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

NOMENCLATURE

a	capital recovery factor
A	heat exchange surface area, m^2
C	annual cost, US$/year
D	tube diameter, m
$\dot{E}x$	entropy generation, W
F	fouling factor, m^2 K/W
h	specific enthalpy, J/kg
h	heat transfer coefficient, W/m^2K
i_R	interest rate, %
k	thermal conductivity, (W/m K)
m	mass, kg
$\dot{m}$	mass flow rate, kg/s
N	life time, y
P	pressure, Pa
$\dot{Q}$	heat, W
s	specific entropy, J/(K kg)
$\dot{S}$	entropy generation rate, W/K
t	time, h
T	temperature, K
U	global heat transfer, W/m^2 K
w	weight
$\dot{W}$	work, W
Z	purchase cost, US$
t_t	tube thickness, m
T	average fluid temperature, K

Greek symbols

η	efficiency
Δ	difference, K
Σ	sum
λ	emission conversion factor, kg/kWh
ϕ	maintenance factor

Subscripts

c	cold
cond	condenser
D	destroyed
el	eletric
env	environmental
evap	evaporator
gen	generator
h	hot
htc	high temperature circuit
i	inner
in	inlet
LM	logarithmic mean
ltc	low temperature circuit
mt	mean temperature
o	outer
op	operation
out	outlet
P	pump

R	ratio
s	isentropic efficiency
shx	solution heat exchanger

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 619.7TW h 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 103MW h, 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 economic competitiveness in industry. Most supermarket energy costs come 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.

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.

THEORY

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.

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

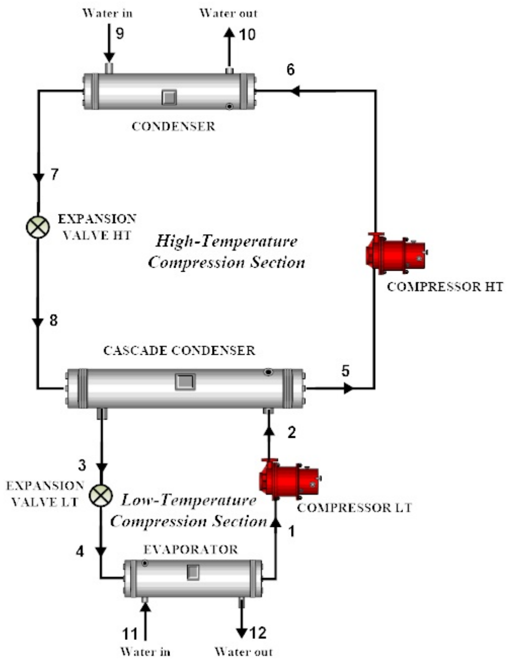

Figure 1. Vapor Compression Cascaded Refrigeration System.

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).

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).

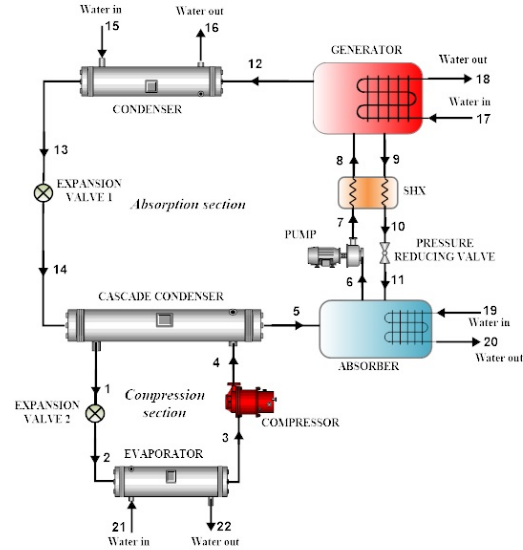

Figure 2. Vapor Compression Absorption Cascaded Refrigeration System

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.

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).

$$\sum \dot{m}_{in} + \sum \dot{m}_{out} = 0 \tag{1}$$

$$\dot{Q} - \dot{W} + \sum \dot{m}_{in} h_{in} - \sum \dot{m}_{out} h_{out} = 0 \quad (2)$$

$$\sum \dot{m}_{out} s_{out} + \sum \dot{m}_{in} s_{in} - \frac{\dot{Q}}{T} = \dot{S}_{gen} \quad (3)$$

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

$$\eta_s = 0.85 - 0.046667 P_R \quad (4)$$

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).

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

The performance coefficient (COP) for VCACRS is calculated, according to Jain et al. (2015) by Eq. 6.

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

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

$$\dot{Q} = UA\Delta T_{LM} \quad (7)$$

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

$$\Delta T_{LM} = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\ln\frac{(T_{h,in} - T_{c,out})}{(T_{h,out} - T_{c,in})}} \quad (9)$$

$\dot{Q}$ 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 kW/m^{2}K, A represents the total area of the heat exchange surface in m^2 and ΔTLM 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.

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_o ambient temperature in K and $\dot{S}_{gen}$ the entropy generation rate in kW/K.

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

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

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).

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

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

The exergy destruction and the exergetic efficiency of VCACRS is calculated by Eq. 14 and 15.

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

$$\eta_{II} = 1 - \frac{\dot{E}x_D}{\dot{E}x_s} \quad (15)$$

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

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.

$$Z = 516.621A_k + 268.45 \tag{16}$$

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

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 λ = 0.968 kg/kWh was used. This value is an emission conversion factor of CO2 given in kg/kWh which represents the amount of gas used to produce 1 kWh of electricity according to Aminyavari et al. (2014). Wt represents the sum of the compressor and pump work in the system in kW and top 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).

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

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).

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

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).

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

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

The cooling capacity of evaporator ($\dot{Q}_{EVAP}$) is considered as 65.0 kW, the environmental temperature (T_{env}) 25.0 C, the evaporator refrigerant outlet temperature ($T_{evap,out}$) – 30.0 C, the condenser temperature in cascade condenser ($T_{cas,cond}$) 13.3 C, the electrical efficiency of pump (η_P 0.9. The evaporation temperature in cascade condenser ($T_{cas,evap}$) can vary from –15.0 to 15.0 C, the degree of overlap (ΔT_{cas}) 3.0 to 12.0 C, condenser temperature (T_{cond}) 35.0 to 45.0 C, evaporation temperature – 45.0 to – 32.0 C, generator temperature (T_{gen}) 81.0 to 89.0 C, absorber temperature (T_{abs}) 36.0 to 45.0 C, effectiveness of solution heat exchanger ($\epsilon_{T,shx}$) 0.6 to 0.8.

RESULTS

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

The optimized values for each parameter of both systems are total annual cost of 30602 US$/year for VCCRS and 27462 US$/year for VCACRS, exergy destruction of 16.8 kW for VCCRS and 27.3 kW for VCACRS, evaporation temperature of -32.0 C for VCCRS and -32.0 C for VCACRS, condensing temperature of 36.2 C for VCCRS and 37.4 C for VCACRS, degree of overlap 3.2 C for VCCRS and 3.0 C for VCACRS, evaporation temperature in cascade condenser of -13.0 C for VCCRS and 5.0 for VCACRS, generator temperature of 86.9 C, absorber temperature of 36.9 C, effectiveness of solution heat exchanger of 0.8 for VCACRS.

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 3,140 US$/year. The cost function consists of operating costs, investment and maintenance costs and finally environmental costs. In Fig. 4 these results are shown.

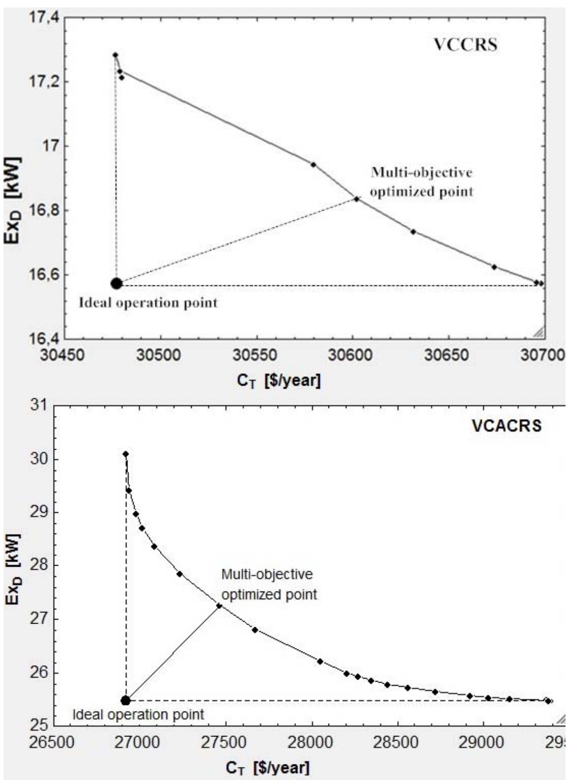

Figure 3. Ideal operation point.

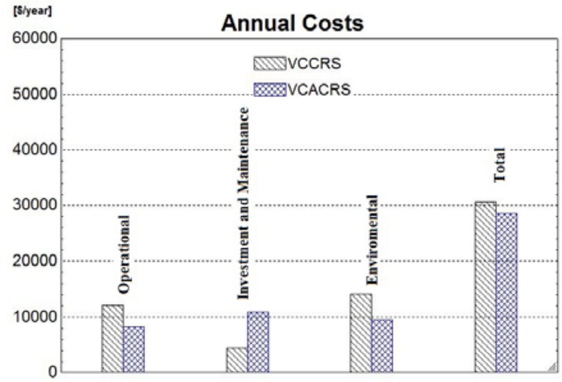

Figure 4. Operating, investment and maintenance, environmental costs.

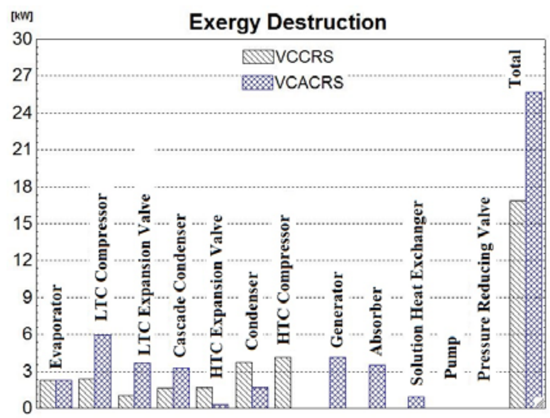

Figure 5. Exergy destruction in VCCRS and VCACRS.

CONCLUSIONS

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.

RESPONSIBILITY NOTICE

The authors are solely responsible for the printed material included in this paper.

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