XL CILAMCE
IBERO-LATIN AMERICAN CONGRESS ON COMPUTATIONAL METHODS IN ENGINEERING
NOVEMBER 11-14, 2019
Praiamar Natal Hotel & Convention
Natal, RN-BRAZIL

ENERGETICAL ANALYSIS OF A REFRIGERATION CYCLE COMBINED WITH CO_2, PROPANE AND BRINE

Lívia M. S. de Pontes
Josegil J. P. de Araújo
liviadepontes@gmail.com
josegil.ufs@gmail.com
Universidade Federal de Sergipe
Marechal Rondon Ave, 49100-000, Sergipe/Aracaju, Brazil

Abstract. The destruction of the ozone layer was mainly caused by chlorofluorocarbons (CFCs) used as refrigerants, which were widely utilized during the twentieth century. Another problem produced by the use of these substances was, and still is, the Global Warming, since these fluids have high "Global Warming Potential" (GWP). Due to these consequences, this category of refrigerant was banned and gradually replaced by natural refrigerants such as carbon dioxide (CO_2 – R744) and propane (C_3H_8 – R290). In this work a cascade refrigeration system was analyzed according to the First Law of Thermodynamics (energy analysis). The cascade cycle consists of three subsystems: a vapor compression cycle using R744 as a working fluid, a system operating with 40% ethylene glycol and a vapor compression cycle which uses R290 as a refrigerant. The propane cycle receives heat that has been absorbed in cycles of R744 and 40% ethylene glycol and rejects it into the atmosphere in the condenser of the R290 cycle. For the cycle analysis, the system was subdivided into 18 control volumes and 23 state points. Equations of mass and energy balances were applied, along with complementary equations, which they were used to complete the system of equations. The solution of the system of equations was obtained by applying Newton-Raphson method. The model was implemented in EES$^{®}$ and simulated under different operating conditions. Finally, the thermal efficiency of the R744 cycle was 8.784; for the R290 cycle, it was 4.09 and 3.393 was the result of the combined cycle.

Keywords: Natural refrigerants, Energy analysis, Newton-Raphson Method, Carbon dioxide, Propane

1 Introduction

As widely disseminated on the media, the planet suffers from two serious environmental problems: the destruction of the ozone layer and the global warming. Since the discovery of these climatic conditions, human beings have been performing some actions in order to minimize or completely solve these adversities. Regarding the destruction of ozone layer, ozone (O_3), responsible for the protection against ultraviolet rays of type B (UR-B), began to react with chlorine (Cl) present in refrigerants of the type chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC). This caused a reduction in the concentration of O_3 in the stratosphere and, consequently, caused risks to human health and environmental impacts. It is in this context that the famous Montreal protocol appeared in 1987. Due to it that 150 countries have committed themselves to replacing substances that were responsible for destroying the ozone layer. According to Calm [1], after the signature of this protocol, the production of the CFC refrigerants was terminated in the United States at the beginning of 1996, and then replaced by hydrofluorocarbon (HFC), starting the third generation of refrigerants. However, this new generation solved only a part of the problem.

Looking from the perspective of global warming, it was observed an elevation of the average temperature of the oceans and the atmosphere due to the greenhouse effect. During the studies, it was perceived that the main cause of this problem was the excessive emission of gases that possess the ability to absorb infrared radiation, in other words, substances that have high Global Warming Potential (GWP), among them, the CFCs, HCFC and HFC were once again highlighted. This time, the solution occurred through the Kyoto protocol proposed in 1988, in which 192 states assumed the rigid commitment to reduce the emission of gases that aggravated this environmental problem. After the signature, there was a drastic reduction in the consumption of these harmful gases and many countries were able to reduce the emission in values even lower than what had been established.

For both protocols to continue to be respected, it was necessary to invest in technology and change in several areas in order to obtain efficient solutions in relation to the substitution of prohibited substances. These changes directly influenced the refrigeration systems, because, according to Pereira [2], the CFCs became the main refrigerant fluid from the period of 1930 and 1940 when they were considered safer than the older refrigerants, such as natural gases. However, Pereira [2] states that the disadvantages in relation to the use of natural gases such as high cycle pressure and low power coefficient, when the temperature in the side of heat rejection is high, which generated the reduction of its consumption after the Second World War, were overcome. In this way, natural gases in refrigeration systems have become the ideal substitute, because they have an Ozone Depletion Potential (ODP) equal to zero and a reduced GWP when they are compared to CFCs, as can be seen in Table 1.

Table 1. ODP and GWP values according to the type of refrigerant. Source: GIZ Proklima

Refrigerant type:	Examples:	ODP:	GWP:
CFC	R11 and R12	0.60 – 1.00	4750 – 14400
HCFCs	R22 and R141b	0.02 – 0.11	77 – 2310
HFC	R32 and R134a	–	124 – 14800
Natural refrigerants	R744 (carbon dioxide), R717 (ammonia) and R290 (propane)	–	0 – 20

The use of natural gases in refrigeration systems has become increasingly widespread, so an in-depth analysis of these thermal systems becomes necessary with the goal of optimizing those systems. So, simulation and modeling are extremely important because they allow a deep analysis, generating results that enable the study of the system in several scenarios and parameters, aiming to obtain the best performance.

The mathematical model applied for the First Law analysis is illustrated in the article written by Araújo [3] and a similar approach will be used in this study. Araújo [3] studied, modeled and simulated a cascade system composed of ammonia (R717) and CO_2 (R744). The R744 circulates through a subcritical cycle at high pressure, which has two evaporation temperature levels, one at medium temperature and another at low temperature with direct expansion. In relation to R717, this refrigerant is located in the low pressure cycle. This paper concluded through the First Law that a cascade system using natural fluids can achieve good results, such as a maximum cycle efficiency of 66.56%.

The application of mathematical computational systems to solve problems of energy systems are being applied in several refrigeration cycles that use natural gases, in which stands out cascade systems that use CO_2 and propane. This is observed in the study conducted by Sawalha [4], it was stated that cascade cycles that use CO_2 in the low temperature stages are being applied in several supermarket facilities and are becoming an increasingly competitive alternative.

In this way, according to the description of Johansson [5], a cascade refrigeration system uses two refrigerant fluids, which they are operating in separate refrigeration cycles. This assembly is performed in order to have systems with lower temperatures and to avoid high compression work due to high pressure differences, which provides high temperature at the compressor outlet. One of the systems operate at a high temperature stage while the other one is in the low temperature stage. The two cycle are connected by two counterflow heat exchangers where both equipments functions as evaporators for the propane cycle side. One of these heat exchangers works as the high temperature heat exchanger for the brine cycle side. The other one operates as the condenser for the R744 cycle side.

Thus, the paper in question realizes the modeling and simulation of a combined refrigeration system (CO_2, 40% ethylene glycol and propane), shown in Fig. 1, through the optics of energy analysis. In this way, the cascade cycle was divided into three subsystems: a vapor compression cycle using the R744 as a working fluid, which its saturation temperature at the evaporator is -32 °C (241.15 K) and its thermal load is 10 kW; a system that uses 40% ethylene glycol as working fluid, which its temperature at the inlet of the low temperature heat exchanger is -11 °C (262.15 K) and its thermal load is 21 KW and a vapor compression cycle which operates with R290, whose saturation temperature is -14 °C (259.15 K) for the two heat exchangers of counterflows. The R290 cycle receives the heat absorbed in the cycle of R744 and the system of 40% ethylene glycol, then it rejects in the atmosphere through a condenser, that operates at a condensing temperature of 35 °C (308.15 K).

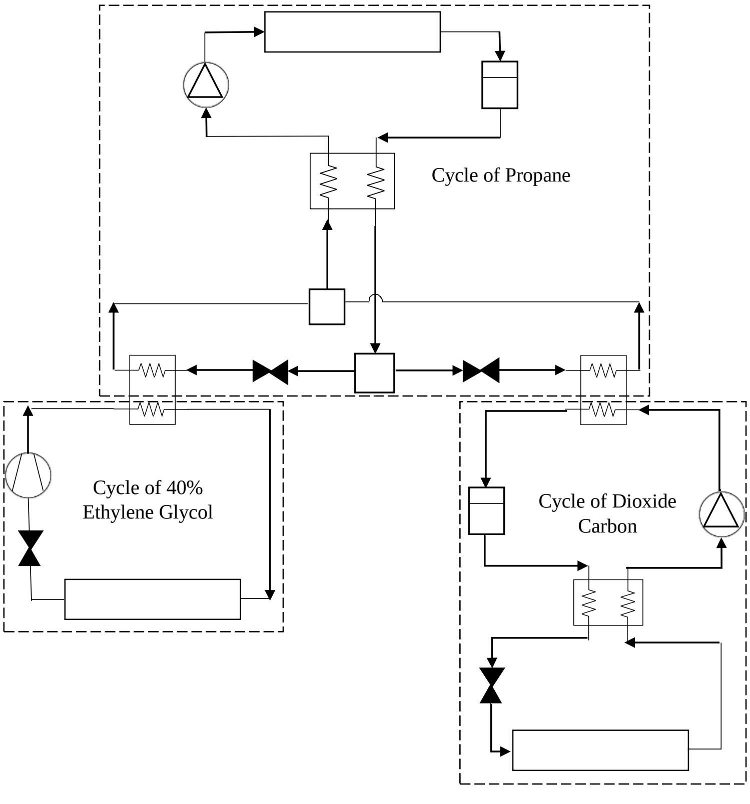

Figure 1. Simplified modeling of the cooling cascade system using CO_2, propane and 40% ethylene glycol

2 Theoretical foundations

During the application of the balance equations in the control volume, the following simplifications were considered: the system operates at steady state; the effects of kinetic and potential energy were neglected; the control volumes operate in an adiabatic process, except the evaporators, the condenser and the low temperature heat exchanger of the brine cycle; the process of expansion and reduction of pressure in the valves were considered to be isenthalpic; the compression and pumping process were estimated to be isentropic; 40% ethylene glycol was studied as being incompressible liquid, in this case, the enthalpy of the state points will be found through this consideration and the pressure losses within the cycles will not be addressed.

2.1 Mass Balance

Çengel [6] described that the mass balance follows the Principle of Mass Conservation, in which the mathematical expression is:

$$\sum \dot{m}_i - \sum \dot{m}_e = \frac{dm_{cv}}{dt} = 0. \tag{1}$$

At Eq. (1), ṁ is the mass flow, the subscript "i" and "e" represents, respectively, the input and exit of the control volume, in which this symbology is used throughout the article. Moreover, $\mathrm{dm_{CV}/dt}$ is the rate of mass variation within the boundaries of the control volume that is reduced to zero when the system operates at steady state as is the case in study.

2.2 Energy Balance

According to Moran [7], the energy balance, with the considerations that were made, can be express as:

$$0 = \dot{Q}_{cv} - \dot{W}_{cv} + \sum \dot{m}_i\, h_i - \sum \dot{m}_e h_e. \tag{2}$$

Equation (2) encompasses general situations, however, not all equipment works with all types of transference occurring at the same time. Table 2 shows the energy balances for equipments present in the cycle studied in this article.

Table 2. Energy balance in various equipments present in a refrigeration system

Components:	Energy Balance:
Liquid accumulators, separators and mixers	$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$
Pumps and compressors	$\dot{W}_i + \dot{m}_i h_i = \dot{m}_e h_e$
Condenser	$\dot{Q}_e + \dot{m}_e h_e = \dot{m}_i h_i$
Evaporator	$\dot{Q}_i + \dot{m}_i h_i = \dot{m}_e h_e$
Heat recuperator	$\dot{m}_i h_i = \dot{m}_e h_e$
Expansion and reduction valve	$h_i = h_e$

2.3 Efficiency of The First Law of Thermodynamics

The efficiency of a refrigeration cycle is defined, according to Çengel [6], through the Coefficient of Performance (COP_R), that is the ratio between the amount of heat that the system receives from the cold body and the work that enters in the system with the aim of cooling this body. In other words, Kanoglu [8] explains that the COP_R is the ratio between what is desirable to obtain (absorb heat or desired output) and how this goal can be achieved (net work or required input). That being said, the COP_R can be expressed as:

$$COP_R = \frac{\dot{Q}_i}{\dot{W}_i}, \tag{3}$$

where $\dot{Q}_i$ is the absorbed heat from the cold reservoir and $\dot{W}_i$ is the input work to the cycle. The COP_R is a measure of efficiency of a real cycle. In this context, it can be compared with the system that provides the highest theoretical performance coefficient, the refrigeration Carnot cycle. This cycle operates with reversible processes between a reservoir at cold temperature and another at hot temperature, in which, according to Moran [7], its COP is given by:

$$COP_{R,\ Carnot} = \frac{T_C}{T_H - T_C}. \tag{4}$$

In Eq. (4), T_C corresponds to the cold reservoir temperature in Kelvin and T_H is the temperature value of the hot reservoir in Kelvin. Thus, it is worth noting that the closer to Carnot a cycle is operating, more similar to reversibility the cycle will be operating, which according to Kanoglu [8] is the perfection for thermodynamics. However, it cannot be reached or surpassed in real systems.

3 System modeling

For the study of the combined refrigeration system illustrated in Fig. 2, the cycle was divided into 18 control volumes, which they are the pump of the brine system (Pump. Brine), the compressors of CO_2 and propane cycles (Comp. R744 and Comp. R290, respectively), the condensers of CO_2 and propane cycles (Cond. R744 and Cond. R290, respectively), the evaporators of CO_2 and propane cycles (Evap. R744, Evap. 1. R290 and Evap. 2. R290, respectively), the high temperature heat exchanger (HT. HE. Brine) and low temperature heat exchanger (LT. HE. Brine) of the brine system, the heat recuperators of CO_2 and propane cycles (HR. R744 and HR. R290, respectively), the separator of propane cycle (Sep. R290), the mixer of propane cycle (Mix. R290), the liquid accumulators of CO_2 and propane cycles (Acum. R744 and Acum. R290, respectively), the expansion valves of CO_2 and propane cycles (TV. R744, TV. 1. R290 and TV. 2. R290, respectively) and the reduction valve of brine system (RV. Brine). Moreover, the system has 23 points of states, which are generally characterized as the input and output of the control volume, in which can be study the properties of the fluid before and after the processes occur. In order to determine the properties of the refrigerants of R744, R290 and 40% ethylene glycol, the information from the EES® library was used.

The CO_2 cycle has been designed to keep an environment at a low temperature ranging from -25 °C to -18 °C. For this purpose, an evaporator (Evap. R744) operating at -32 °C (241.15 K) is used and has the initial thermal load of 10 kW, which may vary according to the operating condition. After the R744 absorbs heat into the evaporator and becomes saturated vapor, it passes through a heat recovery (HR. R744) that has the function of ensuring that superheated vapor enter the compressor (Comp. R744), located after the heat recuperator. Then the absorbed heat flows from the low temperature cycle to the high temperature cycle through a heat exchanger that functions as condenser for the R744 cycle (Cond. R744), whose condensing temperature is -10 °C (263.15 K), on the other hand, it functions as an evaporator for the R290 cycle (Evap. 2. R290), which operates with an evaporating temperature equivalent to -14 °C (259.15 K). In order to ensure that the condenser output is in saturated liquid, the refrigerant passes through a liquid separator (Acum. R744) and follows to the high temperature side of the heat recovery, where will be reduced to a temperature equal to 7 °C. Finally, the fluid, which enters as compressed liquid into the expansion valve, is throttled and the cycle restarts at the inlet of the evaporator.

Looking at the brine cycle, it is held that it is responsible for maintaining an environment between -1 °C and 2 °C. In order to achieve this goal, its low temperature heat exchanger (LT. HE. Brine) has the inlet temperature of -11 °C (262.15 K) and operates normally with a thermal load of 21 kW. After the heat exchanger, the brine passes through a reduction valve (RV. Brine), whose objective is to indicate a pressure drop, enabling the operation of the pump (Pump. Brine). Thus, the fluid circulates in a closed system due to the action of the recirculation pump that promotes a difference of pressure of 0.7 bar (70 kPa) and has a volumetric flow rate of 5.6 m^3/h, in which these data were informed by Bertelsen [9]. Then, the heat is rejected in the propane cycle through a heat exchanger that functions as a high temperature heat exchanger for the 40% ethylene glycol cycle (HT. HE. Brine), whose inlet temperature is approximately -7 °C (266.15 K) and serves only to reduce the temperature of the brine, and works as evaporator for the R290 cycle (Evap. 1. R290) which operates in a saturation temperature equal to -14 °C (259.15 K). After this exchanger, the brine enters again into the low temperature heat exchanger where it will be heated, restarting the cycle.

Finally, the R290 cycle absorbs all the heat from the other cycles through the heat exchangers that function as evaporators, operating at a saturation temperature equal to -14 °C (259.15 K). Then, the refrigerant at saturated vapor is directed to a mixer (Mix. R290) that connects the both sides responsible for the heat absorption of the CO_2 cycle and the brine system. Subsequently, the flow is directed to the low side temperature heat recovery (HR. R290), in order to ensure that superheated vapor is in the compressor (Comp. R290). After that, the pressure rises and the R290 enters the condenser (Cond. R290) which has the condensing temperature of 35 °C (308.15 K). By rejecting the heat to the ambient at 25 °C, the fluid is directed to a liquid accumulator, obtaining saturated liquid in the output. Then, it is directed to the high temperature side of the heat recovery. Finally, after a cooling of 5 °C, the mass flow of the cycle is separated into the separator (Sep. R290) and a part goes to the side of the heat exchanger brine-propane and the other part is directed to the heat exchanger CO_2-propane. Then, on both sides, the fluid is throttled to a lower pressure and the cycle recommences.

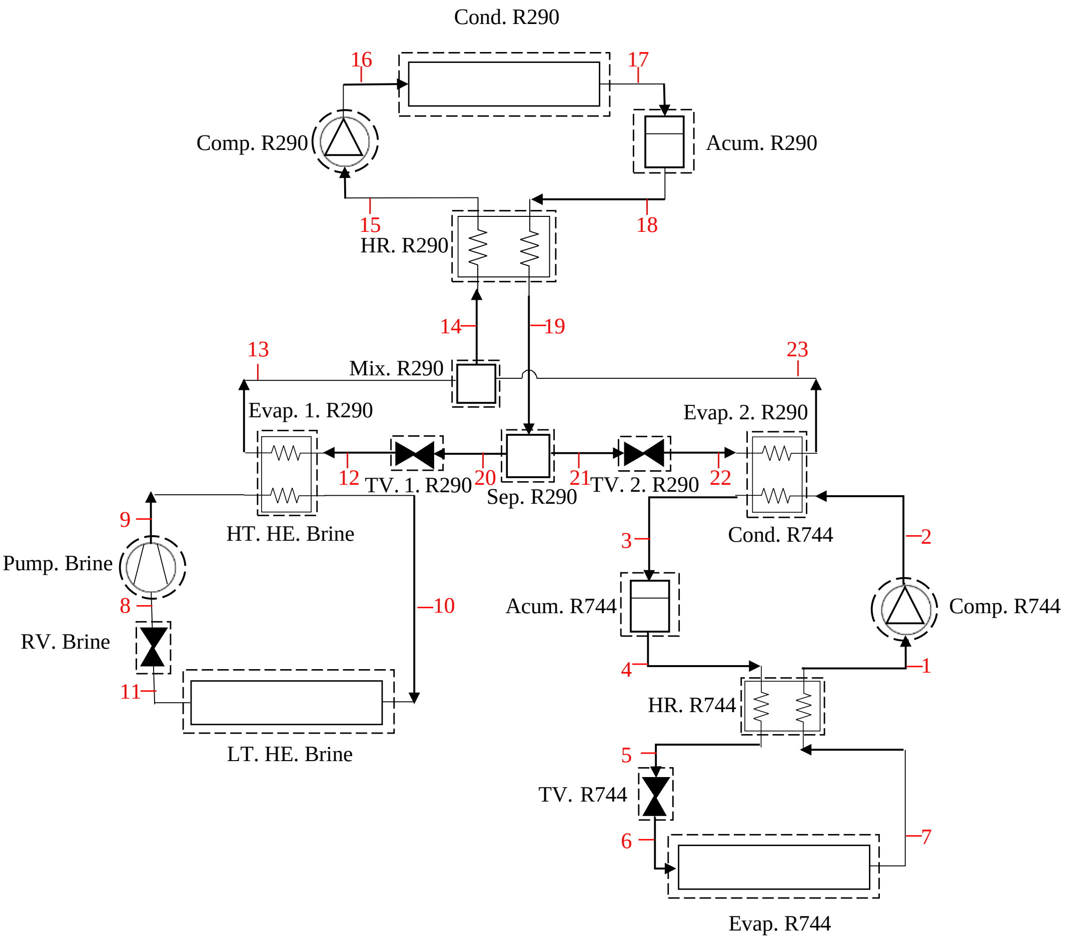

Figure 2. Cascade cooling system using R744, 40% ethylene glycol and R290

As it can be seen, through the analysis of the system, there is a way to determine the properties of all state points. Therefore, it can obtain the results that illustrates the operability of the system through the perception of the First Law. In relation to the input data, the initial conditions are shown in Table 3 and they will be applied in the model implemented in the EES®, in order to simulate the system and understand its functioning in different situations.

Table 3. Initials conditions used for system analysis

Cycle:	**Variable name:**	**Assigned value:**
R744	Evaporator saturation temperature	-32 °C
	Evaporator thermal load	10 kW
	Quality at state point 7	1
	Condenser saturation temperature	-10 °C
	Quality at state point 4	0
	Temperature difference between points 4 and 5 (cooling)	7 °C
40% ethylene glycol	Temperature at the inlet of the low temperature heat exchanger	-11 °C
	Thermal load in low temperature heat exchanger	21 kW
	Temperature at the inlet of high temperature heat exchanger	≈ -7 °C
	Pressure difference in pump and reduction valve	70 kPa
R290	Evaporator saturation temperature	-14 °C
	Quality at state points 13 and 23	1
	Condenser saturation temperature	35 °C
	Quality at state point 18	0
	Temperature difference between points 18 and 19 (cooling)	5 °C

4 Results and discussions

Considering the balance equations applied in the control volumes, the equations for closing the equation system and the initial conditions, the implementation in the EES® resulted in 245 equations with 69 unknowns. For the simulation, the following stop criteria were adopted: number of iterations equals to 250; the relative residual error of 1 x 10^{-6} and tolerance in the change of variables is 1 x 10^{-9}.

Considering the initial conditions shown in Table 3, the system solution was performed in 1 second. The maximum residual error obtained was 9.3319 x 10^{-10} and the tolerance in the maximum variable change obtained was 9.9000 x 10^{1}.

4.1 Determination of state points

The determination of all properties (enthalpy, entropy, quality, pressure and temperature) of each state point was analyzed according to some assumptions, the initial conditions (Table 3). In addition, utilizing the considerations described in Item 2, the state points for the cycles are shown in the Table 4.

Through Table 4, it is observed that, in the cycle of CO_2, the upper limit of pressure is 2649 kPa and the lower is 1334 kPa. For the propane cycle, the maximum pressure is 1218 kPa, while the minimum corresponds to 303.8 kPa. In relation to 40% ethylene glycol, the upper pressure is 1250 kPa, which remains constant in practically the whole cycle, except in the output of RV. Brine, which represents the loss of pressure, its value is 1180 kPa. In this context, the CO_2, besides having the highest pressure variation (1315 kPa), also presents the highest upper pressure. On the other hand, the brine cycle shows the smallest pressure difference, while the propane system has the lowest pressure value.

In relation to temperature, the cycle of R290 stands out, because it has the greatest difference between the maximum and minimum temperatures, which its value is 63.47 K, complementing this information, this cycle also has the highest temperature of the system (322,619 K at the state point 16). The R744 cycle presents a variation between the upper and lower temperature of 61.43 K and the lowest system temperature (241.15 K at state point 6). The 40% ethylene glycol cycle presents a temperature variation corresponding to 3.62 K.

With the data obtained in Table 4, it was possible to perform the P-h and T-s diagrams for the cycles of R744 and R290. With the objective of illustrate the position of each state point in the cycle (Fig. 2) inside the phase envelope.

The P-h and T-s diagrams for the CO_2 and propane cycles are illustrated in Fig. 3 and Fig. 4, respectively. It is possible to realize that the operation of both cycles is similar. In this context, the systems operates between two pressure levels, which correspond to the temperature saturation pressure of the evaporators (-32 °C for the CO_2 cycle and -14 °C for the propane cycle) where the heat is absorbed and of the condensers (-10 °C for the low temperature cycle and 35 °C for the high temperature cycle) where heat is rejected.

Table 4. Properties of all state points

Cycles:	**State Point:**	**Temperature (K):**	**Pressure (kPa):**	**Enthalpy (kJ/kg):**	**Entropy (kJ/kg-K):**	**Quality:**	**Mass flow (kg/s):**
R744	1	255.858	1334.0	-54.45	-0.6678	100	0.0363
	2	302.576	2649.0	-23.05	-0.6678	100	0.0363
	3	263.150	2649.0	-330.20	-1.8230	0	0.0363
	4	263.150	2649.0	-330.20	-1.8230	0	0.0363
	5	256.150	2649.0	-345.90	-1.8840	-100	0.0363
	6	241.150	1334.0	-345.90	-1.8750	0.1029	0.0363
	7	241.150	1334.0	-70.16	-0.7311	1	0.0363
40% ethylene glycol	8	265.768	1180.0	-60.62	18.9000	-	1.7140
	9	265.788	1250.0	-60.55	18.9000	-	1.7140
	10	262.150	1250.0	-72.87	18.8500	-	1.7140
	11	265.768	1250.0	-60.62	18.9000	-	1.7140
R290	12	259.150	301.8	279.30	1.3110	0.2894	0.0755
	13	259.150	301.8	558.90	2.3890	1	0.0755
	14	259.150	301.8	558.90	2.3890	1	0.1154
	15	267.640	301.8	573.00	2.4430	100	0.1154
	16	322.619	1218.0	641.30	2.4430	100	0.1154
	17	308.150	1218.0	293.40	1.3160	0	0.1154
	18	308.150	1218.0	293.40	1.3160	0	0.1154
	19	303.150	1218.0	279.30	1.2700	-100	0.1154
	20	303.150	1218.0	279.30	1.2700	-100	0.0755
	21	303.150	1218.0	279.30	1.2700	-100	0.0398
	22	259.150	301.8	279.30	1.3110	0.2894	0.0398
	23	259.150	301.8	558.90	2.3890	1	0.0398

Figure 5 and Figure 6 show with detailed the cycles. Beginning the analysis, the refrigerants at the outlet of the evaporators (point 7 in the cycle of CO_2, points 13 and 23 in the propane cycle in Fig. 3) are as saturated vapor. Then the flow is directed towards the lower temperature side of the heat recuperator, where the refrigerant is heated at constant pressure and becomes superheated vapors (points 1 in the cycle of CO_2 and point 15 in the propane cycle in Fig. 3). Afterward, in the respective compressors, isentropic compression occurs, which is characterized by the vertical line in the superheated vapor region of the T-s diagram, corresponding to the process 1-2 in the CO_2 cycle, whose the work generated is 1.138 kW, and process 15-16 in the propane cycle, which has a work of 7.885 kW. After that, the heat is rejected at a constant pressure for the high temperature cycle, in the case of the R744, represented by the process 2-3, which rejects 11.138 kW, or for the environment, in the case of the R290, which is shown in the process 16-17, rejecting 40.136 kW. The refrigerant passes through a liquid accumulator, whose the output is saturated liquid, corresponding to point 4 in the cycle of CO_2 and point 18 in the propane cycle. Subsequently, the fluid will be directed to the high temperature side of the heat recuperator, which will be cooled and passed to the state of compressed liquid, indicated by point 5 in the cycle of CO_2 and point 19 in the propane cycle. Finally, the fluid will be throttled through an isenthalpic process in the expansion valves, as shown in process 5-6 in the cycle of CO_2 and processes 21-22 and 20-12 in the propane cycle, and the output obtained is a liquid-vapor mixture at the entrance of evaporators, given by point 6 in the cycle of CO_2 and points 12 and 22 in the propane cycle.

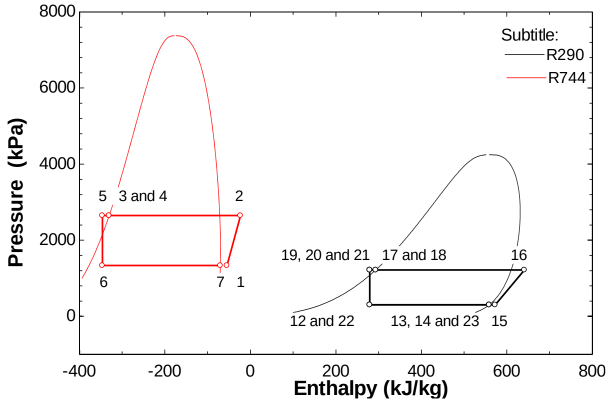

Figure 3. Diagrams P-h for the cycles of R744 and R290

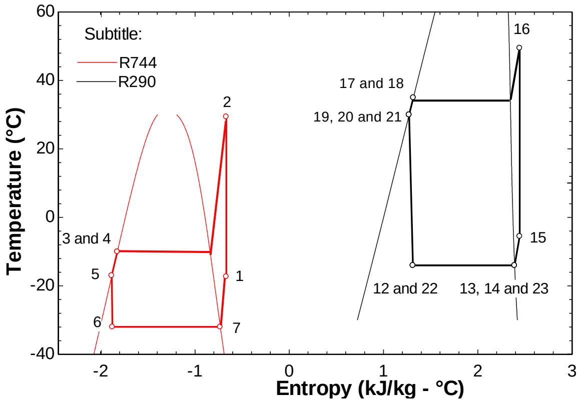

Figure 4. Diagrams T-s for the cycles of R744 and R290

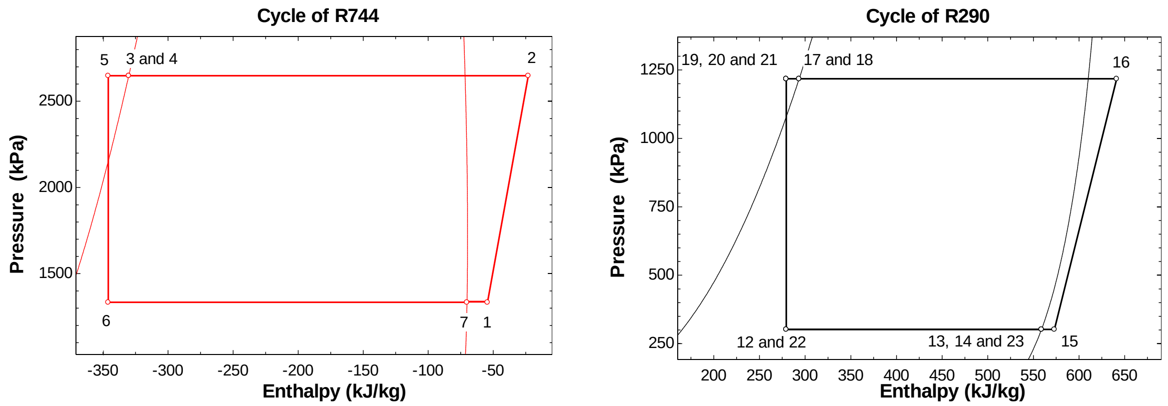

Figure 5. Diagrams P-h for the cycles of R744 and R290 enlarged

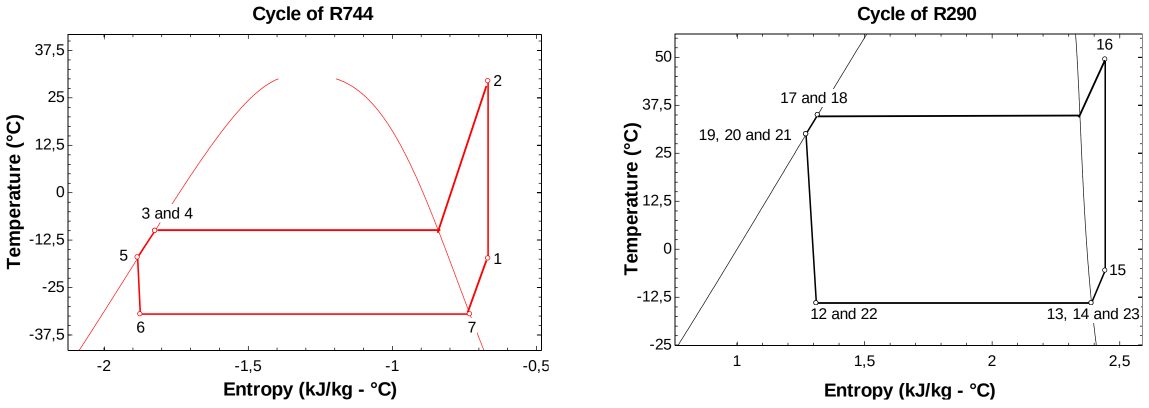

Figure 6. Diagrams T-s for the cycles of R744 and R290 enlarged

4.2 Analysis of the First Law of Thermodynamics

Table 5 shows the following Coefficients of Performance: COP_R, $COP_{R,\ Carnot}$ and the cycle's efficiency (ratio between COP_R and $COP_{R,\ Carnot}$), for the cycle of CO_2, for the propane cycle and for the combined cycle. The 40% ethylene glycol system does not have its COP determined, because this cycle was regarded to be a simple system of cooling and heating of the brine, without phase change of the working fluid. In addition, Table 5 illustrates also the work supplied and the thermal energy transference in or out of the system.

Table 5. First Law of Thermodynamics results

Cycle:	**COP_R:**	**$COP_{R,\ Carnot}$:**	$\frac{COP_R}{COP_{R,\ Carnot}}$:	**Component:**	**Work (kW):**	**Thermal energy transference (kW):**
R744	8.784	31.39	27.980%	Compressor	1.138	-
				Cascade heat exchanger CO_2-propane	-	-11.138
				Evaporator	-	10. 000
40% ethylene glycol	-	-	-	Pump	0.113	-
				High temperature heat exchanger	-	-21.113
				Low temperature heat exchanger	-	21. 000
R290	4.090	7.52	54.402%	Compressor	7.885	-
				Condenser	-	-40.136
				Cascade heat exchanger CO_2-propane	-	11.138
				Cascade heat exchanger brine-propane	-	21.113
Combined System	3.393	5.34	63.498%	Total input work	9.136	-
				Total absorbed heat	-	31.000
				Total rejected heat	-	-40.136

Analyzing the results exhibited in Table 5, it can be concluded that each cycle operating individually, as also the combined system, obey the Principle of Energy Conservation and, therefore, they are following the First Law of Thermodynamics. In other words, the sum of the energies that enters is equal to the sum of the energies that exits, as explained by Moran [7]. Furthermore, in Table 5, the negative sign for some values in the column of thermal energy transference represents that energy is exiting of the component or system.

From the inspection of Table 5, it can be also observed that the cycle of CO_2 has a smaller cycle efficiency, 27.98%, than the cycle of propane, that has this value equal to 54.402%. Moreover, the COP_R of the combined cycle, 3.393, is lower than the R744 cycle (8.784) and the R290 cycle (4.09). However, the efficiency of the combined cycle (63.498%) is the highest.

By directing the analysis to the work of the compressors, it is possible to realize that more energy is required to activate this equipment in the propane cycle than in the CO_2 cycle. Such statement may seen initially odd, since, analyzing Table 4, the elevation of pressure in the R744 cycle is greater than in the R290 cycle, but it is possible to note that the variation of the enthalpy to the high temperature stage is higher than the low temperature stage, as a consequence of the fact that the compressor of propane needs to raise the pressure in a higher temperature range. That generates an increase in the specific enthalpy ($h = u + P\,v$, which u is the internal energy, P is pressure and v is the specific volume, according to Çengel [6]). In addition, the R290 cycle has a higher mass flow than the R744 cycle. These two causes combined promote an elevation in the work of R290 compressor.

Regarding the pump, its work is small compared to others (0.113 kW). This is cause due to the fact that the pressure difference in the pump is reduced, because it serves only to make the brine circulates in the system and overcome the pressure loss. Another factor that contributes to this situation is explained by Moran [7], which said that for the same pressure rise, a compressor would require a much greater work input per unit of mass flow than the pump because the average specific volume of the gas flowing through the compressor would be many times greater than that of the liquid passing through the pump.

Hereinafter, the operating conditions of the combined system will be changed to study the cycle behavior when the operating circumstances are varied in the evaporators, heat exchangers and condensers. The analyses will be done through graphs that illustrate the behavior of the system under certain conditions of operation. Moreover, it is important to emphasize that equipment such as the condenser has a negative heat transfer value, indicating the output of the system energy.

The variables used were: the thermal load of the Evap. R744, the surroundings temperature of the Cond. R290, the thermal load of the LT. HE. Brine, the saturation temperature of Evap. 1. R290, Evap. 2. R290 and Cond. R744. Therefore, the graphs will represent the operation of the system when one of these characteristics is varying while the others remain constant. Thus, the values of this data when they are not being the focus of the graphic analysis, in other words, when they are not changing, are indicated in Table 6. It is important to emphasize that they are practically the same as the initial data in Table 3, except for the surroundings temperature, because it was desired to study the system when it is applied in a location where this condition is extreme.

Table 6. Values of project characteristics

Variable:	**Value applied:**
Evap. R744 Thermal Load	10 kW
Cond. R290 Ambient temperature	42 °C
Thermal load of LT. HE. Brine	21 kW
Saturation temperature of Evap. 1. R290	-14 °C
Saturation temperature of Evap. 2. R290	-14 °C
Saturation temperature of Cond. R744	-10 °C

The thermal load in the evaporator of CO_2 is modified depending on the operating conditions. According to ASHRAE [10], a large peak of thermal load can be generated due to door openings, the frequency of loading food and the manufacture of ice. Furthermore, the season of the year also promotes significant changes in the thermal capacity of a refrigeration system. In this context, the behavior of the system will be illustrated when the thermal load of the R744 evaporator varies between 5 kW and 23 kW, which the normal operating condition is 10 kW.

The variation of the thermal load influences the R744 cycle and the R290. That being said, the thermal load generates direct impacts on the functioning of the equipments contained in this cycle, as illustrated in Fig. 7. Moreover, according to the model implemented, it is through the energy balance of the Evap. R744 (Table 2) that the mass flow of the CO_2 cycle is determinated. Thus, observing Fig. 8, it is perceived that the elevation of the cooling capacity promotes the increase of the mass flow, hence it occurs an elevating of the compressor work of the low temperature cycle (Fig. 7), which the minimum value is 0.5692 kW and maximum of 2.618 kW. Consequently, there is a greater rejection of heat in the condenser of R744, in order to attend the First Law and its variation can reach the lower value of -5.569 kW and the superior number of -25.618 kW (Fig. 7).

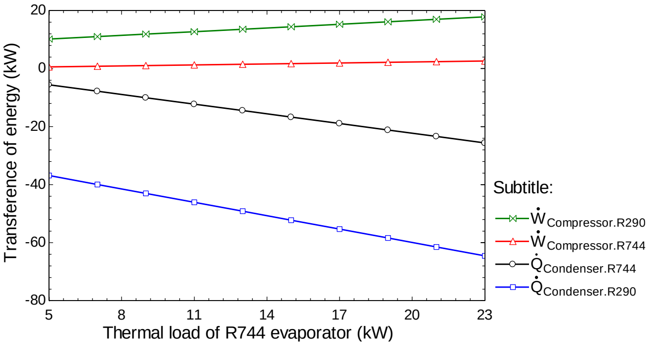

Figure 7. Relation between the thermal load of Evap. R744 and the energy transference

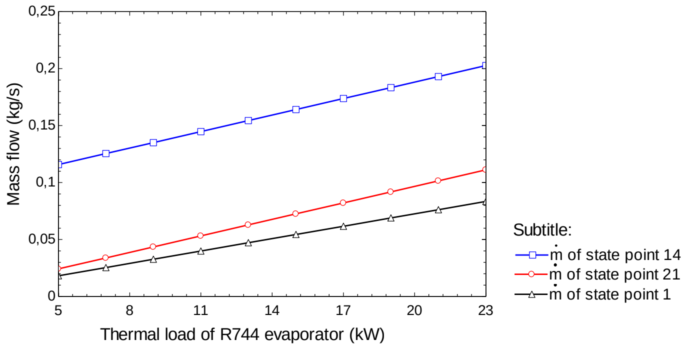

Figure 8. Relation between the thermal load of Evap. R744 and the mass flow

It is through the energy balance in the CO_2-propane heat exchanger that the flow that runs through the propane side is determinated. Then, due to the need to absorb more heat, the mass flow on the propane side increases (Fig. 8). Consequently, the work in the compressor of this cycle rises from 10.17 kW to 17.82 kW (Fig. 7), this will also cause an increase in the removed heat at the condenser of R290 from -36.86 kW to -64.55 kW (Fig. 7), in order to assist the increase of the thermal demand of the system.

The influence of the thermal load extends to the COP values of the cycles through the analysis discussed in Fig. 9 and Fig. 10. It is possible to note that the thermal load only generates impacts on the COP of the combined cycle. The explanation stems from the fact that mass flow in the R744 and R290 cycles is a proportional value of the thermal load elevation, thus the work increases in the same proportion that the heat is absorbed in the evaporator, generating an unchanged COP. However, the combined effect of the sum of the work of the compressors increases higher than the thermal load, then the consequence of this is the decline of the COP, which starts at a maximum value of 2.395, a sharp decline occurs, but tends to smaller variations, until a minimum of 2.141. For the values of Carnot's COP, it is alleged that they remain constant because the thermal load does not has influence in the temperature of the hot or cold reservoirs.

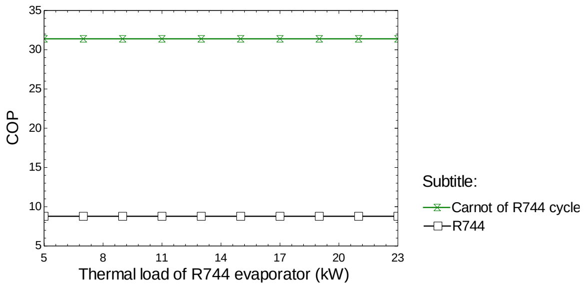

Figure 9. Relation between the thermal load of Evap. R744 and the COP for the CO_2 cycle

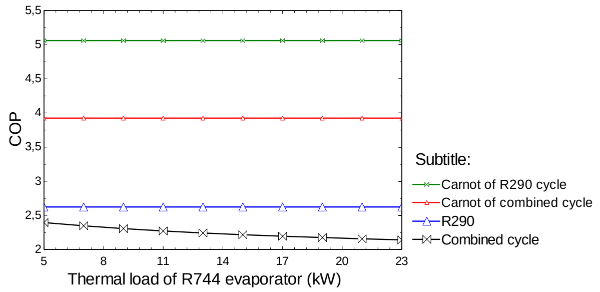

Figure 10. Relation between the thermal load of Evap. R744 and the COP for the R290 cycle and for the combined system

Directing the analysis to the environment where the propane condenser rejects the heat absorbed in the evaporators (Evap. 1. R290 and Evap. 2. R290), it is perceived that the surroundings temperature depends on the day, month, season of the year and even the place where the cooling system will be installed. Thus, evaluate the system under this condition is important, in order to verify if the system serves for the requirements of several locations. The consequences of this variation influence in the saturation temperature of the condenser, according to Çengel [6], in order to have a heat transfer at a reasonable rate, a temperature difference of 5 °C to 10 °C should be maintained between the refrigerant and the medium with which it is exchanging heat, in this paper, it will be use a condensation temperature 10 °C higher than the environmental temperature. It is valid to observe that under normal operating conditions the temperature is 25 °C, but this study will use a variation between 20 °C and 42 °C. In this way, the saturation temperature of the condenser is responsible for the determination of the upper pressure of the cycle. Figure 11 shows that the work of the compressor increases (from 6.84 kW to 12.3 kW), due to the need of elevate the fluid to a higher saturation pressure, hence the rejected heat increases, starting from an initial value of -39.09 kW until -44.55 kW.

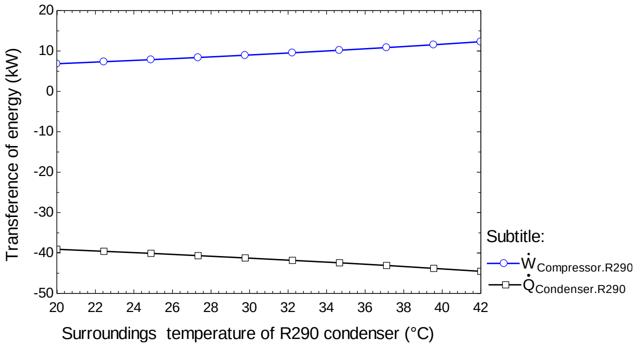

Figure 11. Relation between the ambient temperature of the Cond. R290 and the energy transference

The alteration of the environment temperature promotes a variation of the saturation pressure in the condenser of the propane cycle, which generates a change in all properties of the state points that are located on the constant pressure line of the propane condenser, in other words, changes of properties occur at state points 16, 17, 18, 20 and 21. The variation of the properties of the cycle, mainly of the enthalpy, pressure and temperature, causes modifications in the energy balances of the intermediaries heat exchangers, which promotes a mass flow variation, as noted in Fig. 12. This is caused due to the fact that the increase of the condensation pressure influences in the exit of the expansion valve, in which the refrigerant will have a higher quality after the process of throttling (points 12 and 22 in the propane cycle), as illustrated in Fig. 13, that exhibits two levels of condenser pressure, the smallest corresponds to the ambient temperature of 25 °C and the highest has the ambient temperature at 42 °C. Thereby, according to the energy balance for evaporators illustrated in Table 2, the heat entering is constant and the variation of enthalpy decreases, hence, in order to compensate, the mass flow of this component should increase (Fig. 12).

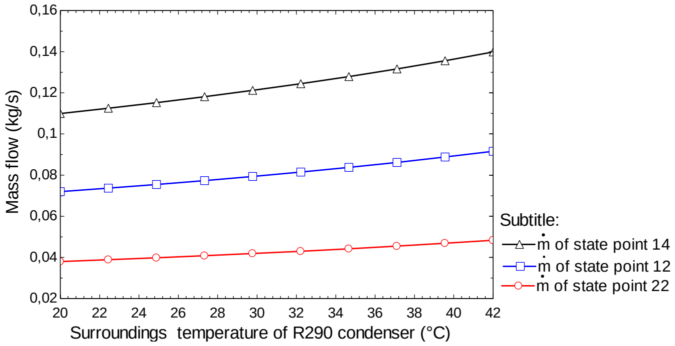

Figure 12. Relation between the ambient temperature of the Cond. R290 and the mass flow

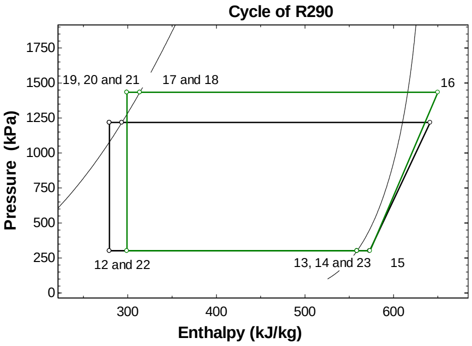

Figure 13. Diagram P-h of the R290 cycle for two values of condenser pressure

The temperature variation of the hot reservoir (temperature of the external environment), promotes alterations in the COP of Carnot and in the real COP, which this analysis is illustrated in Fig. 14. Regarding the efficiency of Carnot refrigeration cycles, it is perceived that the increase in the ambient temperature causes an elevation in the denominator of Eq. (4), which consequently reduces the COP of the reversible cycle. Directing the look at the real COP, the work of the compressor increases, as a consequence of the need to apply in the fluid a higher pressure, however, the heat that enters the evaporators remains constant, this causes a reduction of the real COP.

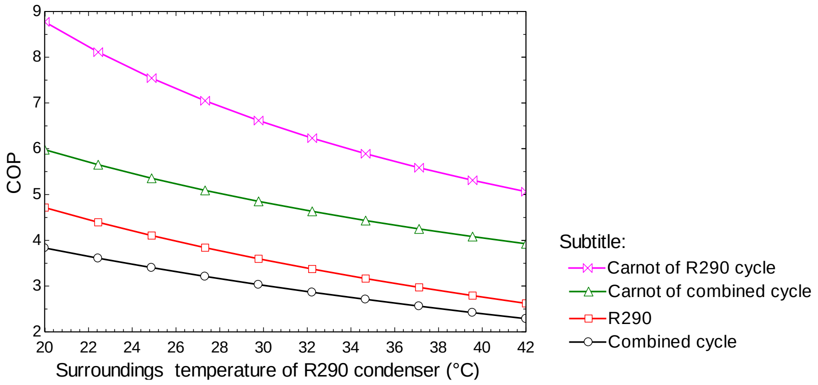

Figure 14. Relation between the ambient temperature of the Cond. R290 and the COP

The 40% ethylene glycol system also generates impacts on the system operation. Under design conditions, the low temperature heat exchanger receives a thermal load of 21 kW to refrigerate products at temperature between -1 °C and 2 °C. Thus, Fig. 15 illustrates the operation of the system when the thermal load of LT. HE. Brine varies between 10 kW and 28 kW. From the analysis of Fig. 15, a similar behavior was observed when it was studied the increase of the thermal load of the R744 evaporator (Fig. 7), in which the work of the compressor, the heat rejected in the high temperature heat exchanger and in the propane condenser enhance. In addition, the work in the pump increases discretely between 0.1128 kW and 0.1129 kW with the increase of the thermal load. Therefore, it is important to emphasize that the value of $\dot{Q}_{\text{Condenser R290}}$ varies between -29.35 kW and -54.22 kW; in relation to $\dot{Q}_{\text{High temperature heat exchanger}}$, the minimum heat rejection value is -10.11 kW and the maximum is -28.11 kW; lastly, the maximum work of Comp. R290 is 14.970 kW and the minimum is 8.104 kW.

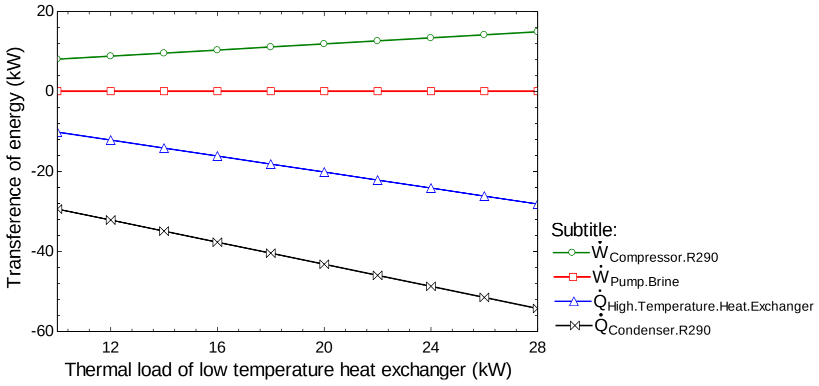

Figure 15. Relation the thermal load of LT. HE. Brine and the energy transference

The reason why the brine cycle influences in the high temperature cycle stems from the fact that the flow of the side responsible for absorbing the heat from the 40% ethylene glycol cycle is determined by the energy balance of the ethylene glycol 40%-propane heat exchanger. Thus, if there is a higher demand of heat rejection on the brine side, it will require a higher mass flow on the side of the R290 in order to absorb the load, as analyzed in Fig. 16. It is important to emphasize that the value of the mass flow of the brine cycle was withdrawn due to the fact that it is a constant and function only of the volumetric flow determined by the pump manufacturer used in the system.

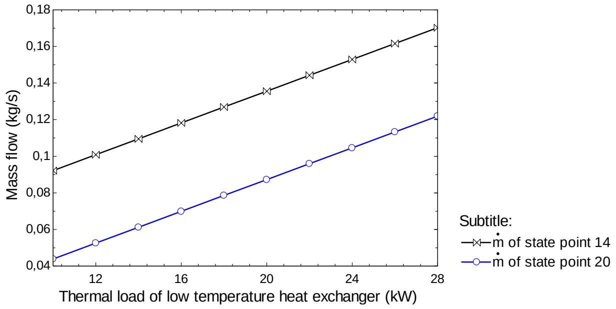

Figure 16. Relation between the thermal load of LT. HE. Brine and mass flow

The Carnot's COP of the R290 and the combined cycle, in addition to the real COP of the R290 cycle, do not undergo major changes with the thermal load variation of the low temperature heat exchanger, as analyzed in Fig. 17. However, a different behavior than the generated when the thermal load of the R744 cycle increased is observed, because the COP of the combined cycle increases to a certain point where it becomes approximately constant. The explanation of this behavior is cause due to the fact that the elevation of the thermal load is higher than the speed of the elevation of work, since the contribution of the pump is small, consequently, the thermal load increases in a proportion higher than the elevation of the work, promoting an increase of the real COP, which starts with the value 2.138 until a maximum of 2.343.

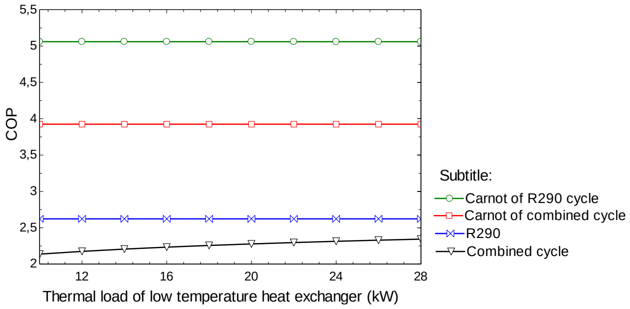

Figure 17. Relation between the thermal load of LT. HE. Brine and the COP

Subsequently, an analysis that involves changing the saturation temperature of the evaporators of the propane from -18 °C to -12 °C will be performed. In Fig. 18, the increased saturation temperature of the evaporator promotes a decrease in both compressors work and heat rejection in the condenser of propane. This happens due to the fact that the lower pressure of the cycle is determined by the saturation temperature in the evaporators. If this temperature increases, it means that the compressor will work in a lower difference of pressure, consequently the variation of the enthalpy will be reduced and also the value of the work, which has an initial value of 13.56 kW and the last of 11.7 kW. In order to obey the First Law of Thermodynamics, the rejected heat in the condenser is reduced as well from -45.81 to -43.95 kW.

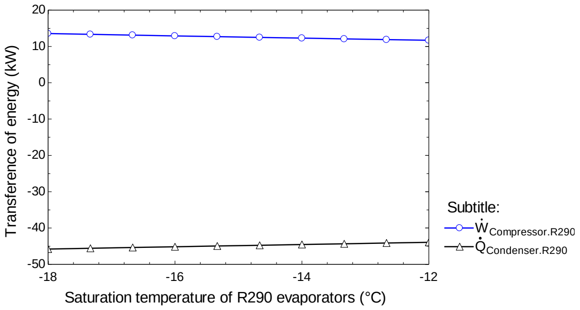

Figure 18. Relation between the saturation temperature of the Evap. 1. R290 and Evap. 2. R290 and the energy transference

On the other hand, there is a reduction in mass flow of the propane cycle (Fig. 19). Figure 20 and Figure 21 show the propane cycle for two evaporations temperature levels, -14 °C (normal operating conditions) and -12 °C. Thus, the lower and upper pressure are closer when the R290 evaporators function at -12 °C, then the enthalpy of the saturated vapor at the exit of the Evap. 1. R290 and Evap. 2. R290 increases, which also generates an increase in the variation of enthalpy, because the input enthalpy remains constant. In order to compensate this situation, since according to the energy balance of an evaporator (Table 2), the inlet heat is constant and the enthalpy variation increases, so the mass flow should reduce.

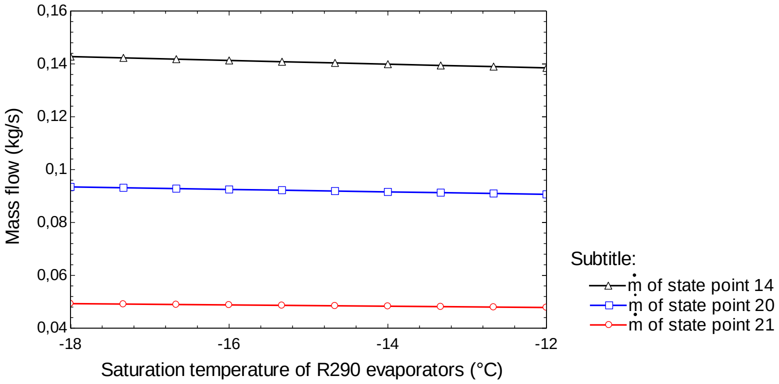

Figure 19. Relation between the saturation temperature of the Evap. 1. R290 and Evap. 2. R290 and the mass flow

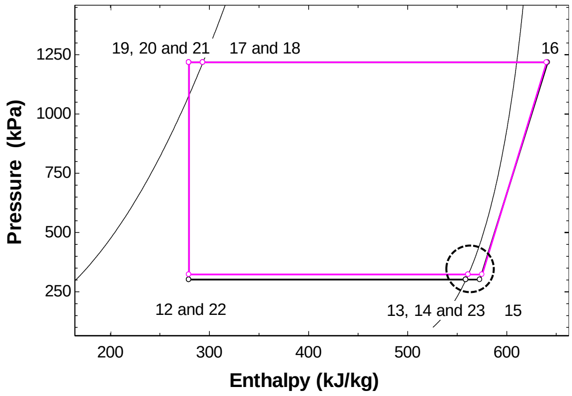

Figure 20. Diagram P-h of the R290 cycle for two evaporator pressure values

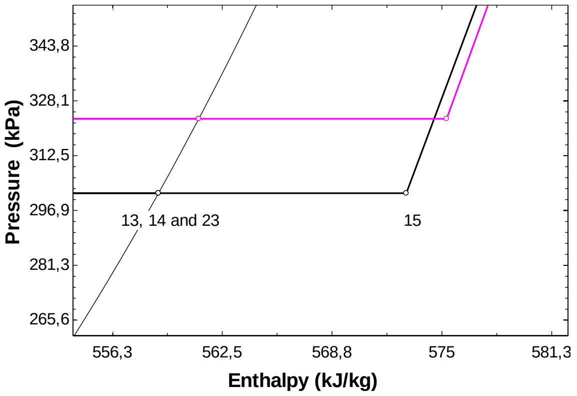

Figure 21. Diagram P-h of the R290 cycle for two evaporator pressure values enlarged in the area inside the dashed line

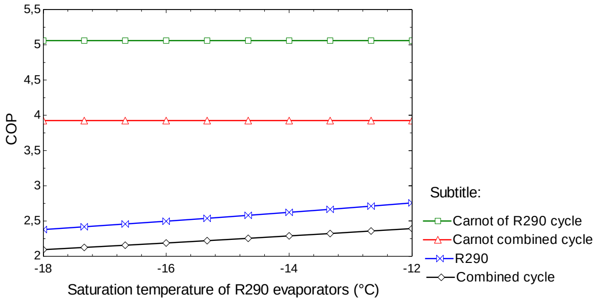

Figure 22. Relation between the saturation temperature of the Evap. 1. R290 and Evap. 2. R290 and the COP of R290 cycle and of the combined cycle

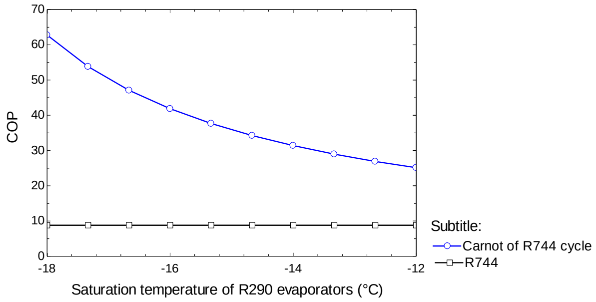

Figure 23. Relation between the saturation temperature of the Evap. 1. R290 and Evap. 2. R290 and the COP of R744 cycle

Furthermore, Carnot's COP variation will occur, because the saturation temperature of the propane evaporators corresponds to the temperature of the hot reservoir of the R744 cycle. Thus, through the Fig. 22 and Fig. 23, it is perceived that the Carnot's COP of the CO_2 cycle reduces from 62.79 to 25.12, due to the increase in the denominator of Eq. (4), but the real COP of the CO_2 cycle does not vary, since it does not depend on this temperature. For the R290 cycle and for the combined system, the increase of this temperature allows an increase in the real COP, because the work decreases and the thermal load of the evaporators remains constant.

The last analysis that will be performed refers to the variation of the saturation temperature of the R744 condenser between -12 °C e -5 °C, which operates under normal conditions at -10 °C. Starting with the energy transfer illustrated in Fig. 24, it is perceived that this variable generates little effect on the system. What can be observed is an increase in the work of both compressors, from 1.019 kW to 1.454 kW in Comp. R744 and from 12.25 kW to 12.42 kW in Comp. R290. Moreover, there is an increase in heat rejected by the condensers, from -11.02 kW to -11.45 kW in Cond. R744 and from -44.39 kW to -44.99 kW in Cond. R290, as a consequence of increased compressor work, in order to comply with the Energy Conservation Principle.

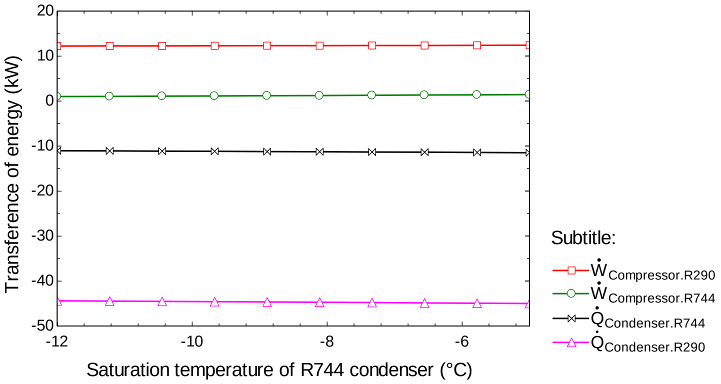

Figure 24. Relation between the Cond. R744 saturation temperature and the energy transference

The explanation for the increase in the work of the R744 compressor derives once again from the distance of the pressure levels. It occurs that with the increase of the condensing temperature, the upper pressure of the cycle rises, then a greater work is needed to make the fluid reach the desired properties. On the other hand, the increase in the work of the R290 compressor is only function of the increase of mass flow (Fig. 25), a value generated in the energy balance of the heat exchanger CO_2-propane.

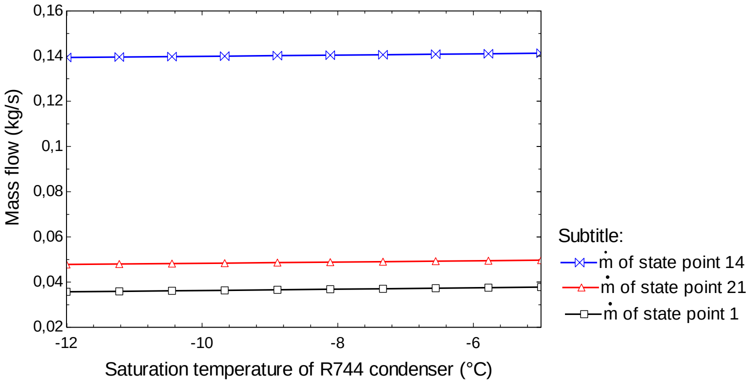

Figure 25. Relation between the Cond. R744 saturation temperature and the mass flow

The mass flow (Fig. 25) of the R744 cycle and of the side of the propane that is connected with the CO_2 through the heat exchanger both increase. Firstly, it occurs that with the increase of this temperature, the saturation pressure of the high pressure side of the R744 cycle will enhance, hence the fluid after the expansion valve will have an higher amount of vapor, and the variation of enthalpy to reach the state of saturated vapor will be lower. This observation can be seen it in Fig. 26 that show the position of the

R744 cycle state point for two saturation pressure, which the lower corresponds to the pressure when the condensation temperature is -10 °C (normal conditions) and the higher refers to the pressure when the condenser has a saturation temperature of -5 °C. Thus, to compensate for these situations, the mass flow of the CO_2 cycle should be increased, accordingly to the condenser balance of energy (Table 2). Moreover, with the fact that there is an increase in the heat rejected in the condenser of R744, it is necessary a greater flow from the side of the propane to absorb this thermal load.

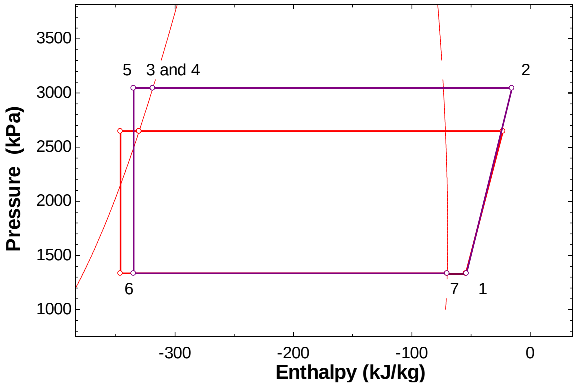

Figure 26. Diagram P-h of the R744 cycle for two condenser pressure values

In relation to the COP, Fig. 27 shows that as the temperature of the condenser of R744 corresponds to the temperature of the cold reservoir of the R290 cycle, thus its increase, according to Eq. (4), promotes an elevation of the COP of Carnot for this cycle from 4.836 to 5.705. On the other hand, the increase in the work in the compressors generates a drop in the real COP in the R744 cycle from 9.809 to 6.877 and in the combined analysis from 2.316 to 2.217, as observed in the Eq. (3).

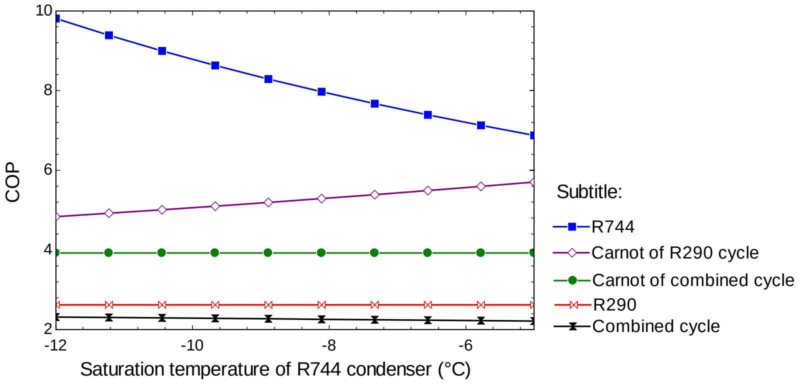

Figure 27. Relation between Cond. R744 saturation temperature and COP

5 Conclusion

In this paper, it was studied a combined refrigeration system, that is compound by two refrigeration cycles of vapor compression, one of them has R744 (CO_2) as working fluid, the other one functions with R290 (propane) and the cascade cycle also has a system with 40% ethylene glycol (brine).

For the modeling and simulation of the combined refrigeration system, the domain in study was discretized in 18 controls volumes and 23 state points. The mass and energy balances were applied in each control volume with the consideration of steady state. The system of equation obtained was solved using the Newton-Raphson method.

Taking into consideration the initial condition that was shown in Table 3, the following relevant results were obtained: the heat transfer of the counterflow heat exchangers was 21.113 kW for the heat exchanger between the 40% ethylene glycol and R290, and was 11.138 kW for the heat exchanger between the R744 and R290; the compression supplied work was 1.138 kW for the compressor of the

R744 cycle and 7.885 kW for the compressor of the R290 cycle. The supplied work of the brine pump was 0.113 kW. The rejected heat to the environmental was 40.136 kW in the R290 condenser. Through the energy analyses, the Coefficient of Performance (COP) obtained was 8.784 for the R744 cycle, 4.090 for the R290 and the result of the combined cycle COP was 3.393. The greatest efficiency was 63.498% of the combined cycle.

The first analyze was the variation of the thermal load in the R744 evaporator between the values of 5 kW and 23 kW. For this study, the result was: the rejected heat in the condenser of R744 and the supplied work of the compressor both increased. This caused an elevation in the mass flow of propane cycle in the R744 side, consequently, the work of the compressor and the heat transfer in the condenser of the R290 cycle both increased as well. Regarding the COP, the only one that had variation was the COP of the combined cycle, which decreased with the elevation of thermal load.

By studying the variation of ambient temperature between 20 °C and 42 °C and maintaining 10 °C difference between the ambient temperature and the condensing temperature of the R290 cycle, the following conclusions were reached: the heat rejected in the condenser varied in absolute values from 39.09 kW to 44.55 kW; mass flow at state points 12 and 22, inlets of R290 cycle evaporators, enhanced with the increase of ambient temperature; Carnot COP and real COP of the combined refrigeration cycle and R290 cycle both decreased with the increase of ambient temperature.

With the study of the thermal load variation of brine system in a range from 10 kW to 28 kW, it was concluded that: the heat transfer in the R290 cycle condenser and the supplied work to the R290 cycle compressor both increased; the supplied work to the brine pump enhanced; the heat transfer in the high temperature heat exchanger of brine cycle increased with elevation of thermal load; the inlet mass flow at the counterflow heat exchanger between brine and R290, more precisely on the R290 side, increased; the Carnot COP of R290 and combined cycle, in addition to the actual R290 cycle COP, did not change significantly with the thermal load variation of the low temperature heat exchanger, but, the combined cycle real COP increased with the elevation of brine cycle thermal load.

By studying the variation of the evaporation temperature of the evaporator of the R290 cycle, varying between -18 °C and -12 °C, the following conclusions could be highlighted: the heat transference in the R290 condenser and the work of the compressor decreased; the real COP of R744 cycle remained constant, but the Carnot COP of this cycle decreased, the real COP of R290 cycle increased, and, consequently, there was an increase in the COP of combined cycle with the elevation of evaporation temperature of Evap. 1. R290 and Evap. 2. R290.

Finally, the study of the variation of condensation temperature of the R744 cycle between -12 °C to -5 °C came to the following conclusions: this variable has little effect in the variation of energy transference; the mass flow slighted increased; the R744 cycle real COP decreased, while the R290 cycle real COP remained constant and the combined cycle COP had a slight reduction with the increase of condensing temperature.

Acknowledgements

The authors would like to thank the Federal University of Sergipe and the institutional program of Scientific Initiation Scholarship (PIBIC) for supporting the research reported in this article. The first author would like to thank her advisor at the Federal University of Sergipe, the second author, who, through his knowledge and experience, promoted great assistance in conducting this research.

References

[1] J. M Calm. The next generation of refrigerants–Historical review, considerations, and outlook. International Journal of Refrigeration, 2008.
[2] G. S. Pereira. Análise de Sistemas de Refrigeração a CO2 em supermercados. Dissertation of Mastering of Mechanical Engineering, Universidade Federal de Pernambuco, 2010.
[3] I. M. Araújo, J. J. P. de Araújo, L. M. Santos and L. C. Sobral. ANÁLISE ENERGÉTICA DE UM

SISTEMA DE REFRIGERAÇÃO DE DOIS ESTÁGIOS COM CO2 (R-744) E NH3 (R-717). X National Congress of Mechanical Engineering (CONEM), 2018.

[4] S. Sawalha. Carbon Dioxide in Supermarket Refrigeration. Doctoral Thesis, Royal Institute of Technology, 2008.

[5] S. Johansson. Evaluation of CO2 supermarket refrigeration systems. Master of Science Thesis, KTH School of Industrial Engineering and Management, 2009.

[6] Y. A. Çengel and M. A. Boles. Termodinâmica. Mc-Graw Hill, São Paulo, 7a ed., 2013.

[7] M. J. Moran, H. N. Shapiro and D. D. Boettner. Fundamentals of Engineering Thermodynamics. John Wiley & Sons, 7a ed., 2011.

[8] M. Kanoglu, Y. A. Çengel and I. Dinçer. Efficiency Evaluation of Energy Systems. Springer Science & Business Media, 2012.

[9] P. Bertelsen and K. Christensen. Use of natural refrigerant in supermarkets. Project Report, Danish Technological Institute, 2003.

[10] American Society of Heating, Refrigerating and Air-Conditioning Engineers. ASHRAE Handbook Refrigeration. Atlanta, Ga, American Society of Heating, Refrigerating and Air Conditioning Engineers, 2018.