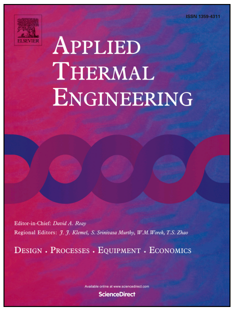

Feasibility limits and performance of an absorption cooling machine using light alkane mixtures

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Feasibility limits and performance of an absorption cooling machine using light alkane mixtures

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Abstract

The performance of a heat-driven vapor absorption chiller with various alkane mixtures as working pairs was studied. A Thermodynamic analysis showed that under specified operating conditions and with a generator temperature below 130°C, temperature achievable with a simple flat plate collector when solar energy is expected as the driving heat source, the application of some of the proposed alkane mixtures is not feasible. Simulations using ASPEN Plus flow sheeting program are then done with the selected working pairs. All simulations were done specifying the Peng-Robinson equation of state as the property method. A parametric study was carried out allowing the investigation of the generator temperature effect on the system performance and the comparison between performances released with each working pair.. Results revealed that a water-cooled absorption machine using the C3H8/n-C9H20 pair as working fluid releases the best performances from a heat driving temperature level of about 100°C.

Keywords: absorption; cooling; chiller; alkane; Oldham; COP

1. Introduction

Research dealing with the production of cold by absorption technique for air-conditioning and refrigeration purposes has been performed, since few decades, in an attempt to better manage the available energy resources and better protect the environment. The important and distinguishing features of the absorption technology compared to the conventional vapor-compression systems have been already widely enumerated, described and analyzed in the relevant literature [1,2].

The absorption system performances and its limiting operating conditions are closely related to the refrigerant/absorbent fluid system. Commonly used working fluids mixtures are NH3/H2O and H2O/LiBr. Despite their many advantages, these working fluids present serious drawbacks that limit the application possibilities of each of them. Water as refrigerant cannot be used for temperatures below 0°C as needed in refrigeration machines. This constraint makes the H2O/LiBr system only suitable for air conditioning purposes. Furthermore, at high temperatures and/or important salt concentrations, the H2O/LiBr solution presents problems of corrosion and crystallization of the salt which are only avoidable by a continuous control of the heat rejection temperature in the machine. In addition, the negative evaporation pressure is a real drawback of the H2O/LiBr system. The water/ammonia system does not present these problems but needs rather high generator temperatures. The drawbacks of the NH3/H2O system are also the relatively difficult refrigerant/absorbent separation requiring a rectification device, high working pressure and ammonia toxicity associated with corrosive behavior towards copper. Regarding these drawbacks and limitations, the search for alternative refrigerant/absorbent mixtures is largely justified.

Hydrocarbons as refrigerants are chemically stable over a wide temperature range, non-toxic and environmentally friendly with a zero ozone depletion potential (ODP) and an extremely low global warmng potential (GWP) [3]. They exhibit good thermodynamic and transport properties [4]: low viscosity and high thermal conductivity, which result in good performances of the condenser and the evaporator. They are also characterized by a good compatibility with copper the material of choice for such devices. The only real problem with the application of hydrocarbons as refrigerants in refrigeration and air-conditioning systems is their flammability. However, this could be avoided by considering some special precautions during installation and handling. We just need to look at almost all Tunisian kitchens which have been since many decades equipped with the 13kg LPG bottles to soon realize that the problem can be easily managed. Some researches are performed to minimize eventual risk when hydrocarbons are used in cooling systems. Reducing the amount of the refrigerant charge of hydrocarbons-based absorption cooling systems without affecting the COP could be an interesting challenge for coming research works. A similar work was done by Fernando et al. with propane heat pumps [5].

Light hydrocarbons and alkane mixtures as refrigerants in vapor-compression-based refrigerating machines and heat pumps were considered in the literature [4,6] but research concerning their use in absorption machines are rare. More extensive investigations are still needed. Semanani-Rahbar and Le Goff [7] analyzed cooling and heating performances in absorption systems using hydrocarbon binary mixtures. They concluded that best performances are obtained when absorbent and refrigerant are most similar. Chekir et al. [8] presented FORTRAN-based simulation results of an absorption refrigeration machine based on mass and energy conservation equations. Ten alkane mixtures were considered with both air and water cooling.

In the present study we analyze, relying on ASPEN Plus [9] simulations, the global behavior and performance of an absorption system when utilizing light alkane mixtures as working fluids. All combinations of C3H8 and n-C4H10 as refrigerants and n-C4H10 to n-C9H20 as absorbents are considered. We begin by determining, under specified operating conditions, the feasible generator temperature range for each working fluid system. Then, parametric analyses are conducted to evaluate the effect of the system key parameters allowing performance comparison of the considered systems. Air and water were alternatively used to cool the absorber and the condenser. Although the low normal boiling point of C3H8 allows it to be used in refrigeration machines, this study is concerned with the performance of an air-conditioning device for chilled water (12-7°C) production.

2. Cycle description

The model of the considered machine is schematically shown in Figure 1. Its main components are a generator, a rectifier, a condenser, an evaporator, an absorber, two expansion valves (VLV1 and VLV2), a circulation pump and two regenerative heat exchangers (HX1 and HX2).

The refrigerant rich solution exiting the absorber (1) is pumped to the generator via the solution heat exchanger (HX1) where it is preheated. Heat from an external heat source is supplied to the generator to separate the refrigerant. The weak solution leaving the generator preheats the rich solution when passing through the solution heat exchanger (HX1). To maintain the absorber-generator pressure difference, the throttling valve assures a pressure drop of the weak solution returning to the absorber at state 6. In the absorber, the weak solution absorbs the refrigerant vapor coming from the evaporator and then leaves it as a low pressure rich solution at state 1. The absorbent cycle is then complete. The high pressure refrigerant vapor generated through the generator is purified in the rectifier if containing some amount of absorbent vapor. The resulting vapor which is formed of pure refrigerant (if the separation is complete) enters the condenser at state 7. It condenses and rejects the heat of condensation to the cooling medium (air or water). The condensate is sub cooled then through the vapor-liquid heat exchanger (HX1) before being throttled in the throttling valve assuring the condenser/evaporator pressure difference. The low pressure, low temperature refrigerant enters then the evaporator where it evaporates producing the required cooling. The refrigerant vapor leaves the evaporator at state 11, passes through the vapor-liquid heat exchanger (HX1) and finally enters the absorber (12) completing so the refrigerant cycle.

3. Thermodynamic property

The present work is an Aspen Plus based analysis of an absorption process. This flowsheeting software has an extensive library of built-in thermodynamic models. The choice of the appropriate model is of crucial importance to get reliable results. The Pen-Robinson equation of state is generally used to predict the thermodynamic properties and to calculate phase equilibrium of light hydrocarbon mixtures. L. Luyben [10] recommends the Chao-Seader model for these systems.

The Peng-Robinson-based predictions and the Chao-Seader-based ones were compared with the experimental vapor-liquid equilibrium data [11] from the literature for ten binary pairs of the considered alkane components. Figure 2, displaying this comparison in the case of the C3/n-C9 system, shows that the Peng-Robinson equation of state is in better agreement with the experimental data. The same result was obtained for all the considered alkane mixtures [12]. To obtain results with a high accuracy level, simulations should be done specifying the Peng-Robinson equation of state as the property method.

4. Feasibility limits

Air at 35°C and water at 25°C are alternatively considered as cooling media for the condenser and the absorber. The condensation and the absorption end temperatures, assumed to be equal are set to 54°C and 34°C respectively. Further assumptions for the following simulations are:

- The heat driving temperature does not exceed 130°C,
- The refrigerant vapor purity is about 99%,
- The evaporator exit temperature is about 0°C,

These assumptions associated with appropriate operating conditions limit the driving heat temperature range which is a key parameter of the system performance. Below a low limit there is no possibility to separate the absorbent/refrigerant mixture; the pressure in the absorber is the equilibrium pressure for the refrigerant exit temperature, from this pressure value and the solution exit temperature is calculated the equilibrium solute concentration. The equilibrium temperature related to this solute concentration and to the desorber pressure is the low driving heat temperature.

On the other hand complete separation does not need to exceed a maximum value. To determine these limiting temperatures the Oldham diagram of the refrigerant/absorbent system is used. Figure 3 shows the procedure in the case of the C3/n-C9 mixture and water cooling. X, Tmin and Tmax respectively represent the solution composition, the minimum and maximum heat driving temperature. Results for all the considered alkane pairs are presented in table 1 in the case of air cooling and table 2 for water cooling.

When considering air cooling the n-C4/n-C5 and the C3/n-C4 Oldham diagrams show that the usage of these pairs as working fluids in absorption machines is not possible with the previously specified refrigerant purity and evaporator exit temperature. Furthermore, for driving temperatures below 130°C, this table shows that only the n-C4/n-C6 system can be used as working fluid in absorption machines. In the case of water cooling, it is shown that all of considered alkane binary mixtures (n-C4/n-C5, n-C4/n-C6, n-C4/n-C7, n-C4/n-C8, n-C4/n-C9, C3/n-C4, C3/n-C5, C3/n-C6, C3/n-C7, C3/n-C8 and C3/n-C9) can be used.

5. Machine model

Processes are defined on the flowsheeting program via a graphical interface. The absorption cooling system in the present investigations is modeled by series of connected unit operations, blocks (figure 4). While absorber, evaporator and condenser are modeled using a single HEATER block, both the solution and the vapor-liquid heat exchangers were modeled using two HEATER blocks connected with a heat stream. The generator-rectifier system is modeled using the RadFrac block and the two throttling valves were respectively represented by the VALVE and the PUMP modules.

The model is based on mass and energy balances applied on each component of the machine. The principal equations of the model are the following

Total mass balance $\sum \dot{m}_{in} = \sum \dot{m}_{out}$

Partial mass balance $\sum \dot{m}_{in}\, z_{i,in} = \sum \dot{m}_{out}\, z_{i,out}$

Total energy balance $\dot{Q} = \sum \dot{m}_{in}\, H_{in -} \sum \dot{m}_{out}\, H_{out}$

with

$\dot{m}$: mass flow rate, kg/s

z_i : mass fraction of the component *i*

H : enthalpy, kJ/kg

$\dot{Q}$: heat transfer rate, kJ/s

6. Analysis, simulation and results

a. Assumptions and operating conditions

The main assumptions and operating conditions considered for the modeling and the simulation of the absorption machine are summarized in table 3. As noted earlier absorption and condensation end temperatures depend upon the cooling medium temperature.

The flowsheeting program used in this work is supported by design features that ensure convergence in multi-loop feed-back connections. After specifying the components, choosing the thermodynamic model and indicating the parameters of each unit operation design specification convergence blocks must be added. They allow adjusting iteratively the values of some variables to meet the problem design specifications. In our model we used two design specifications. The cooling capacity of the evaporator is set to 17.5 kW. To meet this specification the molar flow rate of the rich solution leaving the absorber was adjusted iteratively until convergence. Similarly the pump efficiency is adjusted to ensure an isentropic behavior. In the radfrac column, the number of theoretical stages is specified for each alkane mixture, following an economic optimization based on heuristic rules [10]. The value of reflux ration is adjusted by setting a design specification to ensure the desired vapor refrigerant purity.

We also used a Calculator block incorporating FORTRAN statements to meet the solution heat exchanger pinch temperature specification. The pinch point is usually located at the cold side of a heat exchanger [13-14]. The choice of the pinch point temperature is made in a manner to avoid having large exchangers in one hand, and to guarantee a minimum quality transfer. In our case we applied heuristic rules. Thus, we assumed a pinch of 5°C for the solution heat exchanger HX2 ($T_5 = T_2 + 5°C$) and of 10°C for the liquid-vapor one HX1 ($T_9 = T_{11} + 10°C$).

b. Simulation results

Simulations are performed for the n-C4/n-C6 mixture as working fluid in the case of air cooling and for all the eleven considered mixtures in the case of water cooling. In order to compare the working fluid systems the cooling coefficient of performance COP, defined as the ratio of the evaporator heat duty to the generator heat input power, i.e.

$$COP = \frac{\dot{Q}_{ev}}{\dot{Q}_{gen}}$$

is used.

We note here that we don’t include the pump power consumption in the determination of the COP. In fact, we are interested only about the thermal COP.

On the other hand, in absorption machines the evaporation pressure can be varied between an upper limit corresponding to the saturation pressure of the liquid refrigerant at the evaporator outlet temperature and a lower limit corresponding to the saturation pressure of the poor solution at the absorber temperature [8]. Above the upper limit pressure the refrigerant doesn’t evaporate anymore and below the lower limit, the vapor refrigerant cannot be absorbed.

Figure 5 depicts the evolution of the COP of the C3/n-C9 water-cooled machine with the pressure in the evaporator/absorber compartment. It shows that a maximum COP of 0.45 is reached for a pressure ranging from 4.5 to 4.7 bars. The rest of the simulations are run by specifying a low pressure of 4.5 bars for propane as refrigerant and 1 bar for butane.

The effect of the driving heat temperature on the COP of the machine is depicted in figure 6 for the case of the system C3/n-C9. It is shown that by approaching the lower limit of the heat driving temperature range the COP decreases rapidly. This can be explained by the fact that the refrigerant concentrations in the weak and strong solution are getting closer for decreasing reboiler temperatures, associated with a higher circulation factor for fixed chiller capacity, thus leading to larger energy consumption in the generator. Above 110°C, the COP is almost constant. The highest COP is obtained for a driving heat temperature ranging from 120 to 125°C. Increasing the temperature further is unnecessary and increases the exergy losses, resulting in declining COP. Similar results had been found by Engler et al. [15] when analyzing ammonia-water absorption cycles. The detailed simulation results of the water-cooled machine using the n-C3/n-C9 pair are summarized in tables 4 and 5.

The same procedure, as illustrated in the case of the C3/n-C9 is performed for the rest of working fluid systems, namely, n-C4/n-C6 in the case of air cooling and C3/n-C4, C3/n-C5, C3/n-C6, C3/n-C7, C3/n-C8, C3/n-C9, n-C4/n-C6, n-C4/n-C7, n-C4/n-C8 and n-C4/n-C9 in the case of water cooling. In view of the obtained results several remarks can be formulated:

- When air is used as cooling medium, the maximum COP reached with the n-C4/n-C6 system, the only alkane mixture suitable in this case, is very low (0.11).
- For both propane and n-butane as refrigerants, the heat driving temperature assuring the system best performance increases with the number of carbon atoms of the

absorbent (figure 7). It is important to note here that the highest temperature is about 123°C and remains close to the imposed restriction of 130°C.

- For a given refrigerant, the COP increases with the length of the hydrocarbon chain of the absorbent (figure 8). The best performance is reached thus with the C3/n-C9 system for the propane series and n-C4/n-C9 for n-butane.
- For mixtures with propane as refrigerant, the reflux ratio is about 4.23 in the case of C3/n-C4 and decreases to 0.03 with the C3/n-C9 pair. Results show that the machine performances are closely linked to the generator-rectifier system. These results are in good agreement with findings of Darwish et al.[16].

Compared to those of commercial available single effect H2O/LiBr absorption chillers (COP=0.7-0.8) [17], the performances reported in this study when using the C3/n-C9 pair as working fluid in water-cooled units are of real interest.

Evaluation of an ammonia/water absorption refrigeration system, using Aspen plus software, is done by Darwish et al. [16]. Their simulation results showed the reliability of the used simulator to model and compute such devices. Our two research works present a high coherence particularly in the way of modeling the absorption machine.

Other theoretical studies dealing with binary alkane mixtures as working fluids for absorption chillers have been performed. Chekir et al. [8] found that for a water-cooled chillers the n-C4H10/n-C8H18 and the n-C3H8/n-C8H18 ensure the best performance with a COP of about 0.63 for a driving heat temperature of about 130°C. Our results are in good agreement with these findings, noting that the systems n-C4H10/n-C9H20 and n-C3H8/n-C9H20 were not considered as possible working fluids and that higher driving heat temperature was used.

To our knowledge, no experiments have been published on absorption chillers using alkane mixtures as working fluids. Solely, Ben Ezzine et al. [18] carried out an experimental investigation on an air-cooled low capacity absorption diffusion machines using the mixture (n-C4H10/n-C9H20) as working fluid associated to helium as inert gas. They found that cold is produced at temperatures between -10°C and 10°C for a driving temperature in the rage of 120-150°C, which is in concordance with our results. The performance however is expectedly not comparable.

7. Conclusions

In this paper, the feasibility limits and performances of absorption chiller using alkane mixtures as working fluids and operating with a generator temperature not exceeding 130°C were investigated. Alternative cooling media were considered; water in association with a cooling tower and air. A preliminary thermodynamic analysis showed that under the specified operating conditions solely the n-C4/n-C6 mixture may be considered for air-cooled machine. Aspen plus simulations were then run to evaluate cooling machine performances using this working fluid mixture. It was found, however, that the COP reached is rather poor (0.11) and the system was hence rejected. In the prevailing operating conditions none of the considered mixtures is viable for air-cooling.

In the case of water cooling, all the binary mixture systems can be used. Best machine performances are reached with the C3/n-C9 mixture with a COP about 0.5, comparable to that of the commercial available single effect H2O/NH3 absorption chillers.

It can be concluded that this particular working fluid system C3/n-C9 represents a potentially feasible alternative for heat driven cooling machines with source temperature less than 130°C. It is worth noting that unlike the ammonia system, the machine can be built using copper, the material of choice for this kind of heat transfer device.

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TABLES

Table 1. Feasibility limits for the air-cooled system

	Driving heat temperature (°C)	
	Tmin	**Tmax**
C3/n-C4	Could not be used	
C3/n-C5	129°C, x∈[0.174, 0.174]	159°C, x∈[0, 0.174]
C3/n-C6	139°C, x∈[0.268, 0.268]	201°C, x∈[0, 0.268]
C3/n-C7	149°C, x∈[0.281, 0.281]	240°C, x∈[0, 0.281]
C3/n-C8	155°C, x∈[0.295, 0.295]	274°C, x∈[0, 0.295]
C3/n-C9	159°C, x∈[0.297, 0.297]	305°C, x∈[0, 0.297]
n-C4/n-C5	Could not be used	
n-C4/n-C6	121°C, x∈[0.094, 0.094]	134°C, x∈[0, 0.094]
n-C4/n-C7	130°C, x∈[0.162, 0.162]	168°C, x∈[0, 0.162]
n-C4/n-C8	138°C, x∈[0.191, 0.191]	199°C, x∈[0, 0.191]
n-C4/n-C9	143°C, x∈[0.199, 0.199]	227°C, x∈[0, 0.199]

Table 2. Feasibility limits for the water-cooled system

	Driving heat temperature (°C)	
	Tmin	**Tmax**
C3/n-C4	73°C, x∈[0.188, 0.188]	87°C, x∈[0, 0.188]
C3/n-C5	79°C, x∈[0.334, 0.334]	133°C, x∈[0, 0.334]
C3/n-C6	81°C, x∈[0.420, 0.420]	174°C, x∈[0, 0.420]
C3/n-C7	83°C, x∈[0.422, 0.422]	210°C, x∈[0, 0.422]
C3/n-C8	84°C, x∈[0.435, 0.435]	244°C, x∈[0, 0.435]
C3/n-C9	84°C, x∈[0.463, 0.463]	274°C, x∈[0, 0.463]
n-C4/n-C5	72°C, x∈[0.031, 0.031]	74°C, x∈[0, 0.031]
n-C4/n-C6	75°C, x∈[0.276, 0.276]	110°C, x∈[0, 0.276]
n-C4/n-C7	79°C, x∈[0.310, 0.310]	142°C, x∈[0, 0.310]
n-C4/n-C8	80°C, x∈[0.331, 0.331]	172°C, x∈[0, 0.331]
n-C4/n-C9	81°C, x∈[0.336, 0.336]	199°C, x∈[0, 0.336]

Table 3. Assumptions and operating conditions

	State point
Saturated liquid solution at generator exit	4
Saturated vapor at rectifier exit	7
Refrigerant vapor purity ≥ 99%	7
Liquid sub cooling at condenser and absorber exit, 4°C	8 et 6
Evaporator exit temperature, 2°C	11
HX2 & HX1 temperature pinch, 5 and 10°C	-
Cooling capacity, 17.5 kW	-
Driving heat temperature ≤130°C	4

Table 4. Simulation results of the C3/n-C9 water-cooled machine

Point	**Temperature** T(°C)	**Pressure** P(bar)	**Molar composition** x, y	**Molar flow** $\dot{n}$ (kmol/hr)
1	30	4.5	0.42	23.6
2	30	11.76	0.42	23.6
3	101	11.76	0.42	23.6
4	120	11.76	0.28	19
5	35	11.76	0.28	19
6	36	4.5	0.28	19
7	82	11.76	0.99	4.6
8	30	11.76	0.99	4.6
9	12	11.76	0.99	4.6
10	0	4.5	0.99	4.6
11	2	4.5	0.99	4.6
12	14	4.5	0.99	4.6

Table 5. Thermal performances of the C3/n-C9 water-cooled absorption machine

Generator heat flow rate (kW)	34.5
Rectifier heat transfer rate (kW)	-3.2
Condenser heat transfer rate (kW)	-24.6
Cooling capacity (kW)	17.5
Absorber heat transfer rate (kW)	-24.9
Pump work power (kW)	0.7
Cooling coefficient of performance : COP = 0,51	

FIGURES

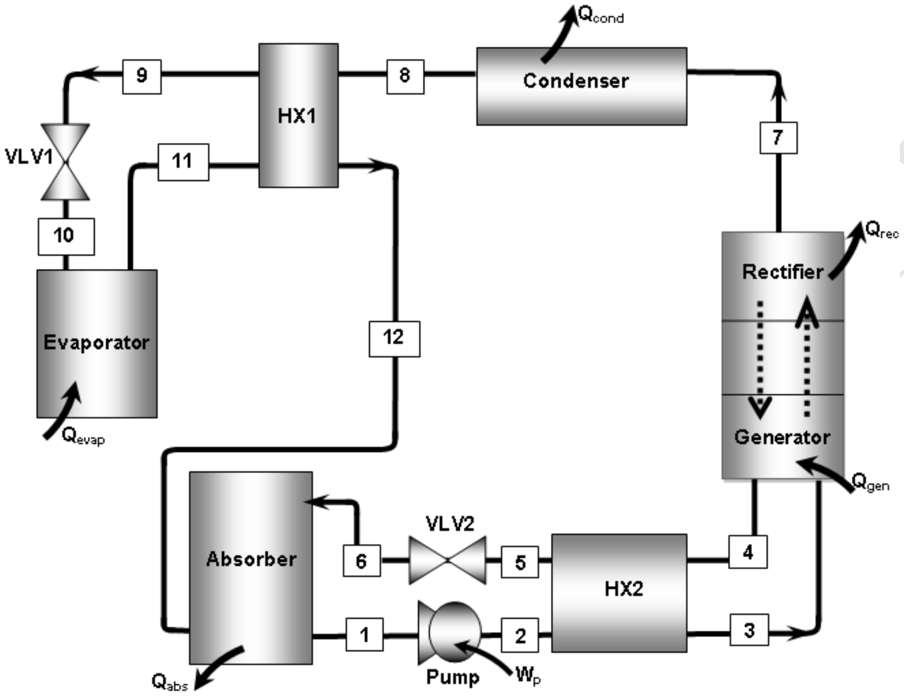

Figure 1. Schematic of the absorption cooling machine

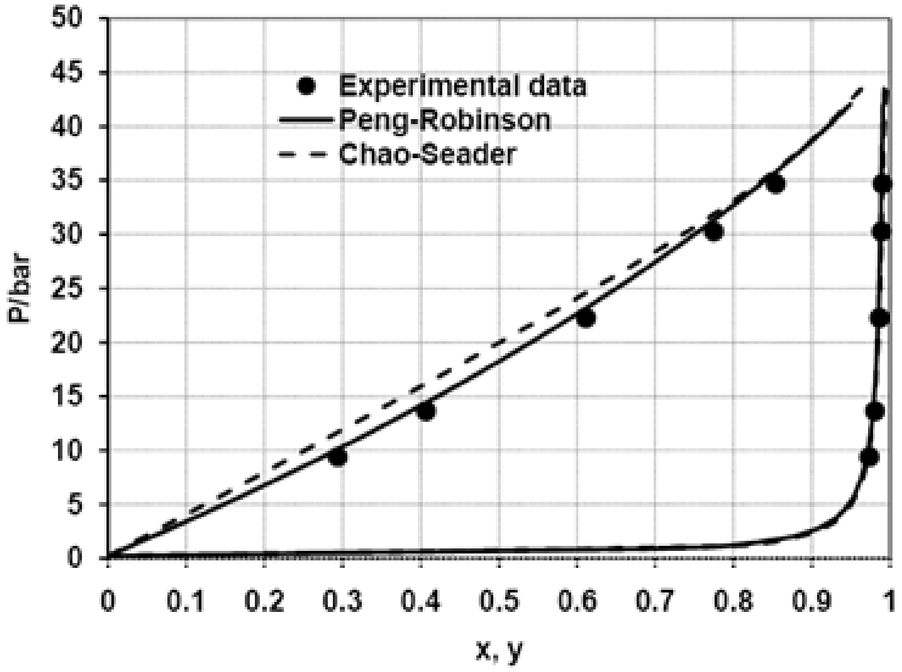

Figure 2. P-x-y diagram of C3/n-C9 (T=376.15K)

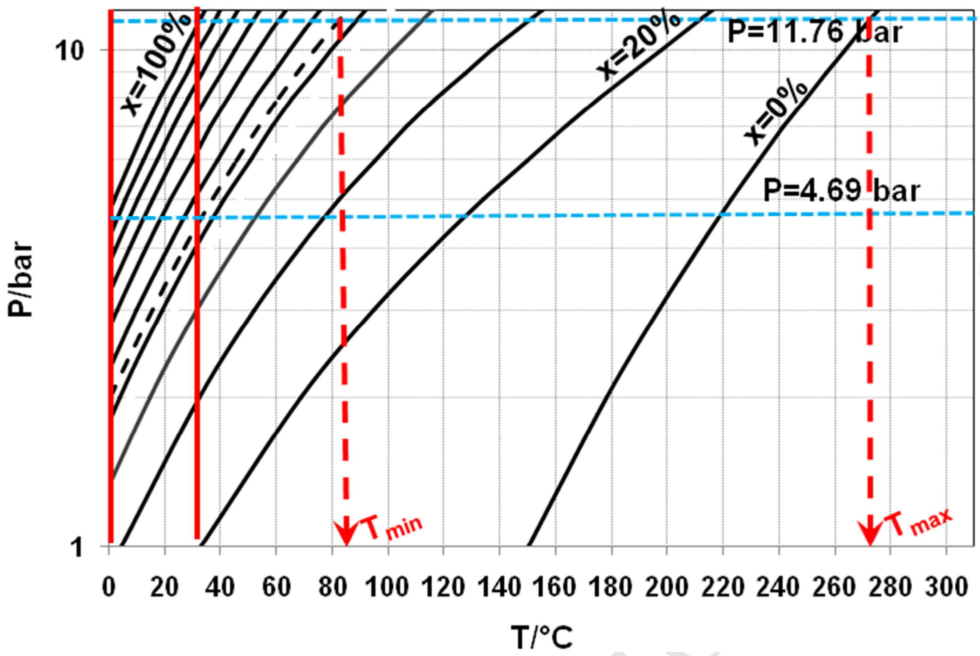

Figure 3. Oldham diagram for C3/n-C9

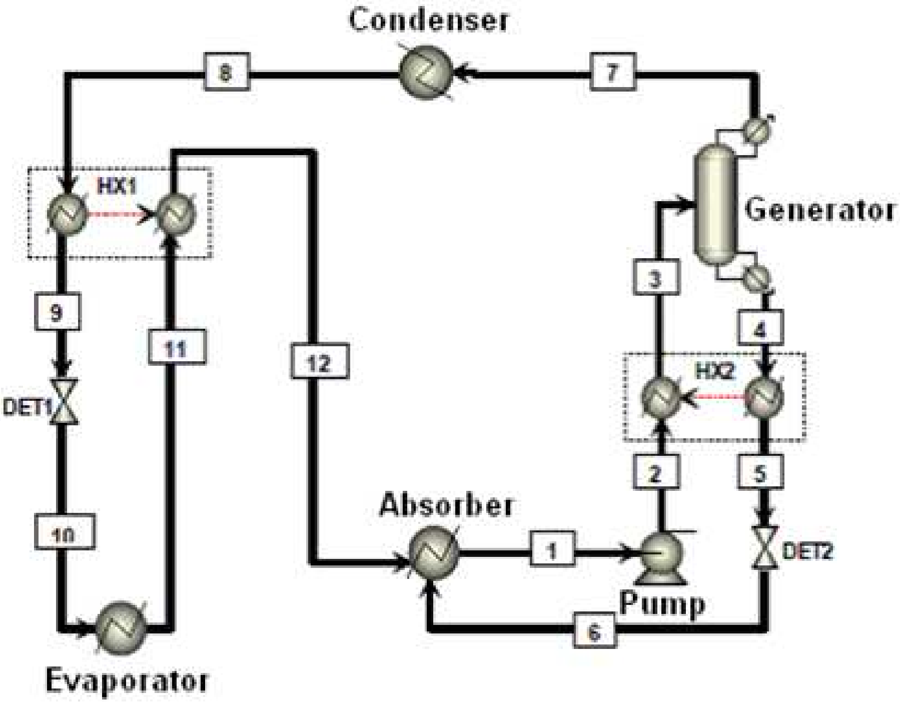

Figure 4. ASPEN Plus flow sheet of the absorption cooling machine

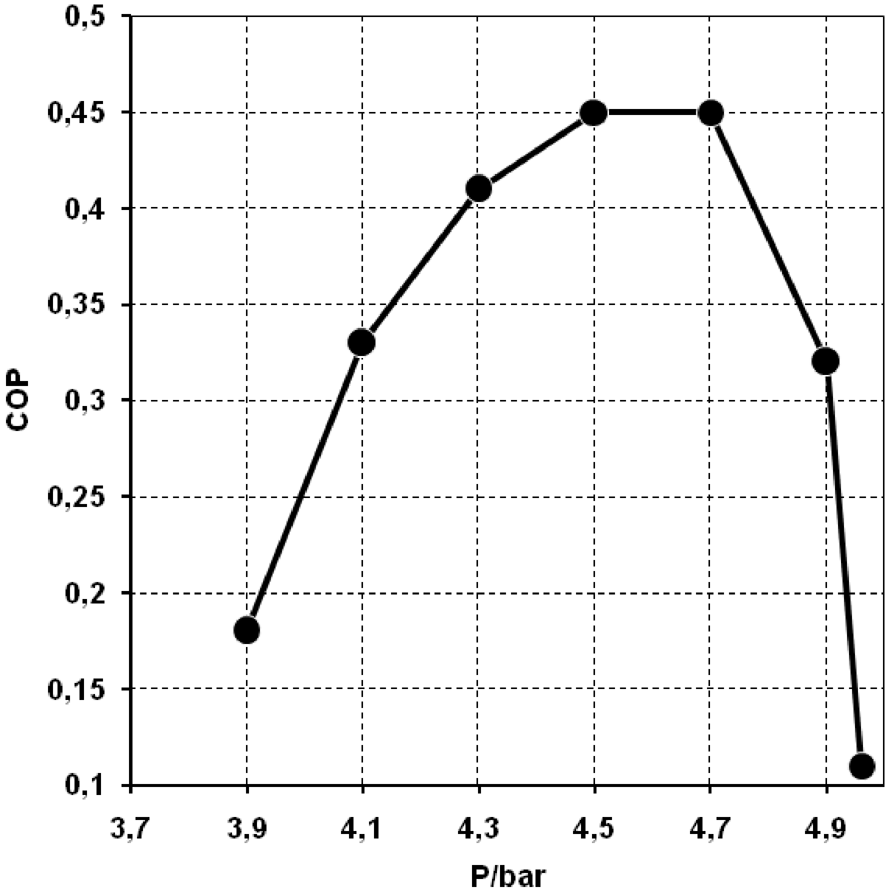

Figure 5. COP vs. system low pressure for the n-C3/n-C9 pair (Water-cooling)

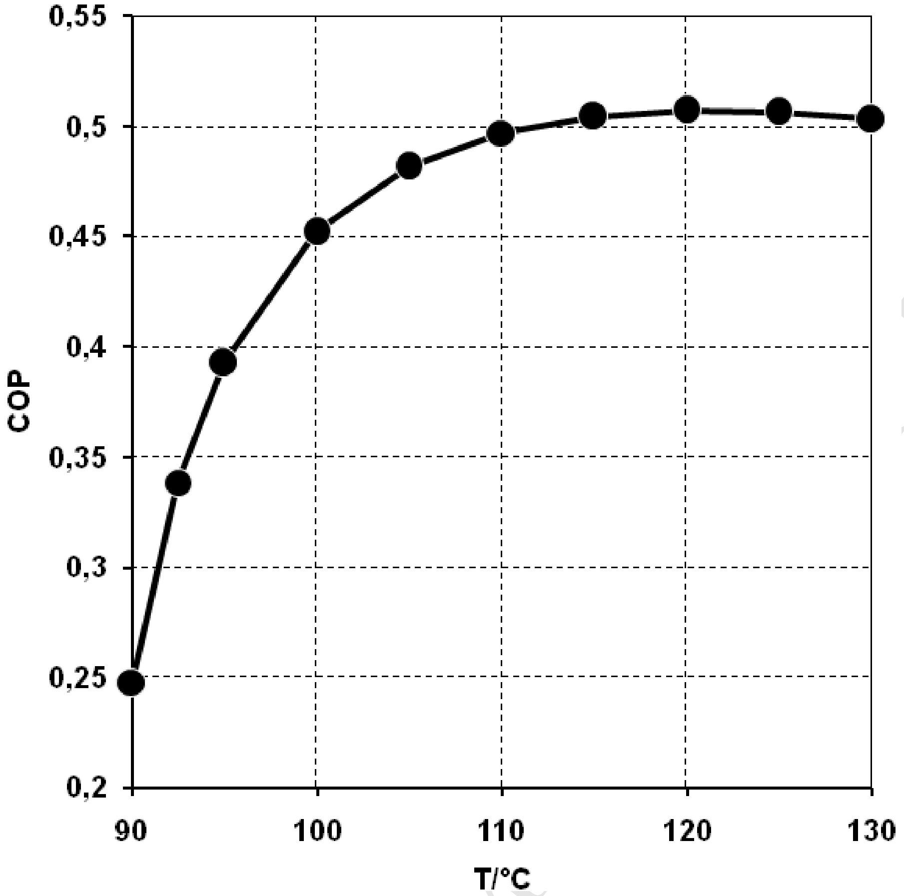

Figure 6. COP vs. reboiler temperature for the n-C3/n-C9 pair (Water-cooling)

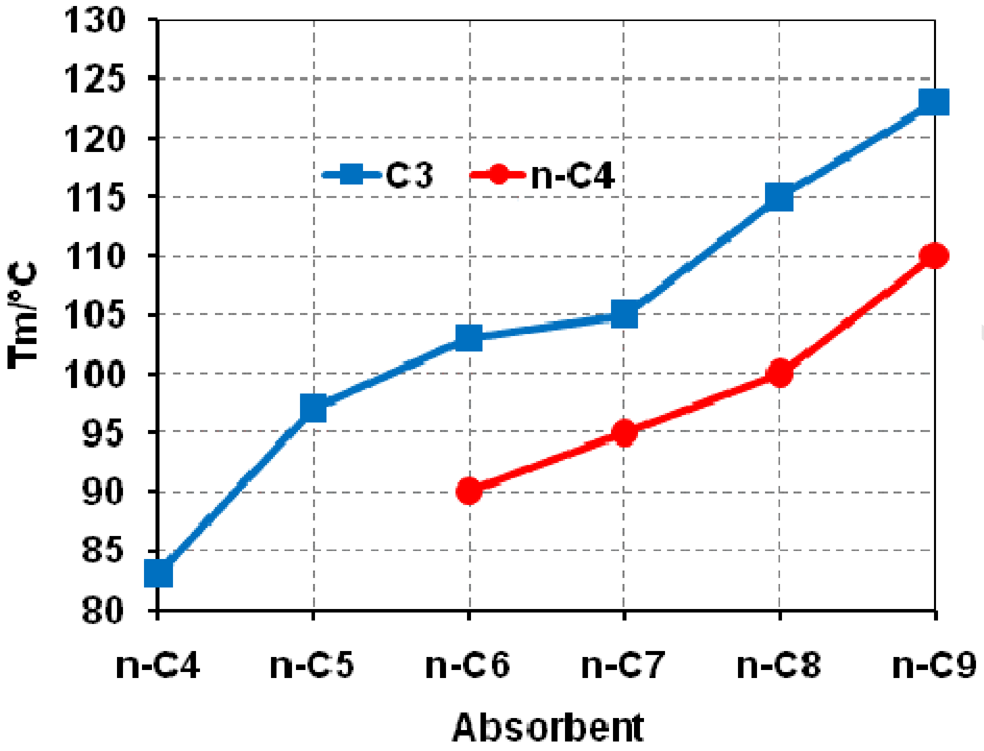

Figure 7. Reboiler temperature for maximum COP with considered alkane mixtures

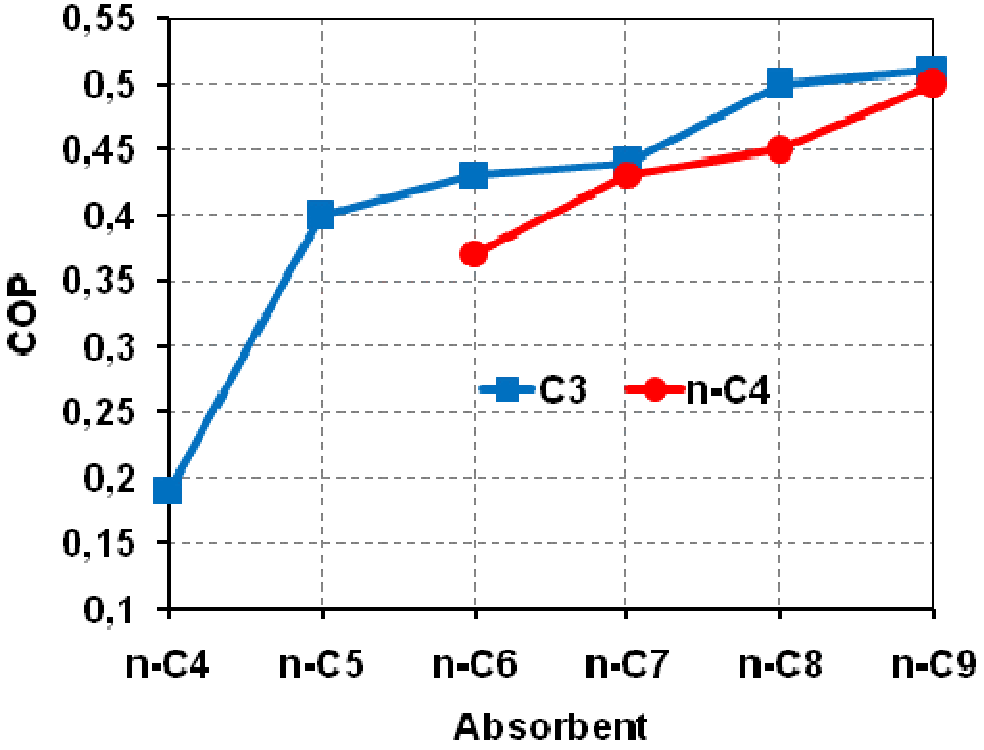

Figure 8. Maximum COP with considered alkane mixtures

Highlights

- Performance of an absorption chiller with various alkane mixtures was studied.
- Some of the proposed alkane mixtures is not feasible.
- Only the n-C4/n-C6 mixture may be considered for air-cooled machine.
- In case of water cooling, C3/n-C9 and n-C4/n-C9 give the best COP.