SIMULATION AND ECONOMIC OPTIMIZATION OF A SOLAR ASSISTED COMBINED EJECTOR – VAPOR COMPRESSION CYCLE FOR COOLING APPLICATIONS

Humberto Vidal
University of Magallanes
P.O Box 113-D
Punta Arenas, Chile
rvidal@ona.fi.umag.cl

Sergio Colle
Federal University of Santa Catarina
P.O Box 88040-900
Florianópolis, SC Brazil
colle@emc.ufsc.br

ABSTRACT

This paper describes the hourly simulation and optimization of a thermally driven cooling cycle assisted by solar energy. The double stage solar ejector cooling cycle is modeled using the TRNSYS-EES simulation tool and the typical meteorological year file containing the weather data of Florianópolis, Brazil. The first stage is performed by a mechanical compression system with R-134a as the working fluid, while the second stage is performed by a thermally driven ejector cycle with R-141b. Flat plate collectors and an auxiliary energy burner provides heat to the ejector cycle. The thermoeconomical optimization is carried out with respect to the intercooler temperature and the flat plate solar collector area, for given specific costs of the auxiliary energy and electric energy, the capital cost of the collectors, ejector cooler, and the capital cost of equivalent mechanical compression cooler. Upper bounds for economical feasibility in terms of the costs of the auxiliary energy and electric energy are also presented.

1. INTRODUCTION

In the field of refrigeration technologies, the solar-driven ejector refrigeration system appears like an attractive alternative to use low temperature heat supply. The major components in that system include solar collectors, hot water storage tank, a combined ejector-vapor compression cycle and an auxiliary burner as shown in Fig. 1. The collector pump circulates water between the collector and the storage tank. The water carries heat from the collector and releases it in the storage tank. Then, the hot water is pumped from the storage tank to the generator which vaporizes the refrigerant. In case the heat delivered from the storage tank is not enough to drive the cycle, additional heat is produced in the auxiliary burner and delivered to the R-141b so that the pressure and temperature conditions required by the ejector are guarantied. At the same time, the vapor leaving the intercooler enters into the ejector and results a mixed stream that is discharged into the condenser. This saturated liquid is divided into two streams; one goes into the pump where is pumped back to the generator and the other goes to the expansion valve. It is then expanded to the intercooler where it is evaporated by the heat rejected from the vapor mechanical compression cycle. At the mechanical subsystem, the compressed R-134a vapor coming from the compressor is condensed in the intercooler. This condensate undergoes a pressure reduction in the throttling valve and then enters the evaporator where it is evaporated to produce the necessary cooling effect. The vapor is finally compressed to a higher pressure by the compressor and then enters to the intercooler, thus completing the combined cycle.

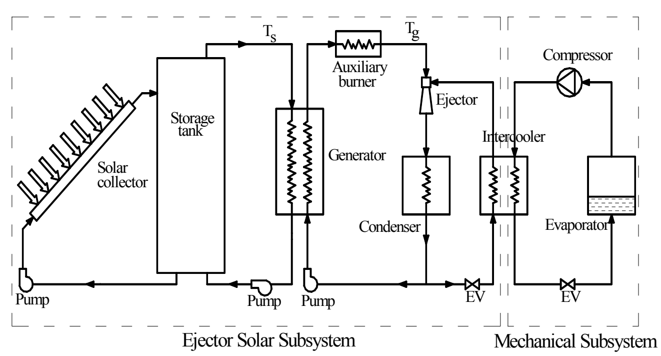

Fig. 1: Solar energy assisted combined ejector-vapor compression system.

In some studies reported in the literature [5,8], it has been assumed constant solar irradiation incident on the tilted solar collector, equal to the average of the incident solar radiation in the year. Since solar irradiation and ambient temperature normally vary considerably during the year from place to place, this variability should be taken into account in order to reproduce the real operation conditions of the solar cooling system. The main objective of this work is to develop a computational model to carry out an hourly simulation of a solar assisted combined ejector-vapor compression cooling system. For the simulation of the system, the well-known computer code TRNSYS [6] is employed. However, the TRNSYS library has not an "ejector cooling cycle" component. Therefore, a TRNSYS component based in the mathematical model developed in [4] that represents the performance of a one-dimensional ejector is written. The "combined ejector cooling cycle" component is built with the computational program EES [7], by using the TRNSYS component Type 66. Finally, the economical optimization of a solar assisted combined ejector-vapor compression using the results obtained with the hourly simulation of the system with TRNSYS is carried out. The economical optimization is carried out with respect to the specific collector area, a_c , and the intercooler temperature, T_e for determine the conditions under which a double stage ejector solar refrigeration system cycle can be economically more attractive than a refrigeration system by vapor mechanically compression for the Florianópolis city.

2. SYSTEM DESCRIPTION

2.1 TRNSYS components modelling

Each TRNSYS component is modeled using mathematical equations which are written in FORTRAN. This way, if some component of the system is not included in the TRNSYS library, its physical model can be programmed in FORTRAN or EES. The main components used in TRNSYS to modelling the solar ejector cooling system are shown in the Fig. 2.

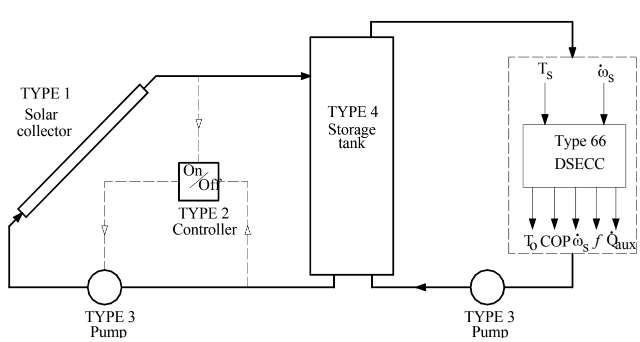

Fig. 2: Main components of TRNSYS model.

A weather data file of a typical meteorological year for Florianópolis [1] is used to model the long-term performance of the solar ejector refrigeration system. The TRNSYS unit Type 66, allows the user to call an EES file, receive data information from TRNSYS component (T_s, $\dot{\omega}_s$) and pass it output data to the others components TRNSYS (T_o , COP, $\dot{\omega}_s$, f, $\dot{Q}_{aux}$), as shown in the Fig. 2. This component is used to host the EES program that contains the model of double stage ejector cooling cycle, called DSECC shown in the Fig. 3.

2.2 DSECC components modelling

The generator of the ejector subsystem is the connection point between solar and combined cooling cycle. In the configuration shows in the Fig.1, the solar heat that drives the refrigeration system it is determined by the operation temperature, T_s (storage tank outlet) which depends on incident solar radiation and thermal losses. In simulation models found in the literature T_s is set equal to T_f , this mean that ideal heat exchanger condition is assumed [8]. In other works, T_s is considered to be 10 °C higher than T_f [5]. However, it should be noticed that the solar fraction f defined as $\dot{Q}_s / \dot{Q}_g$ will depend on the heat transfer process which takes place with phase-change in the heat exchanger and therefore will also depend on the outlet refrigerant temperature, T_f . If the refrigerant vapor does not reach the quality of saturated vapor, an auxiliary heater needs to be considered, as schematically presented in Fig. 3.

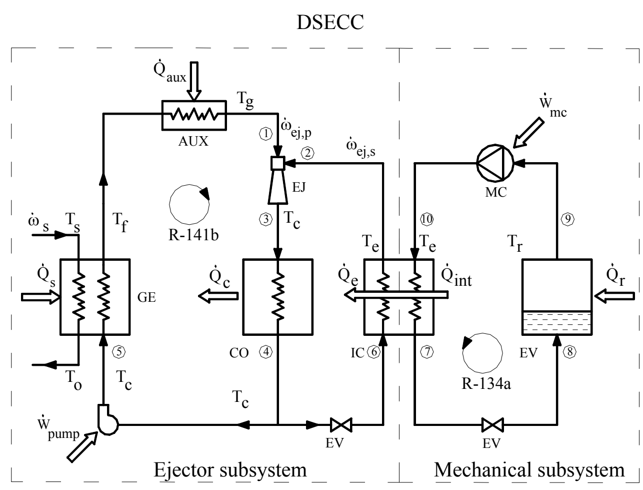

Fig. 3: Ejector cooling cycle diagram.

The maximum $\dot{Q}_s$ to which solar fraction is unitary is $\dot{Q}_g = \dot{Q}_e / COP$, where the COP is calculated for fixed and specified temperatures for the vapor flow in the generator, condenser and evaporator. In the heat transfer process of the ejector cycle generator, the temperature T_s varies with the energy gains and losses of the system and determines the different heat transfer regimes. The governing equations presented here are reported in [2] as follows:

Case I: Sensible Heat Region ($T_f < T_g$)
In this case, the refrigerant fluid temperature T_f at the heat exchanger output is less than the vapor generator temperature T_g . Therefore, the heat exchange is sensible and can be written as

$$\dot{Q}_s = W_{\min}\varepsilon_s\left(T_s - T_c\right) = \dot{\omega}_{ej,p}\,c_{rl}\left(T_f - T_c\right) \tag{1}$$

where $\varepsilon_s = \dfrac{\dot{\omega}_{ej,p}\,c_{rl}\left(T_f - T_c\right)}{W_{\min}\left(T_s - T_c\right)}$ is the heat exchanger effectiveness definition and $W_{\min} = \min\left\{\left(\dot{\omega}\,c_p\right)_s, \dot{\omega}_{ej,p}\,c_{rl}\right\}$ where $\left(\dot{\omega}\,c_p\right)_s$ is the heat capacity of the fluid by the solar system side, c_{rl} is the refrigerant specific heat at T_c and $\dot{\omega}_{ej,p}$ is the stationary mass flow in the vapor generator of the ejector cycle. The maximum value of T_s when $T_f = T_g$ is obtained from Eq. (1) and given by

$$T_{sl} = T_c + \frac{\dot{\omega}_{ej,p}c_{rl}\left(T_g - T_c\right)}{W_{\min}\varepsilon_s} \tag{2}$$

Therefore, if T_s remains less than T_{sl} , $\dot{Q}_s$ must be calculated by Eq. (1). The numerical process is continued until the temperature T_s reaches T_{sl} .

Case II: Two-phase regime ($T_f = T_g$)
In this case, the refrigerant fluid experiences a phase-change and the heat can be written as

$$\dot{Q}_s = \dot{\omega}_{ej,p}\left(h_f - h_c\right) \tag{3}$$

where $h_f = h_f\left(T = T_g, x = x_f\right)$, $h_c = h_c\left(T = T_c, x = 0\right)$ and x_f is the vapor quality that is determined as follow

$$x_f = \left(T_g - T_c\right)\left(\dot{\omega}\,c_p\right)_s\left(\frac{\dot{\omega}_{ej,p}\,c_{rl}}{\varepsilon_s W_{\min}} - 1\right)\left[1 - \exp\left(\frac{-U_{ev}A_{ev}}{(\dot{\omega}\,c_p)_s}\right)\right] \Bigg/ \dot{\omega}_{ej,p}h_{lv}\exp\left(\frac{-U_{ev}A_{ev}}{(\dot{\omega}\,c_p)_s}\right) \tag{4}$$

Using the Eq. (1) and (3), the return hot water temperature of the solar system is determined as follow

$$T_o = T_s - \frac{\dot{Q}_s}{(\dot{\omega}\,c_p)_s} \tag{5}$$

2.2.1 <u>Modelling of the ejector cooling cycle</u>

Assumptions for modelling:
- The system operates at steady state.
- Pressure losses in all the components and the connecting pipes are negligible.
- Heat losses to the ambient are negligible except for the components requiring energy exchange with the environment.
- The working fluid R-141b at the exits of the generator, evaporator and ejector is at saturated vapor state.
- The exit of condenser is at saturated liquid state.
- The temperature rise across the circulation pump is negligible, $h_4 = h_5$.
- The expansion through the expansion valve is a throttling process, $h_4 = h_6$.
- A counter flow arrangement heat exchanger is considered.

The COP of the ejector cooling cycle can be derived as

$$COP = \frac{\dot{Q}_e}{\dot{Q}_g} \tag{6}$$

where $\dot{Q}_e = \dot{\omega}_{ej,s}\left(h_2 - h_6\right)$ and $\dot{Q}_g = \dot{\omega}_{ej,p}\left(h_1 - h_5\right)$
$h_2 = h_2\left(T = T_e, x = 1\right)$; $h_6 = h_4 = h_4\left(T = T_c, x = 0\right)$
$h_1 = h_1\left(T = T_g, x = 1\right)$; $h_5 = h_4 = h_4\left(T = T_c, x = 0\right)$

Eq. (6) can alternatively be expressed by

$$COP = \zeta\,\frac{h_2 - h_6}{h_1 - h_5} \tag{7}$$

where ζ is the entrainment ratio, defined as the ratio of flow rates of secondary, $\dot{\omega}_{ej,s}$, to primary vapor $\dot{\omega}_{ej,p}$. The solar fraction f is defined as follows:

$$f = \frac{\dot{Q}_s}{\dot{Q}_g} \tag{8}$$

where $\dot{Q}_s$ is calculated according to the heat exchange regime in the generator, explained in the cases *I* and *II*. Once the solar fraction is known, the auxiliary heat can be evaluated by the following expression:

$$\dot{Q}_{aux} = \left(1 - f\right)\dot{Q}_g \tag{9}$$

2.2.2 <u>Modelling of the mechanical cooling cycle</u>

Assumptions for modelling:
- In the compressor, an adiabatic reversible compression process is considered.

- The working fluid R-134a at the exits of the evaporator is at saturated vapor state.
- The exit of intercooler is at saturated liquid state.
- The expansion through the expansion valve is a throttling process, $h_7 = h_8$.
- At the intercooler, the ideal heat exchange condition is assumed.

Heat flow rate at the evaporator:

$$\dot{Q}_r = \dot{\omega}_{fr}\left(h_9 - h_8\right) \tag{10}$$

where $h_9 = h_9\left(T = T_r, x = 1\right)$; $h_8 = h_7 = h_7\left(T = T_e, x = 0\right)$

Mechanical power required in the compressor:

$$\dot{W}_{cm} = \dot{\omega}_{fr}\left(h_{10} - h_9\right) \tag{11}$$

where $h_{10} = h_9 + \dfrac{h_{10i} - h_9}{\eta_s}$; $h_{10i} = h_{10i}\left(P = P_{10}, s = s_{10}\right)$;
$P_{10} = P_7 = P_7\left(T = T_e, x = 0\right)$; $s_{10} = s_9 = s_9\left(T = T_r, x = 1\right)$

Heat flow rate at the intercooler:

$$\dot{Q}_{\text{int}} = \dot{\omega}_{fr}\left(h_{10} - h_7\right) \tag{12}$$

Using the ideal heat exchange condition and from an energy balance follows:

$$\dot{Q}_{\text{int}} = \dot{Q}_e \tag{13}$$

$$\dot{\omega}_{fr}\left(h_{10} - h_7\right) = \dot{\omega}_{ej,s}\left(h_2 - h_6\right) \tag{14}$$

Once the T_g , T_c , T_e , T_r , $\dot{Q}_r$ is known and using the equation (14), $\dot{\omega}_{ej,s}$ can be evaluated. On the other hand with T_g , T_c , T_e , the ratio of flow rates at the ejector ζ can be determined and this way the $\dot{\omega}_{ej,p}$ value calculated.

3. THERMOECONOMIC OPTIMIZATION

The lifetime cost saving function for the double stage cooling system shown in Fig. 1 is proved to be given by,

$$\begin{aligned} LCS = {} & P_1 Q_r C_{E1}\left(\frac{1}{COP_{el}} - \frac{1}{COP_m}\right) \\ & - P_1 Q_r C_{F1}\left(1 - f\right)/COP - P_2 C_A A_C \\ & + P_2\left(C_{EL} - C_M - C_{ej} - C_E\right) \end{aligned} \tag{15}$$

The first term in the Eq. (15) is the present value of the difference between the operational cost of a mechanically driven cycle (MDC) equivalent with a given COP_{el}, and the operation cost due to the first stage of the combined cycle. Here, P_1 is the present worth factor PWF (i_F, i_d, N_e) described in the P_1 - P_2 method [3], N_e is the time period (in years) of the economical analysis, i_F, and i_d are the inflation and the discount rate of the fuel cost, respectively, and C_{E1} is the electric energy cost (US$ / GJ). The second term is the present value of the cost of the auxiliary heating of a thermally driven cycle (TDC) with specific cost C_{F1} (US$ / GJ). The third term gives the capital cost due to the collector area. The last term gives the difference among the capital cost, C_{EL}, of an equivalent MDC with COP_{el}, the capital cost of the first stage MDC, C_M, with COP_m , the capital cost of the TDC, C_{ej}, and the cost independent of the collectors area, C_E. P_2 is an economical factor that takes into an account the cost of the investments, insurance, collector resale value and state and federal taxes, as described in [3]. C_A is the collector cost per unit area (US$ / m^2) and f is the annual fraction of the solar energy. The Eq. (15) can alternatively be expressed as follows

$$\begin{aligned} \ell = {} & \alpha_E\left(\frac{1}{COP_{el}} - \frac{1}{COP_m}\right) \\ & - \alpha_F\left(1 - f\right)/COP - a_c + d/C_A \end{aligned} \tag{16}$$

where $\ell = LCS / P_2\, C_A\, Q_r$, $\alpha_E = P_1\, C_{E1} / P_2\, C_A$, $\alpha_F = P_1\, C_{F1} / P_2\, C_A$, $d = (C_{EL} - C_M - C_{ej} - C_E) / Q_r$, and $a_c = A_c / Q_r$.

By assuming only the case for which $\ell = 0$, from equation (16) it follows

$$\alpha_E\left(\frac{1}{COP_{el}} - \frac{1}{COP_m}\right) + \frac{d}{C_A} - a_c = \frac{\alpha_F\left(1 - f\right)}{COP} \tag{17}$$

The above inequality shows that the specific area a_c is bounded by some maximum specific area a_{max} defined as

$$a_{\max} = \alpha_E\left(\frac{1}{COP_{el}} - \frac{1}{COP_m}\right) + \frac{d}{C_A} \tag{18}$$

Taking the partial derivative of ℓ given by equation (16) with respect to a_c to vanish it follows,

$$\alpha_F = COP / \frac{\partial f}{\partial a_c} \tag{19}$$

For the bound-case corresponding to $\ell = 0$ equation (16) can be written as

$$a_{\max} - a_c = \alpha_F\left(1-f\right)/COP \tag{20}$$

Replacing α_F from equation (19) into equation (20) it leads to

$$\left(a_{\max} - a_c\right)\frac{\partial f}{\partial a_c} = 1 - f \tag{21}$$

For each specified value of a_{max} and a given temperature T_e, equation (21) can be solved in terms of a_c and therefore the loci corresponding to $\ell = 0$ and $\partial\ell/\partial a_c = 0$ can be plotted as a function of the parameters α_F and a_{max} , for a fixed value of d (Fig. 7). The Eq. (16) can be rewritten as

$$\ell = \alpha_E\ /COP_{el} - \psi - a_c + d/C_A \tag{22}$$

where ψ is given by

$$\psi = \alpha_E\ /COP_m + \alpha_F\left(1-f\right)/COP \tag{23}$$

Taking the partial derivative of ℓ with respect to T_e in equation (22) it leads to

$$\frac{\partial\psi}{\partial T_e} = \frac{-\alpha_E}{COP_m^2}\frac{\partial COP_m}{\partial T_e} - \frac{\alpha_F}{COP}\left[\frac{\partial f}{\partial T_e} + \frac{\left(1-f\right)}{COP}\frac{\partial COP}{\partial T_e}\right] \tag{24}$$

By making the derivative given above to vanish, the optimum value for T_e can thus be found. On the other hand, using the $\partial\psi/\partial T_e = 0$ condition, α_E can be expressed as a function of α_F as follows

$$\alpha_E = \frac{\alpha_F\, COP_m^2\left[\dfrac{\partial COP}{\partial T_e}\left(f-1\right) - \dfrac{\partial f}{\partial T_e}COP\right]}{\dfrac{\partial COP_m}{\partial T_e}COP^2} \tag{25}$$

The thermoeconomical optimization of the system resulting in the combination of solar energy and auxiliary energy of lower cost requires knowledge of the solar fraction f. The method used for estimate f depends of the climatic data availability. If an hourly database is available then a dynamic simulation is recommended. On the other hand, if a monthly average daily database is available the $f-\overline{\phi}$ chart method modified for the ejector cooling cycle [2] can be used. In the present case, both databases are available. A comparison between two methods in terms of f for the particular case analyzed here is shown in Fig. 4. A good agreement is found for f but not for the derivation of f with respect to a_c . Therefore, the $f-\overline{\phi}$ chart method fitted for ejector cooling cycle is shown to be a good approach to determine the solar fraction.

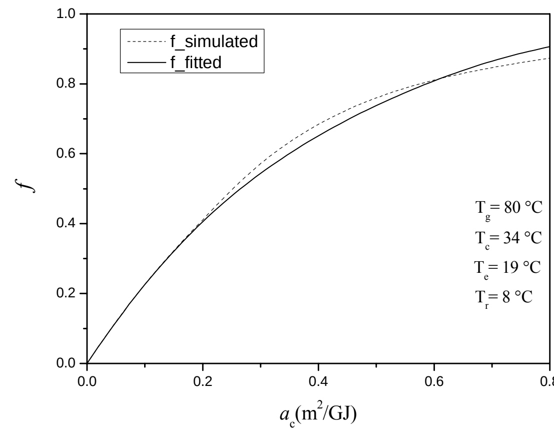

Fig. 4: Solar fraction for T_e =19 °C

However, when the critical LCS condition is examined using equation (21), a significant disagreement is obtained as can be verified in the Fig. 5.

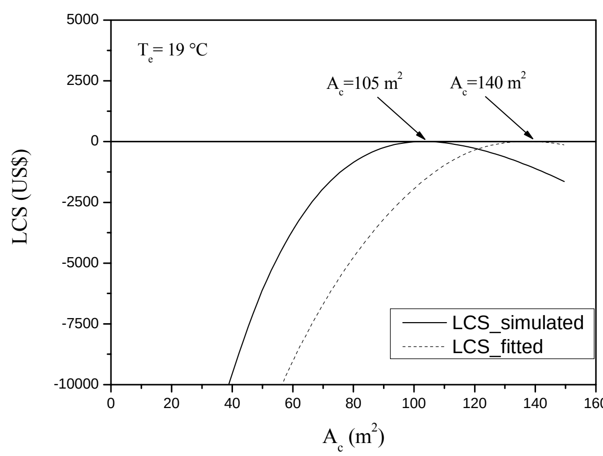

Fig. 5: LCS for T_e =19 °C

In the present optimization process, the solar fraction is calculated using hourly simulations of the double stage solar ejector refrigeration system modeled with the TRNSYS-EES numerical coupling. A large number of TRNSYS simulations are carried out in order to find the optimum values for a_c and T_e . After this optimization process, the Fig. 6 is obtained. This plot represents the solution of Eq. (21) in terms of a_c , for given values of T_e, and the solution of Eq. (24) in terms of T_e, for given values of a_c . The optimum point illustrated in the Fig. 6, corresponds to a specific area a_c = 0.63 m²/GJ (corresponding to 105 m² of collector area) and an intercooler temperature of 19 °C.

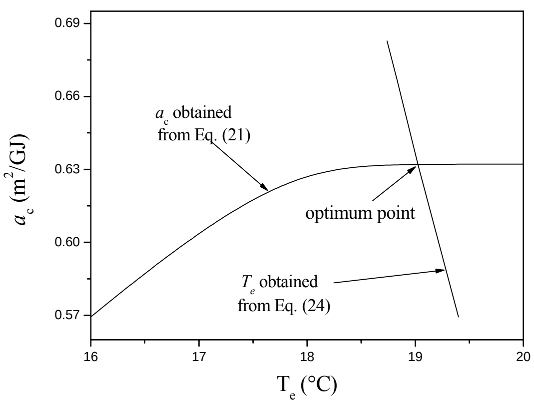

Fig. 6: Optimum solution of the system.

The Fig. 7 illustrates the solution of Eq. (21) for an optimum value of T_e and the ℓ = 0 condition where $h = (a_{\max} - a_c)\partial f / \partial a_c$.

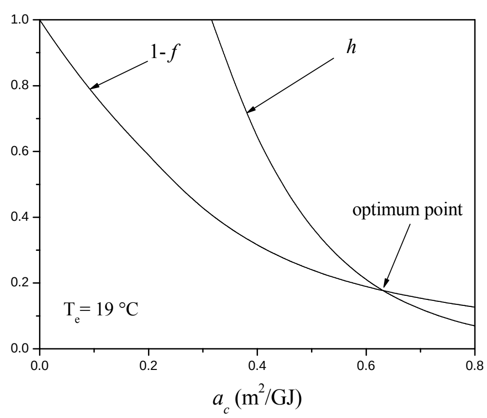

Fig. 7: Optimum solution of a_c .

In the Fig. 8, points on the right and down the curve of LCS correspond to the economically feasible regions.

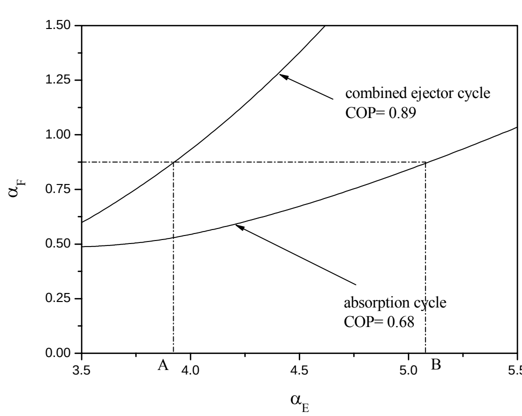

Fig. 8: Bound curves of LCS=0 as a function of α_E.

Therefore, the economical feasible region corresponding to the combined ejector cycle is larger than the feasible region corresponding to an absorption cycle for the same cooling capacity. Also can be seen from Fig.8 that the lesser the value of COP the greater the required electricity cost in order to reach a feasible point, as expected.

4. CONCLUSIONS

A TRNSYS-EES computational model of a solar assisted combined ejector-vapor compression system has been developed to perform a thermoeconomical optimization in order to determine the conditions under which the combined ejector refrigeration system may be economically competitive with an equivalent conventional refrigeration system. The final optimized system for a 10.5 kW cooling capacity consists of 105 m^2 of flat plate collector and an intercooler temperature of 19 °C resulting in a solar fraction of the system equal to 82% and a COP of the combined ejector cycle equal to 0.89 .

5. REFERENCES

(1) Abreu, S.; Colle, S.; Almeida, A.; Mantelli, S. Qualificação e recuperação de dados de radiação solar medidos em superfície em Florianópolis-SC. 8th Brazilian Congress of Thermal Engineering and Sciences, 2000.

(2) Colle, S.; Vidal, H.; Pereira, G. Limites de validade do método de projeto $f - \overline{\phi}$ chart para ciclos de refrigeração de ejetor assistidos por energia solar. Proceedings of CIES 2004, Vigo, Spain, 2004.

(3) Duffie, J. A.; Beckman, W. A. Solar engineering of thermal processes. 2nd Ed., New York, John Wiley & Sons, 1991.

(4) Huang, B. J.; Chang, J. M.; Wang, C. P.; Petrenko V. A. A 1-D analysis of ejector performance. International Journal of Refrigeration, Vol. 22, pp. 354-364, 1999.

(5) Huang, B. J.; Chang, J. M.; Petrenko, V. A.; Zhuk, K.B. Solar ejector cooling system using refrigerant R141b. Solar Energy, Vol. 64, Nos. 4-6, pp. 223-226, 1998.

(6) Klein, S. A., et al. TRNSYS 15 – Reference Manual 15 Ed. Madison; Solar Energy Laboratory, University of Wisconsin, Madison, 2000.

(7) Klein, S. A.; Alvarado F. L. EES- Engineering Equation Solver. F-Chart Software, Middletown, Wisconsin, 2000.

(8) Sokolov, M.; Hershgal, D. Solar-powered compression-enhanced ejector air conditioner. Solar Energy, Vol. 51, No. 3, pp. 183-194, 1993.