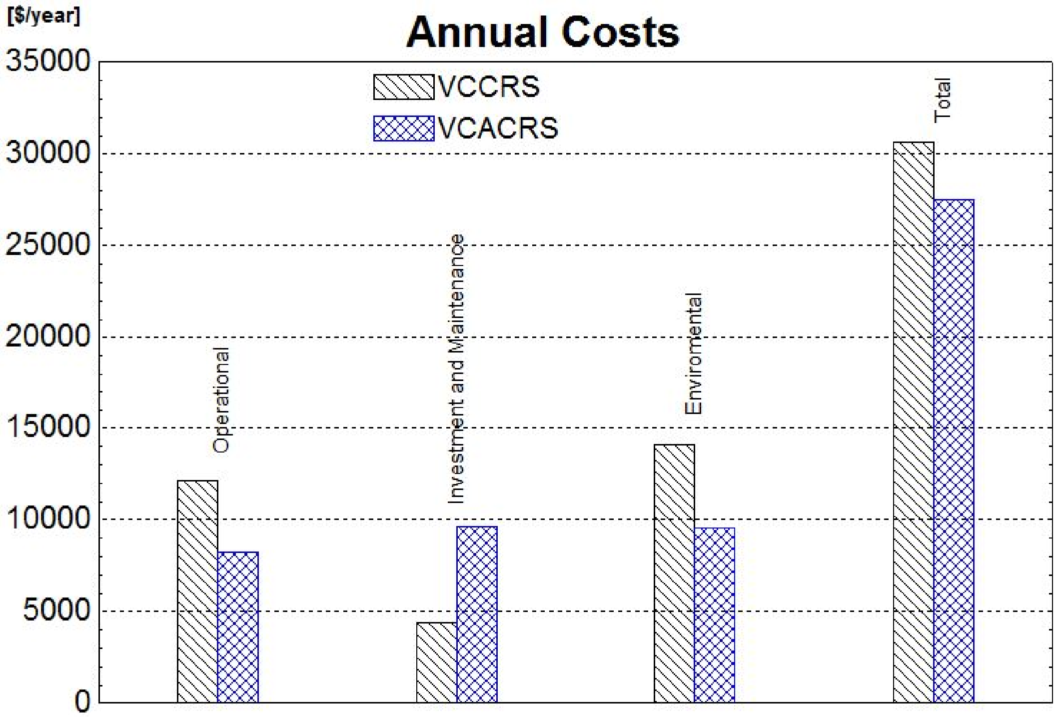

Figure 4. Operating, investment and maintenance, environmental costs

Source: The Author

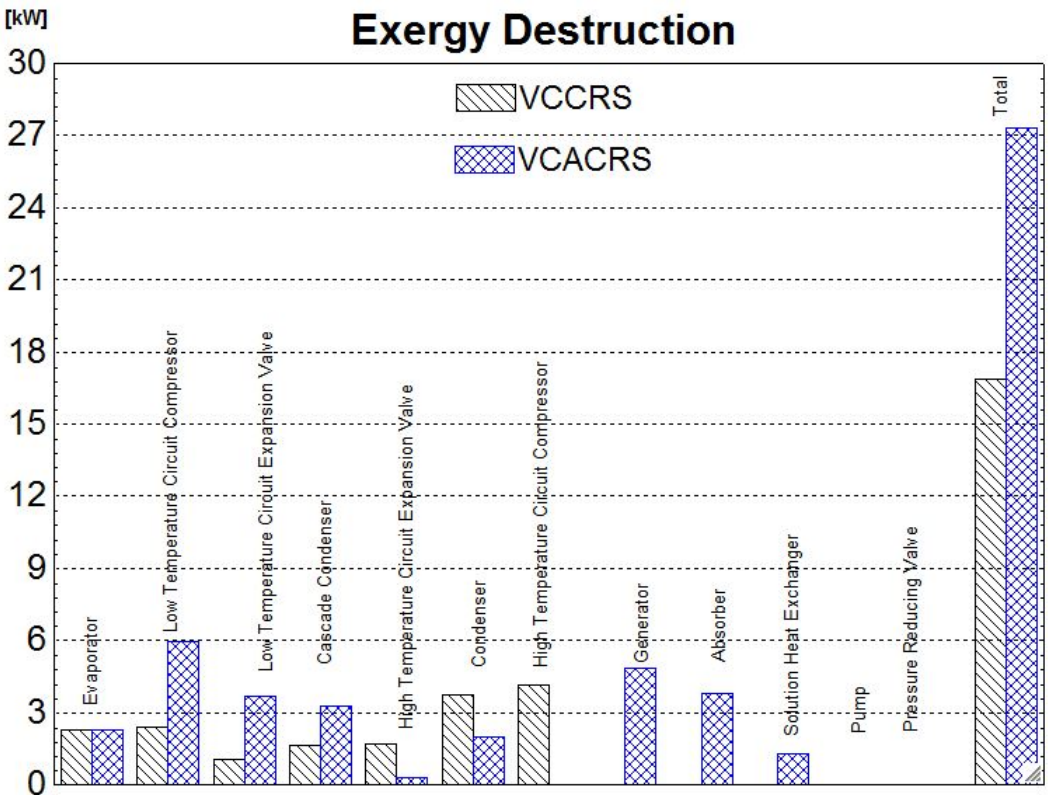

Figure 5. Exergy destruction in VCCRS and VCACRS

Source: The Author

4. CONCLUSION

In this study two refrigeration system models were compared the Vapor Compression Cascaded Refrigeration System (VCCRS) and the Vapor Compression Absorption Cascaded Refrigeration System (VCACRS). Two system models were developed and compared under the thermodynamic aspect through the minimization of the exergy destruction function and under the economic aspect through the minimization of the total annual cost which is composed of the cost of operating the system, acquisition and maintenance costs and environmental cost.

Regarding the economic aspect VCACRS that uses a renewable energy source proved to be more advantageous when compared to VCCRS. The total annual cost (C_T) of VCACRS was 10.26% lower than VCCRS. However it should be noted that in this study the cost associated with the acquisition of a renewable energy system such as solar was not considered. The results showed with respect to the thermodynamic aspect a 38.46% smaller exergy destruction of VCCRS compared to VCACRS which demonstrates that VCCRS has advantage in this aspect.

The present work made a comparison not observed in the articles currently developed which is the application of VCACRS in refrigeration. This system has been studied preferentially for air conditioning using the R410A fluid in the low temperature cycle. Such an approach compared to a conventional VCCRS system brings a potential for study mainly for renewable energy applications for generator heating but in this work its application to the commercial refrigeration system using the R744 was presented.

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