The use of a low-grade thermal energy in a combined refrigeration system for multi-temperature cold production

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ABSTRACT

In this paper a combined refrigeration system of cold production with two levels of temperature including an ejector and a mechanical compressor was proposed and investigated. The objective being to use the low-grade thermal (waste energy) to power the ejector for the low temperature cold production (freezing) and the compressor for the intermediate-temperature (refrigeration), in combined refrigeration cycle with the R134a refrigerant. The equations of the model for which the resolution is carried out by the EES software [1] use the state variables based on Martin-Hou equation of state. The results of the described model permitted the analysis of the influence of generator temperature (heat source) and that of the condenser (surrounding) on the system efficiency (Coefficient of Performance). For generator temperatures close to 100°C, the obtained performances permitted the consideration of certain applications in the tropical zones, notably where the solar radiation density can enable having such temperatures with a single collector.

Key-Words: combined refrigeration systems, multi-temperature systems, ejector, low-grade thermal sources, refrigerant R134a, coefficient of performance

RESUME

Dans le présent travail est proposée et étudiée une installation combinée de production de froid, à deux niveaux de températures, comportant un éjecteur et un compresseur mécanique. L'objectif étant d'utiliser les sources thermiques de faible potentiel (effluents thermiques) pour faire fonctionner l'éjecteur à basse température (congélation) et le compresseur en moyenne température (réfrigération), dans un cycle combiné au réfrigérant R134a. Les équations du modèle dont la résolution est faite à l'aide du logiciel EES [1] utilisent les variables d'état basées sur l'équation d'état de Martin-Hou. Les résultats du modèle décrit ont permis d'analyser l'influence des températures du générateur (source de chaleur) et du condenseur (milieu ambiant) sur l'efficacité du système (coefficient de performance). Pour des températures au générateur voisine de 100°C, les performances obtenues sont telles qu'on puisse envisager des applications, notamment en zone tropicale où la densité des radiations solaires permet d'avoir de telles températures avec un simple collecteur.

Mots-Clé: système frigorifique combiné, double température, éjecteur, sources thermiques de faible potentiel, frigorigène R134a, coefficient de performance

NOMENCLATURE

V	velocity, m/s
	heat rate , kW
	mass flow rate, kg/s
	power, kW
p	pressure, bar
p_{cr}	critical pressure, bar
T	temperature, °C
s	specific entropy, kJ/kg°C
h	specific enthalpy, kJ/kg
COP	coefficient of performance
x	quality
w	mass flow entrainment ratio
k	mass flow rate ratio through evaporators 1 and 2

Greek Letters

η	isentropic efficiency

Subscripts

1, 2,...	points of the cycle
a, b	points of the ejector
l	liquid
g	gas, generator
p	pump
e_1	evaporator1
e_2	evaporator 2
c	condenser
d_1	expansion valve1
d_2	expansion valve2

I- INTRODUCTION

Vapour compression refrigerating systems are the most widely used for cold production in the refrigeration as well as in the air conditioning systems due to their high COP. However, these devices use complex mechanical compressors that consume essentially high quality electric energy from such devices like an electric motor, a mechanical power engine etc. Ejection refrigerating systems have a low COP and consequently are less prevalent in cold production industries [2]. On the other hand, the latter have the advantage of being functionally simple, have no moving parts, and use a thermal energy source.

Cold production at two different temperature levels in the same system is carried out by double expansion. In this case, the realisation of a cycle, in most of the cases necessitates the use of a two-stage compression, without which an inadmissible compression ratio may be obtained, and this is accompanied in general by a high consumption of mechanical or electrical energy. When ever this energy is obtained from a thermal power plant or a combustion engine, it is not only expensive, but greenhouse gases and other pollutants are generated as well as usable waste energy [2]. Previous works on ejector refrigerating systems [2] showed that generators of these devices can efficiently use this waste heat, even when available thermal source is of a low-grade [3] like unused condensates, or collected solar thermal energy.

Coupling the ejector to the mechanical compressor, a hybrid system is realized which presents both theoretical interest as well as practical. Da Wen Sun [4], Huang et al. [5], presented papers on compression-ejection combined cycles destined for air conditioning and refrigeration at average temperature. Tuagi and Murty [6] equally demonstrated the utility of these refrigerating systems for the valorisation of thermally wasted energy. According to this coupling cycle, these systems nevertheless, present much interest for low and multi-temperature refrigeration; but no significant study has been carried out in this domain, notably where the temperature conditions for tropical zones is considered. On the other hand, the ejector (with low efficiency) has been the object of many works, notably those of Chou, Yang and Yap [7] and of Alexis and Rogdakis [8]. The former developed model for the prediction of the maximum mass flow ratio due to secondary flow choking for different characteristic of flow entrainment zones of the ejector, while the latter developed a mathematical model that contributed significantly on the understanding and amelioration of the performance of the ejection refrigerating systems at various conditions. But a complete study on combined systems coupling an ejector to a compression system is still to be carried out.

In this paper, a theoretical study of a combined compression-ejection refrigerating system where the low-pressure compressor of a two-stage system is replaced by an ejector is carried out. The two evaporators of such a system with different temperatures could integrate different compartments of cold chambers. The two temperature levels of evaporation chosen for this study corresponding to refrigeration and freezing are: t_{e1} = -5°C and t_{e2} = -25°C. The heat source susceptible of producing the motive stream to the generator are very varied, but it will be considered that the heat is of a solar source and that the vapour temperature (supposed saturated) lies between 85°C and 100°C. The help of solar collectors endowed with accumulators can realize this and it should be noted that, this temperature range has the advantage of maintaining the pressure of the R134a below that of the critical point.

After presenting the model describing the behaviour of the ejector in the system, the COP (that characterises the efficiency of the combined system) will be determined. Later, the evolution of the performance of the system as a function of the temperature level of the heat source, the temperature level of the condenser, and the mass flow rate ratio of the refrigerant in the two evaporators of the system is investigated.

II- DESCRIPTION OF THE CYCLE

The schematic of the proposed system in figure1 is a combination of a power cycle and a reversed Rankine cycle. The power cycle is composed of a generator, an ejector, a condenser and a refrigerant pump, while the reversed cycle is of a two-stage type where the low-pressure compressor is replaced by an ejector. The work done in the power cycle is used in the latter to compress the low-pressure vapour in evaporator2, while the mechanical compressor sucks and discharges at the condensation pressure, the average-pressure vapour mixture from evaporator1 and the ejector at the expense of the absorbed mechanical energy. Finally, the condenser (common to the both cycles), constitutes the heat source, while the two evaporators serve as the heat sinks. This reversed Rankine cycle is completed by a double expansion in the expansion valve1 and expansion valve2 respectively.

Figure2 represents the T-s diagram of the thermodynamic cycle of the system. The working of this system consists of so many thermodynamic transformations. The thermal energy from the thermal source (solar collector) heats up and vaporizes the liquid fluid from the pump until a dry saturated vapour is obtained (point1). These motive vapours so-produced (supposed at stagnation condition) penetrate the ejector where they are carried along and compressed with those from the evaporator2 (point2) at the low pressure P_{e2}. The resulting mixture is discharged (point 3) at an intermediary pressure equal to that of the evaporator1. These vapours of the fluid ejected under pressure mix with those from the evaporator1 (point4) and the whole mixture (point5) is admitted into the mechanical compressor that compresses them again before discharging them to state (6) at the pressure P_c. Condensation takes place at this pressure with heat rejection to the

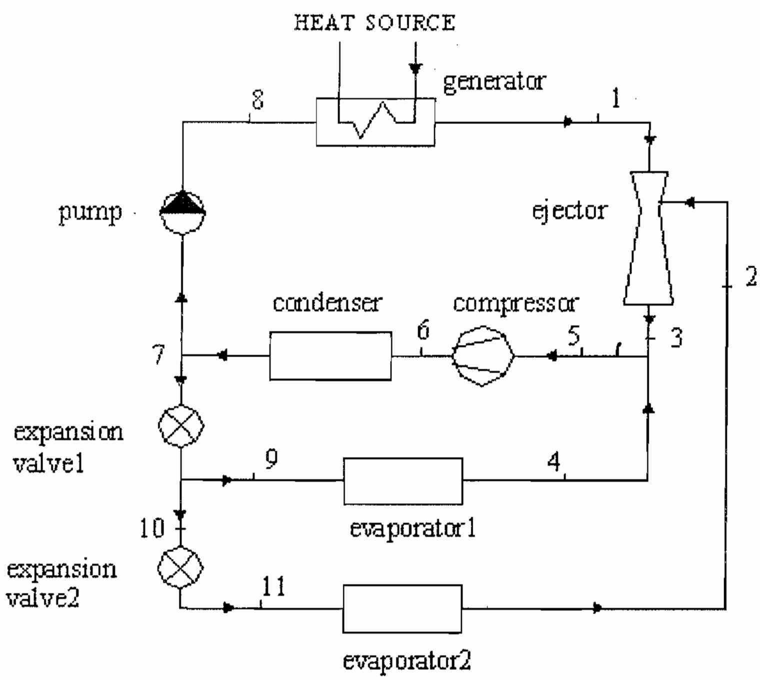

Figure1. Schematic view of the system

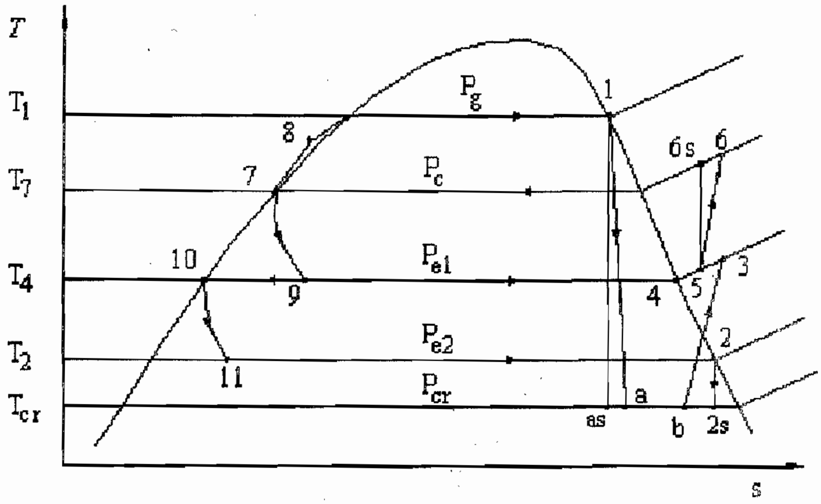

Figure2. T-s chart of the thermodynamic cycle

cooling medium. The obtained condensates (point7) divide themselves into two flows. The first quantity, $\dot{m}_1$ is sucked by the pump and discharged to state (8) at the pressure P_g , while the second undergoes a primary expansion up to a pressure of P_{e1}. At this state, a saturated mixture of liquid and vapour is formed. One part of the mass flow rate $\dot{m}_4$ (point9) is injected into evaporator1 and the rest $\dot{m}_2$ is collected (point10) and expanded up to a pressure, P_{e2} which corresponds to the temperature of evaporation $T_{e2} = T_2$ as represented in the T-s diagram. After evaporation, the vapours (considered dry saturated) are at states (4) and (2) respectively. The cycle closes and starts all over again.

III- DESCRIPTION OF THE EJECTOR AND ANALYSIS OF THE CYCLE

The ejector is the heart of an ejection refrigerating system and the performances of such a system depend strongly on its behaviour [6]. Figure3 shows the schematic of the ejector. This device is made up of a Laval's nozzle, a convergent, a mixing chamber of constant cross section and a diffuser (where the mixture is compressed and discharged). The mass flow rate in states (1), (2) and (3) is stationary.

It is considered in this paper that, the expansion of the motive vapour in the nozzle of the ejector is accompanied by an energy loss and that the associated entropy production (transformation (1)-(a) in figure2) corresponds to an isentropic efficiency, $\eta_a = 0.8$. On the other hand, the mixture of the two streams takes place at the critical pressure ($P_{cr} = 0.62$ bar) at point (a), and this pressure corresponds to the critical velocity of the secondary flow [6]. After this, the supersonic mixture evolves in the convergent (b)-(c) at a decreasing velocity up to the entrance of the mixing chamber where shock occurs. This phenomenon is accompanied by an increase in entropy (energy loss due to wave shock and viscosity flows). Later, the compressed mixture goes through the diffuser at a subsonic velocity up to the exit (3) where the flow becomes stationary again.

It should be admitted that, between the mixing zone (b) and the exit of the ejector (3), the mixture

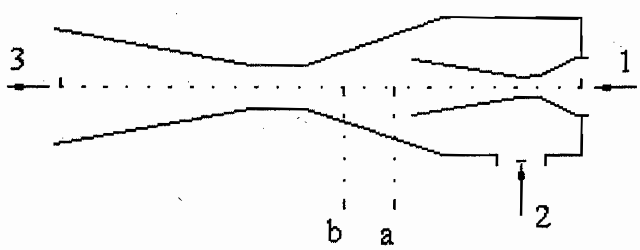

Figure3. schematic view of the ejector

of the two flows undergoes an energy loss corresponding globally to an isentropic efficiency, $\eta_b = 0.8$.

From the principle of fluid momentum conservation in the mixing section, we have:

$$V_{a1} + wV_{a2} = (1+w)V_b \qquad (1)$$

with $w =$, (2)

the energy balance equation of the ejector is written as follows :

$$h_1 + wh_2 = (1+w)h_3 \qquad (3)$$
$$h_1 = h(P_g,T_1) \qquad (4)$$
$$h_2 = h(P_{c2},T_2) \qquad (5)$$

Different equations can be written for each of the sections of the ejector.

-For the nozzle, region (1)-(a), the energy balance equation gives:

$$= h_1 - h_{a1} \qquad (6)$$

The specific enthalpy h_{a1} is determined from the following equations:

$$s_1 = s(P_g,T_1) = s_{as} = x_{as}.s_{gas} + (1- x_{as})s_{\ell as} \qquad (7)$$
$$h_{as} = x_{as}h_{gas} + (1- x_{as})h_{\ell as} \qquad (8)$$
$$\eta_a = (h_1-h_{a1})/(h_1-h_{as}) \qquad (9)$$

- intake pipe: the energy balance equation between points (2) and (2a) is *:*

$$= h_2 - h_{a2} \qquad (10)$$

The specific enthalpy h_{a2} at point (a2) is determined from equations:

$$s_2 = s(P_{c2},T_2) = s_{a2} = x_{a2}.s_{ga2} + (1- x_{a2})s_{\ell a2} \qquad (11)$$
$$h_{a2} = x_{a2}h_{ga2} + (1- x_{a2})h_{\ell a2} \qquad (12)$$

- converging cone (b)-(c) : By fixing a value for w ($0<w<1$), the specific enthalpy h_3 can further be determined from equation (3).

The energy balance equation between points (b) and (3) gives:

$$= h_3 - h_b \qquad (13)$$

The specific enthalpy h_b at the point (b) is calculated from equations (1) and (13).

-The region between section (b) and the exit of the diffuser (3).

The temperature T_3 of the compressed vapours is determined from the equations below:

$$s_b = s(T_b,P_{cr}) \qquad (14)$$
$$s_{3s} = s_b = s(T_{3s},P_{c1}) \qquad (15)$$
$$h_{3s} = h(T_{3s},P_{c1}) \qquad (16)$$
$$\eta_b = (h_{3s}-h_b)/(h_3-h_b) \qquad (17)$$
$$h_3 = h(T_3, P_{c1}) \qquad (18)$$

This new value of h_3 is then compared with that calculated earlier from equation (3). Using iterative calculations, the maximum value of the ratio w can then be found as a function of different parameters at the points (1), (2) and (3) of this ejector.

-Points of the cycle.

The points (1), (4), (7) and (10) are determined as a function of their respective saturation temperatures. The isenthalpic expansions in the expansion valve1 and valve 2 give respectively:

$$h_9 = h_7 \qquad (19)$$
$$h_{11} = h_{10} \qquad (20)$$

The point (6) is determined by supposing that, the vapours of the fluid refrigerant admitted into the compressor at point (5) undergo a polytropic compression with an increase in entropy with an efficiency, $\eta_{co} = 0.85$. Hence the relations:

$$\eta_{co} = (h_{6s}-h_5)/(h_6-h_5) \qquad (21)$$
$$s_{6s} = s_5 = (T_5,P_{c1}) \qquad (22)$$
$$h_{6s} = h(T_{6s,}P_c) \qquad (23)$$
$$h_6 = h(T_6,P_C) \qquad (24)$$

The characteristics of point (5) are known from the consideration of the energy balance equation of the vapour mixtures from point (4) and the ejected stream at point (3),

$$h_5 = \quad h_3 + \quad h_4 \qquad (25)$$

and also the equation of mass conservation:

$$= \quad + \qquad (26)$$

Considering the equation of mass conservation in the ejector:

$$= \quad + \quad = \quad (1 + w) \qquad (27)$$

and combining equations (26) and (27), equation (25) can be rewritten as :

$$(1+ w + w \quad) h_5 = w \quad h_4 + (1+w)h_3 \qquad (28)$$

Table1. System's characteristics at Tc=45°C

k	T_g	T_{e1}	T_{e2}	w	COP
0.5	85	-5	-25	0.6	0.631
	90			0.605	0.664
	95			0.61	0.655
	100			0.6	0.704
1	85	-5	-25	0.6	0.758
	90			0.605	0.767
	95			0.61	0.785
	100			0.6	0.842
2	85	-5	-25	0.6	0.983
	90			0.605	0.993
	95			0.61	1.014
	100			0.6	1.079
3	85	-5	-25	0.6	1.17
	90			0.605	1.185
	95			0.61	1.207
	100			0.6	1.278

Table2. System's characteristics at Tc=50°C

k	T_g	T_{e1}	T_{e2}	w	COP
0.5	85	-5	-25	0.6	0.625
	90			0.605	0.633
	95			0.61	0.649
	100			0.6	0.699
1	85	-5	-25	0.6	0.74
	90			0.605	0.75
	95			0.61	0.768
	100			0.6	0.843
2	85	-5	-25	0.6	0.944
	90			0.605	0.954
	95			0.61	0.974
	100			0.6	1.037
3	85	-5	-25	0.6	1.111
	90			0.605	1.124
	95			0.61	1.147
	100			0.6	1.213

Denote $$k=\frac{\dot{m}_4}{\dot{m}_2} \qquad (29)$$

h_5 can be determined from relation (28) and consequently point (5) for a fixed value of k.

In conformity with the law of energy conservation, we can write for each component of the cycle the following equations,

Evaporator1: $\dot{Q}_{e1} = \dot{m}_4\ (h_4\text{-}h_9)$ (30)

Evaporator2: $\dot{Q}_{e2} = \dot{m}_2\ (h_2\text{-}h_{11})$ (31)

Heat Generator: $\dot{Q}_g = \dot{m}_1\ (h_1\text{-}h_8)$ (32)

Refrigerant pump: $\dot{W}_p = \dot{m}_1\ (h_8\text{-}h_7)$ (33)

Mechanical compressor: $\dot{W}_{co} = \dot{m}_5\ (h_6\text{-}h_5)$ (34)

Coefficient of performance of the system:

$$COP=(\dot{Q}_{e1} + \dot{Q}_{e2})/(\dot{W}_p + \dot{W}_{co} + \dot{Q}_g) \qquad (35)$$

From equations (2), (29), (30).. (33), the expression for COP becomes :

$$COP=w\ \frac{k.(h_4-h_9)-(h_2-h_{11})}{(h_1-h_7)-(1+w+k.w)(h_6-h_5)} \qquad (36)$$

Remark: The mass flow rate ratio, k in evaporator 1 and evaporator 2 is proportional to the ratio of the refrigerating power of these evaporators which is in conformity with equations (29), (30) and (31).

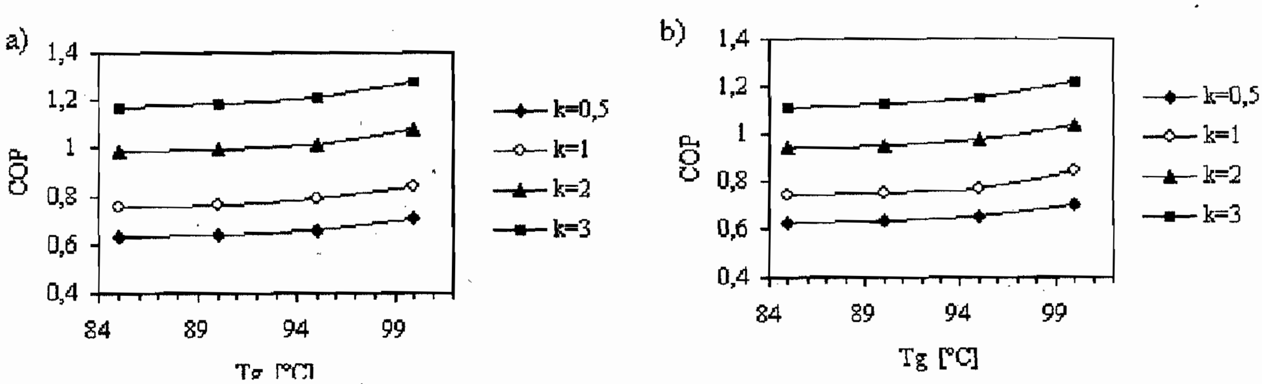

Figure4. Behavious of the coefficient of performance
a) Tc=45°C w=0,6 b) Tc=50°C w=0,6

Keeping the values of k fixed, the solution of the above equations was obtained for a range of generator temperatures. Table1 and table2 show the results for the condensation temperatures t_c= 45°C and t_c= 50°C respectively. The data in table1 and table2 were used to construct the respective curves that illustrate the behaviour of COP as a function of the temperature level of the thermal source for different refrigerating production ratios at the adopted temperature levels.

IV- DISCUSSION OF RESULTS

The curves (a) and (b) of figure 4 are analysed from the fact that, the heat used by replacing the mechanical (or electric) energy for the working of the ejector can only be justified by the :

- Amelioration of the efficiency of the ejector,
- Quantitative and qualitative availability of thermal energy,
- Amelioration of the total performance of the combined cycle which is characterized by its COP and by the ratio of the reduction of the expected mechanical energy consumption.

According to curves (a) and (b) of figure4, the COP increases with the level of the generator temperature T_g in a way that makes it difficult to predict its maximum. For a fixed level of generator temperature, the performances of the system increase with an increase in the mass flow rate ratio in evaporator 1 and evaporator2 (that is when the proportion in mass flow rate is such that the low-temperature refrigerating power is found to be even lower and consequently, the stream flow rate sucked by the ejector to be even lesser). This shows that the ejector has the tendency to reduce the performances of the system, while the compressor tries to raise them.

On the other hand, all things being equal, the effectiveness of the system falls when the condensation temperature goes from 45°C to 50°C. The increase in the condensation temperature and consequently that of the surrounding also gives a negative influence to the performances of the system. From the data in table1, the COP attains the value 1.278 at a generator temperature of 100°C. This is largely above the value one could expect for an ejection refrigerating system, because according to Charles A. Garris et al.[2], this system becomes competitive compared to compression systems only if their COP attains at least 0.75.

V- CONCLUSION

In the scope of this paper, a hybrid model of a double temperature refrigeration system susceptible to function with the help of a solar heat source was presented and analysed. The analysis of the behaviour of the performance of the obtained cycle showed a net amelioration. This theoretical study should be completed by an experimental prototype permitting the validation of the obtained results, and could permit the envisagement of applications in cold production industries in the tropics, where solar radiation (of a high density) can easily be collected and finally supplied to the generator.

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