ANALYSIS OF A DIRECT EXPANTION AIR CONDITIONING SYSTEM USING AN ICE THERMAL STORAGE TANK FOR REDUCTION OF THE ELECTRICITY DEMAND DURING PEAK TIME PERIODS

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Abstract. *This work presents results of computational simulations performed for a new air conditioning system of the type "split" launched in the market. This system has a direct expansion evaporator and an ice thermal storage tank. This type of system has been recommended by its manufacturer for use in commercial buildings that present load profiles coincident with the electricity utility peak time period. The main advantage of this system is to promote the displacement of the electricity consumption for the off peak time periods of the electricity companies. This strategy contributes to increase the electricity load factor postponing the high investments required to expand the electricity generation and distribution capacity of these companies. Further the commercial buildings can take advantage of the lower electricity rates for the off peak time periods. It was analyzed the coefficients of performance "COP" for the daily and night periods, as well as, the global daily efficiency of this new system. Also a conventional split system (without ice storage tank) was simulated in order to compare the "COP" values. For the new split system the "COP" increases during the peak time period due to the sub-cooling of the liquid. On the other hand the "COP" decreases during the off time period due to the lower evaporating temperatures required to make the ice in the tank. The global daily efficiency obtained for this system is lower than that obtained for a typical conventional direct expansion system, therefore, this new split system is able to promote only reduction of the electricity demand (kW).*

Keywords*: ice, thermal, storage, air, conditioning*

1. INTRODUCTION

Brazilian commercial buildings are responsible for about 13% of the total electricity consumption of the country which on average 20% is due to the air conditioning systems, including the commercial and public sectors, according SEBRAE (2002). Beyond that in commercial offices buildings and some commercial establishments (e.g., shopping centers, supermarket) the air conditioning electricity consumption can reach up to 50% of the monthly electricity bill, according SEBRAE (2002). Indirect expansion central air conditioning systems using chilled water are applied generally in large commercial buildings (e.g., shopping centers, supermarket, hotels) while expansion direct systems (e.g., split, multi split, self contained and roof top systems) are used more frequently in small and medium commercial buildings (e.g., multi story office buildings, branches of banks), (McQuiston et al., 2000) e (ASHRAE, 2002).

Energy storage systems have been used in Brazil mainly in large commercial buildings such as shopping centers and hypermarkets that use air conditioning central systems. The peak load of these buildings occurs at night from 5 p.m to 9 p.m., and is coincident with the peak load period experimented by the electricity utility companies due mainly to the electrical domestic showers. Thermal energy storage became attractive for these buildings because both the electricity demand (kW) cost and the energy consumption cost (kWh) are higher during the peak period than during the off peak period of the electricity utilities (Dorgan e Elleson, 1994). Energy storage systems can contribute to postpone the high investments necessary by the utilities in order to attend short daily periods of time. Also the postponing of constructions of new hydroelectric power plants can contribute to the reduction of the social and environmental impact caused by the necessity to flood large fertile areas as well as the displacement of entire communities from their living places.

Therefore, as already occurs in Europe, United States of America and Japan, in the near future Brazil can have a second electricity peak period at late afternoon caused by office buildings air conditioning systems. The perspective of growth of the Brazilian economy during the next years will speed up the construction of new commercial office buildings that will require air conditioning systems to attend the comfort levels of their occupants. According Pinheiro, (2006) is expected an increase of the use of direct expansion systems in the near future for this type of building since the price of the equipment is dropping and the investment is lower compared to indirect air conditioning system. Therefore, the use of split energy storage air conditioning systems can be a good alternative to reduce operational costs of the buildings through the shifting of the energy consumption to the late night period.

A recent air conditioning system has been launched in the Brazilian market. This equipment has a direct expansion split system incorporated to an ice storage tank. The goal of this work is to present results of thermal analyses obtained from computational simulations for this equipment in order to verify its technical viability.

2. DESCRIPTION OF. THE SYSTEM

The air conditioning system analyzed in this work uses a non- azeotropic bend named “R-407-C” wish is composed by the refrigerants, type HFC, R-32, R-125, R-134a, corresponding to 23% ,25% and 52% in mass proportion, respectively. The non-azeotropic blends have a different behavior compared to the azeotropic blends during the evaporation and condensation process. The non-azeotropic blends are characterized by the increase in their temperatures during the isobaric phase change from liquid to vapor. During the condensation process the non azeotropic blends are characterized by the decrease in their temperatures, (Stoecker, 2002). The system has two main components: (i) a split air conditioning composed by an evaporator unit installed indoors, inside the conditioned space, and a condenser unit installed outdoor; (ii) an ice tank responsible to storage latent energy. The system works in two different modes: (i) the energy storage mode, during the period of night, when the ice tank is charged corresponding to the “ice making period”; (ii) the discharging mode, during the day, when the ice tank is discharged corresponding to the “ice melting period”. Figure 1 shows the refrigeration circuit referent to the charging mode, “ice formation period”

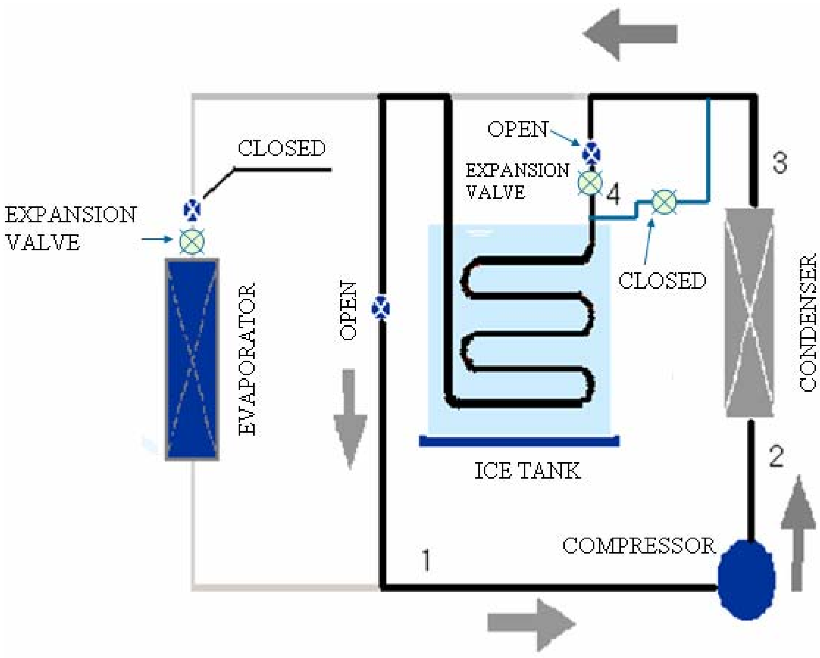

Figure 1. Refrigerant circuit during the “ice formation period”.

The ice tank is composed by an internal copper coil where the refrigerant flows during the phase change process. Around the coil there is still water that changes phase from liquid to solid and vice versa. During the charging mode the tank is initially full of liquid water. As the refrigerant flows through the coil the solidification process starts because the evaporating temperature of the refrigerant around -5ºC is lower than the freezing point of the water 0ºC. At the end of the charging mode the water is in the solid phase. The cycle corresponding to the charging mode is “1-2-3-4-1” as can be seen in figure 1.

During the discharging mode the refrigerant circulates thought the coil at a temperature higher than the freezing point (around 40 ºC) melting the ice. The ice tank is of the type “internal melting” which means that the melting occurs outward in the radial direction, initiating in the thin layer of ice located on the external pipe wall surface until the radius of the external ice layer surface. Figure 2 shows the refrigeration circuit referent to the discharging mode, “ice melting period”, where can be seen the cycle “1- 2-3´-4´´-1”.

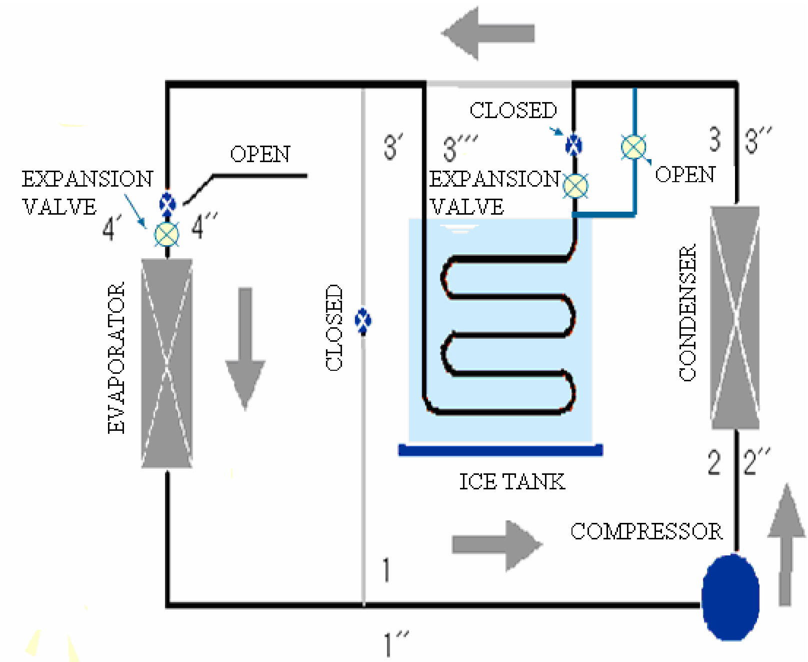

Figure 2. Refrigerant circuit during the “ice meting period”.

3. ANALYSIS AND RESULTS

Figure 3 shows the diagram “Pressure x Enthalpy related to figure 2 of refrigeration cycles used in direct expansion air conditioning systems (e.g., multi-split, split and roof top) considering azeotropic blend. The cycle “1-2-3-4´-1” refer to the conventional cycle that does not have the ice tank, while the cycle “1-2-3-3´-4´-1” refer to same system, however, with the addition of the ice tank. In the cycle “1-2-3-3´-4´-1” refrigerant liquid leaving the condenser flows through the ice tank coil where it is sub cooled due to the meting of the ice.

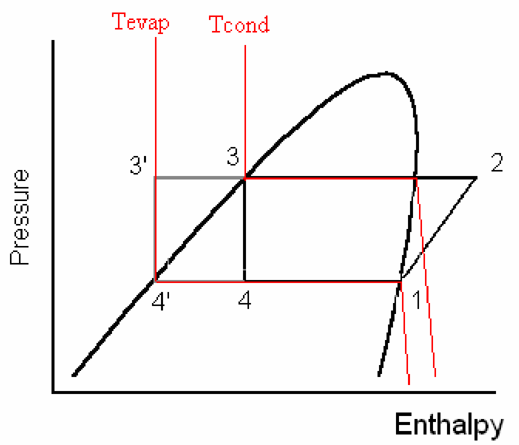

Figure 3. Diagram pressure x enthalpy of. the conventional cycle and the energy storage cycle.

The cycle “1-2-3-4-1” is composed by the following processes: (i) process “1-2”, refer to the adiabatic compression in the compressor; (ii) process “2-3” refer to the condensation, considering the refrigerant state liquid saturated at the outlet of the condenser; (iii) process “3-4” refer to the adiabatic expansion in the expansion valve; process “4-1” refer to the evaporation in the evaporator, considering the refrigerant state saturated vapor at the outlet of the evaporator. The cycle "1-2-3-3´-4´-1” is composed by the following processes: (i) process “1-2” refer to the adiabatic compression in the compressor; (ii) process “2-3” refer to the condensation, considering the refrigerant state liquid saturated at the outlet of the condenser; (iii) process “3-3´ ” refer to the sub cooling of the liquid due to its circulation through the ice tank coil releasing heat to melt the ice; process 3´- 4´ ” refer to the adiabatic expansion in the expansion valve; process “4´ -1” refer to the evaporation in the evaporator, considering the refrigerant state saturated vapor at the outlet of the evaporator.

Analyzing figure 3 it can be observed that cycle “1-2-3-3´-4´ -1” is more efficient than cycle “1-2-3-4-1” because it shows lower average temperature during the rejection heat transfer process (2-3´). The sub cooling also provides reduction of refrigerant mass flow because the evaporator enthalpy difference (h1 - h4´) is higher than (h1 - h4). This carries out a decrease in the compressor power and consequently decreases the COP value.

3.1 Pressure Analises

The effect of the evaporating pressure in the performance of the cycle is analyzed in this section for a non-azeotropic blend “R-407C”. Analyzing the thermodynamic tables of R-407C it can be figured out that for most of the evaporating pressures the evaporating temperature increases approximately 5ºC during the phase change from liquid to vapor. On the other hand, the condensing temperature decreases approximately of the same magnitude. This fact is important since it offers the opportunity to increase the evaporating pressure in order to increase the coefficient of performance “COP”.

For example, we will compare two refrigerating cycles: one is conventional and operates with an azeotropic blend while the other is non conventional, having an ice tank incorporated to the system to sub cool the liquid, and operating with a non-azeotropic blend. It has been assumed that both systems must work at an evaporating temperature of 10ºC and that the evaporator pressure drop is small and can be neglected. For the conventional system the evaporating temperature does not change and is 10ºC along the evaporator. For the non conventional system, the evaporating temperature increases during the phase change process, therefore, the ice tank needs to sub cool the liquid to 5ºC in order the blend at the outlet of the evaporator reach 10ºC. This represents an average evaporating temperature of the approximately 7.5ºC and, consequently a lower “COP” value compared to the conventional system. However, the “COP” of this non conventional system can be improved increasing the blend temperature entering the evaporator to 7.5ºC. This will guarantee a blend temperature at outlet of evaporator of 12.5ºC and, consequently, an average evaporator temperature of 10ºC, which will be the same of the conventional system. Figure 4 shows a simplified diagram “P x h” with the conventional cycle “1-2-3-4” and the non conventional cycle “A-B-C-D-A”.

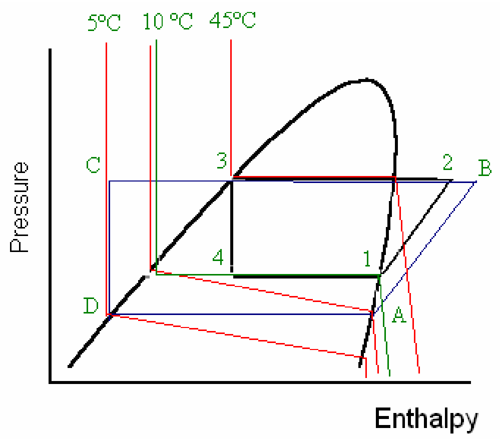

Figure 4. Diagram pressure x enthalpy of the conventional and non conventional cycles.

Analyzing figure 4, for the non conventional system it can be suggested an increasing in the evaporating pressure in order to promote an increase in the average evaporating temperature with a consequent decrease in the compression ratio and increase in the “COP” value.

Similar analysis can be done to verify the effect of the condensing pressures in the conventional and non conventional cycles. The condensing temperature of the conventional system is fixed at 45ºC. For the non conventional system the blend needs to enter the condenser at 50ºC in order to go out at 45ºC. To keep an average condensing temperature of 45ºC the inlet temperature can be reduced to 47.5 ºC which implies in a leaving temperature of 42.5ºC Therefore, it can be suggested a decreasing in the condensing pressure in order to promote an decreasing in the average condensing temperature with a consequent reduction of the compression ratio and increase in the “COP” value.

Figure 5 shows the conventional cycle “1-2-3-4-1” and the non conventional cycle “1”-2”-3”-3”’-4”-1”, also shown in figure 2, which combines the effect of the increasing of the evaporating pressure and decreasing of the condensing pressure. These two effects contribute to the decreasing of the “COP” value.

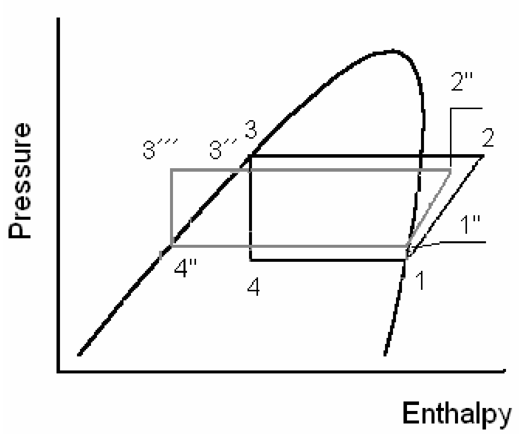

Figure 5. Diagram pressure x enthalpy showing the pressures effect for the non conventional cycles.

3.2. Efficiency Analysis

The coefficient of performance "COP" is given by Eq. (1) and is defined as the ratio between the heat transferred to cool and dehumidify the air entering the conditioned space "Qe" and the mechanical power required by the compressor "W".

$$COP = \frac{\dot{Q}_e}{\dot{W}} \quad (1)$$

The global daily coefficient of performance is given by Eq. (2) and is defined as the ratio between the total heat transfer transferred during the period of one day and the total mechanical power consumed by the compressor at the same period of time. It is assumed that "Qe" and "W" are constants during their respective periods of time.

$$COP\,daily = \frac{\int_0^{Nd} \dot{Q}_e dt}{\int_0^{Nd} \dot{W}_d dt + \int_0^{Nn} \dot{W}_n dt +} = \frac{\dot{Q}_e \times N_d}{\dot{W}_d \times N_d + \dot{W}_n \times N_n} \quad (2)$$

Where: "COP daily" is the coefficient of performance averaged during the period of one day; "Nd" is the compressor operating hours during the discharging mode (i.e., melting ice period) when the air conditioning system is working to meet the load at day time period; "Nn" is the compressor operating hours at the charging mode (i.e., ice formation period) when the tank is being charged at night; "Wn" is the mechanical compressor power required by the compressor at the charging mode; and the "Wd" is the mechanical power required by the compressor during the discharging mode.

Different computational simulations were performed using the software of refrigeration "Cool Pack". It was made the following assumptions for all cycles: (i) the adiabatic compressor efficiency equal to 65%; (ii) the pressure loss in the suction and discharge compressor lines corresponding to 0.5 K of evaporating temperature drop, according the dimensioning criteria described in (Stoecker, 2002); (iii) the superheating degree equal to 0.5 K; (iv) the sub cooling degree of the refrigerant at the outlet of the condenser equal to 0.5 K; (v) the heat loss of the compressor and the heat gain by the refrigerant at the compressor suction line negligible; (vi) no presence of heat exchanger in the compressor suction line. These simulations allowed analyzing the "COP" for two types of cycles named conventional and non conventional refrigerating cycles. The conventional refrigerating cycles have as primary components: evaporator; compressor; condenser and expansion valve. The non conventional refrigerating cycles have all the primary components of the conventional cycle, however, they also have an ice tank used to store energy and to provide sub cooling of the operating fluid during the discharging mode. Table 1 shows "COP" and other results for the different cycles and different operating conditions at daytime period when the refrigerating systems attend the air conditioning load as follows: Cycle I, conventional cycle using refrigerant R-134A of type HFC (hydro-fluor-carbon); Cycle II, cycle non conventional using refrigerant R-134A; Cycle III, cycle conventional using refrigerant R-22 type HCFC (hydro-chloro-fluor-carbo); Cycle IV, cycle non conventional using R-22; Cycle V, cycle conventional using the R-507A, azeotropic refrigerating blend; Cycle VI, cycle non conventional using the R-507A, azeotropic refrigerating blend; Cycle VII, cycle conventional, using the R-407C, zeotropic blend; Cycle VIII, cycle non conventional using the R-407C, zeotropic blend; Cycle IX, cycle non conventional using the R-407C, zeotropic bend, considering the increase of the evaporating pressure; Cycle X cycle non conventional using the R-407C, zeotropic bend, considering the decrease of the condensing pressure; Cycle XI cycle non conventional using the R-407C, zeotropic bend, considering both the increase of the evaporating pressure and decreasing of the condensing pressure. In Table 1 "COPd" is the coefficient of performance; "QL" is the heat transfer rate in the evaporator; "QH" is the heat transfer rate in the condenser; "QSC" is the heat transfer rate in the ice tank, "Wd" is the mechanical compressor power, "M" is the mass flow rate of the refrigerant or blend, "Tin" and "Tout" are the inlet and outlet temperatures, respectively, in the different components.

Table 1. Results from simulations for different cycles.

Cycle	**REFRIGERANT TYPE**	**COPd**	**EVAPORATOR**			**CONDENSER**			**ICE TANK**		**COMPRESSOR**	
			QL W	**Tin ºC**	**Tout ºC**	**QH W**	**Tin ºC**	**Tout ºC**	**QSC W**	**Tout ºC**	**Wd kW**	**M Kg/s**
I	R-134A HFC	4.378	56	10	10	68.9	45	45	NO TANK		12.8	0.369
II	R-134a HFC	5.588	56	10	10	66.2	45	45	14.3	10	10	0.289
III	R-22 HCFC	4.323	56	10	10	69.1	45	45	NO TANK		13	0.344
IV	R-22 HCFC	5.31	56	10	10	66,7	45	45	12.3	10	10.6	0.28
V	R-507A AZEOTROPIC	3.989	56	10	10	70.4	45	45	NO TANK		14	0.492
VI	R-507A AZEOTROPIC	5.76	56	10	10	66	45	45	20.3	5	9.7	0.341
VII	R-407C ZEOTROPIC	3.527	56	5.1	10	72	49.8	45	NO TANK		15.9	0.351
VIII	R-407C ZEOTROPIC	4.717	56	5	10	68	49.8	45	16.3	5	11.9	0.262
IX	R-407C ZEOTROPIC	5.022	56	7.5	12.5	67.2	49.8	45	15.5	7.5	11.2	0.265
X	R-407C ZEOTROPIC	4.973	56	5	10	67.3	47.5	42.5	15.2	5	11.3	0.262
XI	R-407C ZEOTROPIC	5.315	56	7.5	12.5	66.6	47.5	42.5	14.4	7.5	10.5	0.265

From Tab.1 it can be observed the following:

1. The non conventional cycles show higher COP values compared to their conventional cycles pairs. The ice tank promotes heat transfer from the liquid to the ice, sub cooling of the liquid, and reducing the average temperature of the hot source (condenser and ice storage), with consequent increasing of the "COP" values.
2. For the conventional refrigerating cycles I, III, V e VII the higher "COP" value is obtained for cycle I, (R-134A), even though it is only slightly higher than that obtained for cycle II (R-22). Cycle VII (R-407C) shows the lowest "COP" value due to its lowest average evaporating temperature (7.55ºC) compared to the other conventional cycles (10ºC).
3. For the non conventional refrigerating cycles II, IV, VI, VIII the highest "COP" value is obtained for cycle VI (R-507A). The lowest "COP" value is still obtained for cycle VIII (R-407C) due its lowest average evaporating temperature (7.5 ºC) compared to the other non conventional cycles (10ºC).
4. The cycle VIII (R-407C) has an average evaporating temperature of approximately (7.5ºC) which is lower than the other non conventional cycles (10ºC). Therefore, cycle IX (R-407C) considers the possibility of the increase of the evaporating temperature through the increase of the evaporating pressure. In this case the temperatures at the inlet and outlet of the evaporator are 7.5ºC and 12.5ºC, respectively, which increase the evaporating average temperature to 10ºC. Cycle IX shows a "COP" value higher than cycle VIII, however, it is still lower than those of the other non conventional cycles (II, IV and VI).
5. The cycle VIII (R-407C) also has an average condensing temperature of approximately (47.4ºC) which is higher than the other non conventional cycles (45ºC). Therefore, cycle X (R-407C) considers the possibility of the decrease of the condensing temperature through the decrease of the condensing pressure. In this case the temperatures at the inlet and outlet of the condenser are 47.5ºC and 42.5ºC, respectively, which decreases the condensing temperature to 45ºC. Cycle X shows a "COP" value higher than cycle VIII, however, it is still lower than those of the other non conventional cycles (II, IV, and VI).
6. The cycle XI (R-407C) is a combination of cycles IX and X, which work with higher evaporating pressure and lower condensing pressure, respectively, compared to the cycle VIII. The "COP" value of cycle XI is the

highest of all non conventional cycles operating with R-407C (cycles VIII, IX, X and XI). The “COP” value obtained for cycle XI is smaller than those obtained for cycles II (R-134A) and VI (R-507A), however, it is slighter higher than the one obtained for cycle IV (R-22). Therefore, the use of R-407 C, in non conventional cycles, can be a good alternative only for non conventional refrigerating systems which operate with R-22.

7. Cycles VIII, IX, X and XI which operates with R-407C show values of the refrigerant mass flow rate lower than those of the other cycles. This can be appointed as other possible advantage due to the fact of the possibility of reduction of the compressors size.

For calculation of the “COP daily” is necessary to determine the compressor power “Wn” during the charging mode (ice formation period) as shown in Eq. (2). The compressor power “Wn” can be determined from computer simulations for the charging period during nigh time period when the ice is being formed. For these simulations was necessary to define the evaporating temperature during the ice formation period “Te,n”, the condensing temperature and to determine the heat transfer rate necessary to make the ice “Qe,n”. The evaporating temperature was assumed to be -7ºC, that is a typical evaporating temperature encountered in commercial air conditioning installations using ice thermal energy storage systems. The condensing temperature was assumed equal to 45ºC. The heat transfer rate necessary to make the ice “Qe,n” is the total latent energy storied in the tank during the period of one day divided by the number of hours of operation of the refrigerating system to make the ice and is given by Eq. (3).

$$\dot{Q}_{e,n} = \frac{\int_0^{Nd} \dot{Q}_{SC}\, dt}{N_n} = \frac{\dot{Q}_{SC} \int_0^{Nd} dt}{N_n} \tag{3}$$

Where “QSC” is shown in Tab (1) being defined as the heat transfer rate required to sub cool the liquid that comes from the condenser during the discharging mode (ice melting period) and given by Eq. (4).

$$\dot{Q}_{SC} = \dot{m} \times (h_3'' - h_3''') \tag{4}$$

Where “m” is the refrigerant or blend mass flow rate during the discharging period; h3” and h”’ are the enthalpies of the refrigerant or blend at the inlet and outlet of the ice tank according Figs (2) e (4) shown previously.

Table 2 shows the values “COP daily” obtained for different non conventional cycles using different operating fluids. The “COP daily” were calculated according Eq.(2), which uses values of “Qe,n”, calculated from Eq. (3), and values of “Wn” obtained from simulations, considering a refrigeration capacity “QL” of 56 kW and a discharging and charging periods of 10 hours and 12 hours, respectively. Additionally, Tab. 2 also shows the coefficient of performance at charging mode (ice formation period) obtained from simulations.

Table 2. Daily and night performance for the making and melting ice periods for different non conventional cycles.

CYCLE	REFRIGERANT TYPE	COP daily	Nd Hour	Nn Hour	COPn	EVAPORATOR			CONDENSER		COMPRESSOR	
						Qe,n (kW)	Tin (ºC)	Tout (ºC)	Tin (º C)	Tout (ºC)	Wn (kW)	M (kg/s)
II	R-134A HFC	3.597	10	12	2.578	11.9	-7	-7	45	45	4.6	0.084
IV	R-22 HCFC	3.663	10	12	2.583	10.2	-7	-7	45	45	4.0	0.065
VI	R-507A AZEOTROPIC	3.217	10	12	2.309	14.8	-7	-7	45	45	6,4	0.141
VIII	R-407C ZEOTROPIC	3.027	10	12	2.461	13.6	-9.3	-4.7	47.5	42.5	5.5	0.087
IX	R-407C ZEOTROPIC	3.206	10	12	2.461	13.0	-9.3	-4.7	47.5	42.5	5.3	0.083
X	R-407C ZEOTROPIC	3.211	10	12	2.461	12.7	-9.3	-4.7	47.5	42.5	5.1	0.082
XI	R-407C AZEOTROPIC	3.415	10	12	2.461	12.0	-9.3	-4.7	47.5	42.5	4.9	0.077

From Tab.2 it can be observed the following:

1. Cycles VIII, IX, X, and XI show an average evaporating temperature of -7ºC, corresponding to an inlet and outlet temperatures of -9.3ºC and -4.7 ºC, respectively. Further, the average condensing temperature of these cycles is 45ºC, being the inlet and outlet temperatures of 47.5 ºC and 42.5ºC, respectively.

2. The “COPn” is approximately half of the “COPd”, shown in Tab. 1. This occurs due to the fact that the average evaporation temperature at the charging mode be lower (-7ºC) than those of the discharging mode (10ºC).
3. The “COPn” is approximately the same for all cycles, even though cycle IV (R-22) shows the highest value and cycle VI (R-507A) shows the lowest value.
4. The heat transfer rate “Qe,n” during the charging of tank has the highest value for cycle VI (R-507A) and the lowest value for cycle IV (R-22).
5. The operant fluid mass flow rate shows the lowest value for cycle XI (R-407C) and the highest value for cycle VI (R-507A). The small flow rate can be an advantage due the possibility of use of smaller compressor sizes.
6. The highest “COP daily” occurs for cycle IV (R-22) even though it is only a little bit higher than that obtained for cycle II (R-134A). The explanation is because cycle II has the smallest energy stored in the tank and, consequently, the smallest “Qen” and “Wn”.
7. Cycle XI shows the third higher value of “COP daily”. Cycle VI (R-507A) shows “COP daily” value inferior compared to cycle II (R-134A), cycle IV (R-22) and cycle XI (R-407C).
8. The relative difference between cycles II (R-134A) and XI (R-407C), in terms of the “COP daily” values, implies in energy savings of only 5 % for cycle II. However, this small difference can be offset if it is considered that the average mass flow rate of the fluid operant for cycle XI (R-407C) is approximately 5% lower than cycle II (R-134A). Therefore, this fact can reduce the size of the compressor and, consequently, the compressor cost.

Table 3 shows the influence of the outside air temperature during night time for the non conventional cycles II, IV, VI, and XI. The outdoor ambient temperature time variation affects the condensing temperature and, consequently, the values of “COPn” and “COP daily”. The “COP daily” is determined from Eq (2). The values of “COPd” and “COPn” are obtained from simulations considering the same air conditioning heat transfer rate of 56kW and the same average evaporator temperature of 10ºC and -7ºC for the charging and discharging modes, respectively. It was analyzed three condensing temperatures of 45ºC, 35ºC and 25ºC, respectively. Table 3 also shows the values of “Qen”, “Wd”, “Wn” and temperatures obtained for different non conventional cycles using different operating fluids.

Table 3. Analyses of the performance of the refrigerating system considering the effect of the outside air temperature.

CYCLE	**REFRIGERANT TYPE**	**COP daily**	**COPd**	**COPn**	**Nd Hour**	**Nn Hour**	**EVAPORATOR**			**CONDENSER**		**COMPRESSOR**	
							Qe,n (kW)	**Tin (ºC)**	**Tout (ºC)**	**Tin (º C)**	**Tout (ºC)**	**Wd (kW)**	**Wn (kW)**
II	R-134A HFC	3.597	5.589	2.578	10	12	11.9	-7	-7	45	45	10.0	4.6
		3.935	5.589	3.398	10	12	11.9	-7	-7	35	35	10.0	3.5
		4.283	5.589	4.682	10	12	11.9	-7	-7	25	25	10.0	2.5
IV	R-22 HCFC	3.663	5.308	2.583	10	12	10.2	-7	-7	45	45	10.6	3.9
		3.951	5.308	3.377	10	12	10.2	-7	-7	35	35	10.6	3.0
		4.244	5.308	4.628	10	12	10.2	-7	-7	25	25	10.6	2.2
VI	R-507A AZEOTROPIC	3.217	5.761	2.309	10	12	14.8	-7	-7	45	45	9.7	6.4
		3.652	5.761	3.162	10	12	14.8	-7	-7	35	35	9.7	4.7
		4.089	5.761	4.466	10	12	14.8	-7	-7	25	25	9.7	3.3
XI	R-407C ZEOTROPIC	3.415	5.313	2.461	10	12	12.0	-9.3	-4.7	47.5	42.5	10.5	4.9
		3.750	5.313	3.280	10	12	12.0	-9.6	-4.5	37.8	32.3	10.5	3.7
		4.091	5.313	4.576	10	12	12.0	-9.8	-4.2	28	22	10.5	2.6

From Tab.3 it can be observed the following:

1. For all non conventional cycles the “COPn” and, consequently, the “COP daily” increases as the condensing temperature decreases in response to the decreased of the outside air temperature.
2. The highest values of “COP daily” and “COPn” occur for cycle IV (R-22), even though the values are very close to ones of cycle II (R-134A). The lowest value occurs of “COP daily” and “COPn” occur for cycle VI (R-507A).
3. The relative differences between “COP daily” values of cycle II (R-134A) and cycle VI (R-407C), for all condensing temperatures, are smaller than 5%. This small difference can be offset due to the fact that the operating fluid mass flow rate of the cycle XI (R-407C) be approximately 5% lower than cycle II (R-134A).

This fact can again contribute to the reduction of the size of the compressor and, consequently, the reduction of the compressor cost.

4. CONCLUSIONS

This work shows results obtained from various computer simulations aimed to compare the global daily efficiency of direct expansion air conditioning systems with and without ice energy thermal storage systems, for application mainly in offices buildings or in small and medium commercial buildings that operates during the daytime period. Different types of refrigerants and blends of refrigerants are simulated in this work.

It was verified that global daily coefficients of performance “COP daily” of the conventional systems, that do not have ice storage tank, show higher values than those obtained for the non conventional systems that have ice storage tank The relative differences are 17.8%, 15.3%, 19.4%, respectively, for R-134A (HFC), R-22 (HCFC), and R-507A (azeptropic blend). For cycles VIII, IX, X and XI, which work with the zeotropic blend R-407C, the relative differences compared to the conventional cycle VII (R-407C), are 14.2%, 9.1%, 8.9%, and 3%, respectively. Therefore, this work shows that the use of energy storage systems do not promote energy savings. The advantage of thermal energy storage systems is that they provide a good opportunity to reduce the compressor electricity power during the daytime period, as well as, can contribute to the reduction of the electricity costs due to differentiated on peak and off peak electricity costs offered by the utilities companies. Further, the energy storage systems help the electricity utilities to decrease their electricity peak load, increasing their load factor and, consequently reducing their operational energy costs. The displacement of electricity from daytime to nighttime period corresponds to 22% (R-134A), 27.6% (R-22), 30.7% (R-507A) and 34% (R-407C).

It was performed computer simulations to verify the effect of the external ambient temperature (outdoor air) in the performance of the cycles. It has been analyzed the condensing temperatures of 45 ºC, 35 ºC and 25 ºC. The “COP daily” increases 9.4% (R-134A), 7.9%, (R-22), 13.5% (R-507A) and 9.8% (R-407C) when the condensing temperature decreases from 45ºC to 35 ºC. The efficiency “COP daily” increases 19% (R-134A), 16%, (R-22), 27% (R-507A) and 19.8% (R-407C) when the condensing temperature decreases from 45ºC to 25 ºC. Therefore, the global energy efficiency of the non conventional cycles can be improved due to drop of outdoor air temperature during the charging mode (ice formation period) causing increase of the nocturnal “COPn”.

5. REFERENCES

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