# THERMOECONOMIC EVALUATION AND OPTIMIZATION OF SOLAR ASSISTED THERMALLY DRIVEN COOLING CYCLES WITH IRREVERSIBILITY CONSTRAINT

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**Abstract** – The present work focuses the economical analysis and optimization of a double stage cooling cycle assisted by solar energy. The first stage is performed by a mechanical compression system with R134a as the working fluid while the second stage is performed by a jet nozzle cooling cycle with R114 as the working fluid. The $f-\bar{\phi}$ chart method is used as a tool to optimize solar collector area by maximizing the lifetime cost savings (LCS). The condition of optimization for both, the flat plate collector area and the intercooler temperature are set down, for given specific costs of the auxiliary energy and electric energy, the capital cost of the collectors, the jet nozzle cooler, and the capital cost of equivalent mechanical compression cooler. A second law analysis is also carried out in order to obtain the optimum condition for a given irreversibility, or exergetic efficiency of the cycle. The approach presented here might be useful to determine the condition under which, a double stage solar assisted cooling cycle can be economically competitive with a single stage mechanical compression cooling cycle, for fixed condenser and evaporator temperatures. It can also be used to determine the optimum intercooler temperature in a double stage cooling cycle, for given energy costs and capital costs.

## 1. INTRODUCTION

The increasing cost of the electricity generation and the increasing environmental restrictions against the fossil fuelled energy systems has given rise to the investigation of more environmentally friendly systems. Among these systems, thermally driven cooling systems assisted by solar energy, have proven to be in many respects economically attractive. These systems can either be driven by photovoltaic through mechanical compression cycles, or by thermal solar collector through absorption or ejector cycles. The coefficient of performance (*COP*) of a thermally driven cycle (TDC) is much smaller than the *COP* of mechanically driven cycles (MDC), for the same sink temperatures. Therefore, the cost of the driving energy of TDC has to be lower than the cost of the work to drive MDC, in order to reach competitiveness with respect to the MDC.

Cooling systems based on concentrating solar collectors with lithium-bromide absorption cycle are currently provided by many manufacturers. Flat plate collectors are shown to be advantageous as a mean of providing heat to these cycles, in the case of a high utilizability as reported in (Klein and Beckman, 1979), for Albuquerque - New Mexico. It has been shown in (Colle and Vidal, 2001) that flat plate collectors are economically attractive for lithium-bromide absorption cooling, with costs that are competitive with mechanically driven systems. The technology of ejector cooling cycles is presently being improved and investigated by many authors (Cizungu *et al*., 1999); (Huang et al., 1999); (Sun, 1997); (Sokolov and Hershgal, 1993); (Medina and Colle, 2001). The *COP* of ejector cycles are usually smaller than the *COP* of absorption cycles. On the other hand, the construction of the ejector system itself is simpler and less expensive in comparison to the absorption system. Ejectors perform well with many environmentally friendly working fluids as reported in (Huang et al., 1999).

Ejector cycles can be optimised with respect to the fluid flow ratio, the boiler temperature, and the solar collector area, as shown in (Sokolov and Hershgal, 1991). The optimization reported in (Sokolov and Hershgal, 1991) is carried out for a fixed value of the solar radiation incident on the tilted solar collector. However solar radiation and ambient temperature vary considerably from place to place and therefore these variability should be taken into account in optimization. The $f-\bar{\phi}$ chart method presented in (Klein and Beckman, 1979) was proposed to design solar heating and absorption cooling systems. The method is based on the concept of utilizability and therefore takes into account climatic factors variability. The $f-\bar{\phi}$ chart method can be used straightforwardly as a tool to optimize solar collector area by maximizing the lifetime cost savings (LCS). This method was used in (Colle and Vidal, 2001) to optimize the collector area of a solar assisted lithium-bromide absorption cycle. The results are compared with the case of an ejector cycle. Upper bounds for optimum feasible ranging parameters were also determined, as a function of the auxiliary energy cost and the electric energy cost.

The *COP* of the ejector cycle strongly decreases as the evaporator temperature decreases and therefore ejectors are less efficient for processes that require too low evaporator temperatures, as is the case of refrigeration applications. The *COP* of any cooling cycle decreases with the difference between of the temperatures of the two heat reservoirs. Low *COP* can be avoided and higher temperature difference can be achieved by a double stage cycle.

The present paper presents the economical evaluation and optimization of a double stage cooling cycle assisted by solar energy. The first stage is performed by a MDC operating between the evaporator and the intercooler while the second stage is performed by a TDC, which in the present case is an ejector cycle. The TDC operates between the intercooler and the condenser. The working fluid of the MDC is R134a while the working fluid of the TDC is R114. Flat plate collectors and an auxiliary burner provide heat to the TDC. The economical optimization is carried out with respect to the intercooler temperature and the solar collector area. The $f-\bar{\phi}$ chart method is used here, to correlate both, the solar collector parameters and the monthly means of the solar radiation incident on the tilted panels, with the fraction of solar energy. A previous version of this work is reported in a paper included in the proceedings of (Colle *et al*, 2002). In the mentioned paper, only the results corresponding to the Carnot cycle limit were presented. In the present paper, the analysis is carried out for a real ejector cycle and an ideal mechanical driven refrigeration cycle.

## 2. ECONOMICAL EVALUATION AND OPTIMIZATION

The double stage cycle under analysis is shown schematically in Fig.1. The energy collected by the solar collector is transferred to the TDC by means of a heat exchanger. Auxiliary heat is supplied to the heat exchanger whenever the solar heating system is unable to supply this heat at some minimum temperature $T_{min}$. For the particular case of the ejector cycle, all heat supplied at temperatures higher than the condenser temperature $T_c$ is useful. Therefore, the minimum temperature above which the utilizability of the solar collector is positive can be made equal to the condenser temperature $T_c$.

The *COP* of the TDC is defined by

$$COP_{th} = Q_e / Q_g \quad (1)$$

while the *COP* of the MDC is defined as

$$COP_m = Q_r / W_m \quad (2)$$

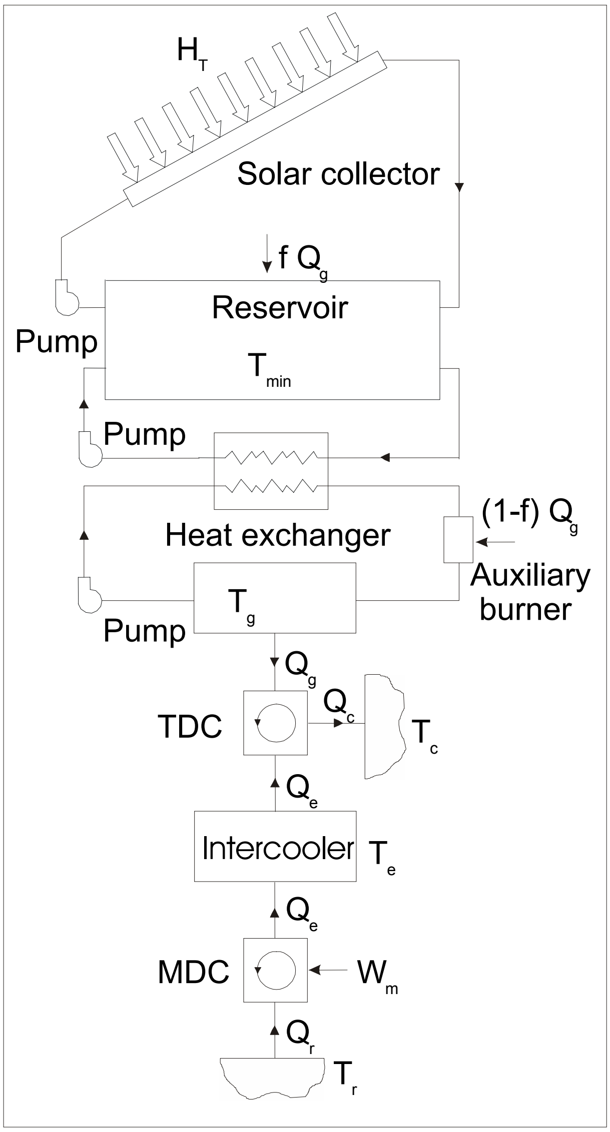


Figure 1. Schematics of the double stage cooling cycle.

where $W_m > 0$ is the mechanical work input to the cycle.

Since $Q_e = Q_r + W_m$, by equation (2) $Q_e = Q_r\left(1 + \frac{1}{COP_m}\right)$. Defining the *COP* of the double stage cycle by $COP = \frac{Q_r}{Q_g}$, the later expressions lead to

$$COP = COP_{th} COP_m / (1 + COP_m) \quad (3)$$

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

$$LCS = P_1 Q_r C_{E1}\left(\frac{1}{COP_{el}} - \frac{1}{COP_m}\right)$$
$$- P_1 Q_r C_{F1}(1-f)/COP - P_2 C_A A_C$$
$$+ P_2 (C_{EL} - C_M - C_{TH} - C_E) \quad (4)$$

The first term of the above equation is the present value of the difference between the operational cost of an MDC equivalent with a given $COP_{el}$, and the operation cost due to the first stage MDC. Here, $P_1$ is the present worth factor *PWF* ($i_F$, $i_d$, $N_e$) described in the $P_1$ - $P_2$ method (Brandemuehl and Beckman, 1979); (Duffie and Beckman, 1991), $N_e$ 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 the 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, $C_{TH}$, 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 (Brandemuehl and Beckman, 1979). $C_A$ is the collector cost per unit area (US$ / m$^2$) and $f$ is the annual fraction of the solar energy as given in (Klein and Beckman, 1979). The annual solar fraction is expressed in terms of the monthly heat input $Q_{gi}$ and the annual heat input $Q_g$ as follows

$$f = \sum_{i=1}^{12} f_i Q_{g_i} / Q_g \quad (5)$$

By replacing $Q_{g_i}$ and $Q_g$ in terms of $Q_{r_i}$ and $Q_r$ , respectively, as expressed by equation (1) it follows

$$f = \sum_{i=1}^{12} f_i Q_{r_i} / Q_r \quad (6)$$

According to (Klein and Beckman, 1979), the solar fraction $f_i$ is expressed by the following correlations,

$$f_i = \bar{\phi}_{max,i} Y_i - 0.015(e^{3.85 f_i} - 1)(1 - e^{-0.15 X_i})$$
$$\times (R_S)^{0.76} \quad (7)$$

In the present analysis the parameters $X_i$ and $Y_i$ are modified in terms of *COP* as follows

$$Y_i = A_C F_R (\tau\alpha)_n \left[\frac{(\overline{\tau\alpha})}{(\tau\alpha)_n}\right]_i \overline{H}_{T_i} N_i \, COP / Q_{r_i} \quad (8)$$

$$X_i = A_C (F_R U_L) 100 \Delta t_i \, COP / Q_{r_i} \quad (9)$$

where $\Delta t_i = 86400\, N_i$, $N_i$ is the number of days of month (i); $F_R U_L$ and $F_R(\tau\alpha)_n$ are the collector efficiency coefficients, $\overline{H}_{T_i}$ is the monthly average of the solar radiation incident on the tilted collector plate, and $R_S$ is the correction factor due to the ratio of the reservoir volume to the collector area, which is assumed here to be equal to the unity. The monthly fraction $f_i$ as well as its derivatives with respect to $A_c$ and $T_e$ are evaluated implicitly from equation (7).

$$\phi_{max,i} = \bar{\phi}(\overline{X}_{c\,min,i}) \quad (10)$$

$$\bar{\phi}(\overline{X}_c) = \exp\left[(a + bR_n / \overline{R})(\overline{X}_c + c\overline{X}_c^{\,2})\right] \quad (11)$$

where *a, b,* and *c* are functions of the average clearness index $\overline{K}_T$ for each month (i) as given in (Klein and Beckman, 1979) and (Duffie and Beckman, 1991).

$$\overline{X}_{c\,min,i} = F_R U_L (T_{min} - \overline{T}_{a_i}) / F_R(\tau\alpha)_n$$
$$\times \left[(\overline{\tau\alpha})/(\tau\alpha)_n\right]_i (r_{t,n} R_n \overline{H})_i \quad (12)$$

Equation (4) can alternatively be expressed as follows

$$\ell = \alpha_E \left(\frac{1}{COP_{el}} - \frac{1}{COP_m}\right)$$
$$- \alpha_F (1-f)/COP - a_c + d / C_A \quad (13)$$

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

By assuming only the case for which $\ell \geq 0$, from equation (13) it follows

$$\alpha_E \left(\frac{1}{COP_{el}} - \frac{1}{COP_m}\right) + \frac{d}{C_A} - a_c \geq$$
$$\alpha_F (1-f)/COP \geq 0 \quad (14)$$

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

$$a_{max} = \alpha_E \left(\frac{1}{COP_{el}} - \frac{1}{COP_m}\right) + \frac{d}{C_A} \quad (15)$$

Taking the partial derivative of $\ell$ given by equation (13) with respect to $a_c$ to vanish it follows,

$$\alpha_F = COP / \frac{\partial f}{\partial a_c} \tag{16}$$

For the bound-case corresponding to $\ell = 0$ equation (13) can be written as

$$a_{\max} - a_c = \alpha_F (1 - f)/COP \tag{17}$$

Replacing $\alpha_F$ from equation (16) into equation (17) it leads to

$$(a_{\max} - a_c)\frac{\partial f}{\partial a_c} = 1 - f \tag{18}$$

For each specified value of $a_{max}$ and a given temperature $T_e$, equation (18) 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 $\alpha_F$ and $a_{max}$ , for a fixed value of $d$. Alternatively, for each fixed value of $\alpha_F$ and $T_e$, the area ratio $a_c$ can be evaluated from equation (16). Replacing $a_c$ into equation (18), $a_{max}$ can be found and thus $\alpha_E$ can be obtained from equation (15), once $d$ is given. Therefore for each values of $\alpha_F$ and $d$, a value of $\alpha_E$ can be found from the curve for which $\ell = 0$. Typical curves of $\ell = 0$ are plotted in Fig. 3, while Fig. 2 illustrates a solution for $a_c$, for given values of $T_e$, $d$, and $\ell = 0$.

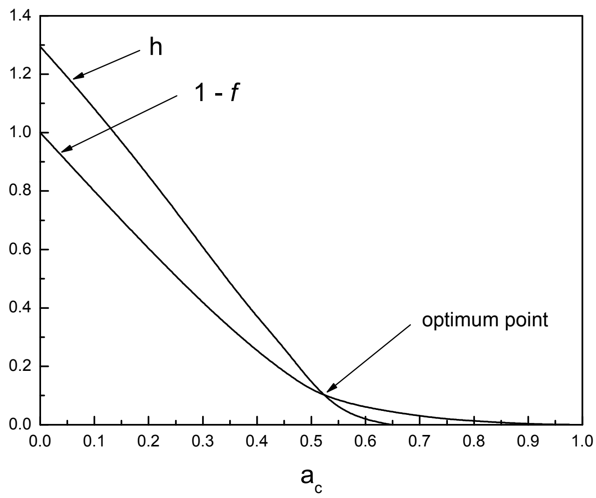


Figure 2. Optimum solution for $a_c$, for $T_e = 15$ °C, and $\ell = 0$, where $h = (a_{\max} - a_c)\frac{\partial f}{\partial a_c}$, for the same case corresponding to Fig. 3.

Equation (4) can be rewritten as

$$\ell = \alpha_E / COP_{el} - \psi - a_c + d/C_A \tag{19}$$

where $\psi$ is given by

$$\psi = \alpha_E / COP_m + \alpha_F (1 - f)/COP \tag{20}$$

Taking the partial derivative of $\ell$ with respect to $T_e$ in equation (19) it leads to

$$\frac{\partial \ell}{\partial T_e} = -\frac{\partial \psi}{\partial T_e} = \left[\alpha_E + \alpha_F (1 - f)/COP_{th}\right] \times \frac{\partial COP_m}{\partial T_e} / COP_m^2 + \alpha_F \left[(1 - f) \times \frac{\partial COP_{th}}{\partial T_e} / COP_{th} + \frac{\partial f}{\partial T_e}\right] / COP \tag{21}$$

By making the derivative given above to vanish, the optimum value for $T_e$ can thus be found. On the other hand, $\alpha_E$ can be expressed as a function of $\alpha_F$ as follows

$$\alpha_E = \alpha_F \left\{ \left[ -\frac{\partial f}{\partial T_e} - (1 - f)\frac{\partial COP_{th}}{\partial T_e} / COP_{th} \right] / COP - (1 - f)\frac{\partial COP_m}{\partial T_e} / COP_m^2 COP_{th} \right\} / \left( \frac{\partial COP_m}{\partial T_e} / COP_m^2 \right) \tag{22}$$

If $\alpha_F$ given by equation (16) is replaced in the above equation in favor of $a_c$, $\alpha_E$ can than be expressed as a function of $a_c$ and $T_e$ . Therefore the curve along with the $\partial \psi / \partial T_e$ vanishes, can be plotted in the coordinates $\alpha_F$ and $\alpha_E$, as a function of $a_c$, for constant values of $T_e$.

Fig. 3 illustrates the particular case corresponding to $T_e = 15$ °C.

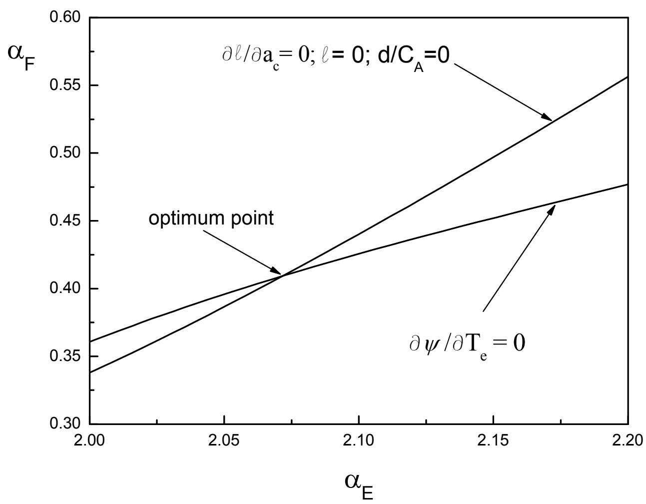


Figure 3. Curves of $\ell = 0$, for $d/C_A = 0$, $T_r = 5$ °C, $T_e = 15$ °C, $T_c = 40$ °C, and $T_g = 80$ °C.

*2.1 Coeficient of performance*

The coefficient of performance for the ideal MDC, for the working fluid R134a used in (Sun, 1997) is fitted by the following correlation,

$$COP_m = y_o + A_1\, e^{(-T_e/t_1)} + A_2\, e^{(-T_e/t_2)} \tag{23}$$

where $y_o$ = 7.6712, $A_1$ = 222.748, $A_2$ = 149260.8, $t_1$ = 6.16277, $t_2$ = 0.89368, for $T_r$ = 5 °C and $T_e$ ranging in the interval of 5 °C to 40 °C. The $COP_m$ for R134a is nearly the same for R114, for the same temperature range considered.

The *COP* of the TDC for the working fluid R114, is fitted by the following correlation,

$$COP_{th} = A_0 + A_1\,\theta + A_2\theta^2 + A_3\theta^3 +$$

$$\left(B_0 + B_1\,\theta + B_2\theta^2 + B_3\theta^3\right)\times$$

$$\exp\left(C_0 + C_1\,\theta + C_2\theta^2 + C_3\theta^3\right)T_e \tag{24}$$

where $\theta$ = $T_g$ /100, $A_o$ = 0.02563, $A_1$ = -1.119, $A_2$ = 1.699, $A_3$ = -0.72, $B_o$ = -0.4026, $B_1$ = 2.107, $B_2$ = -2.195, $B_3$ = 0.8017, $C_o$ = 0.0479, $C_1$ = 0.2346, $C_2$ = -0.182, $C_3$ = 0.04833, for $T_e$ ranging in the interval of -10 °C to 40 °C and $T_g$ ranging in the interval of 60 °C to 110 °C. The above correlation is fitted against the data of a full calculation of the jet nozzle cycle, by using the calculations routines proposed by (Sokolov, 1993) and (Huang, 1999).

The value of $\psi$ at $T_e = T_r$ is shown to be given by

$$\psi(T_r) = \alpha_F\left(1 - f_r\right)/\,COP_{th}\left(T_r, T_c, T_g\right) \tag{25}$$

where $f_r$ is the value of $f$ evaluated at $T_e = T_r$. The value of $\psi$ at $T_e = T_c$ is given by

$$\psi(T_c) = \alpha_E\left(h_2(T_c) - h_1\right)/\left(h_1 - h_3(T_c)\right) \tag{26}$$

where $h$ is the enthalpy of the working fluid according to Fig. 4.

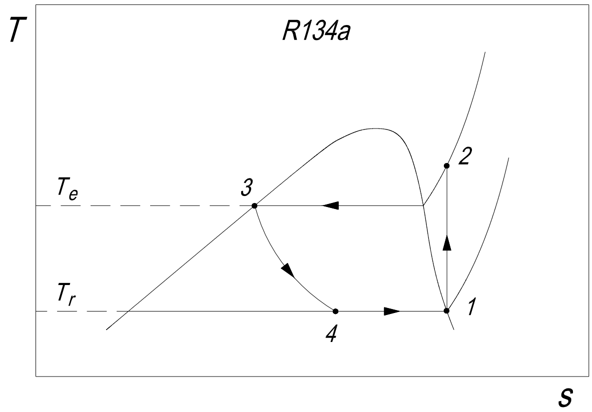


Figure 4. Temperature-entropy diagram for the ideal MDC

Let us define $\lambda_c = \alpha_E / \alpha_F$ and the parameter $\lambda_r$ as

$$\lambda_r = \frac{\left(1 - f_r\right)}{COP_{th}\left(T_r, T_c, T_g\right)} \frac{\left(h_1 - h_3(T_c)\right)}{\left(h_2(T_c) - h_1\right)} \tag{27}$$

Three cases arise from equations (25),(26), and (27) as follows,

Case (i): In this case
$\lambda_c = \lambda_r$
which leads to $\psi(T_r) = \psi(T_c)$,
Case (ii): $\lambda_c < \lambda_r$, which leads to $\psi(T_r) > \psi(T_c)$ and
Case (iii): $\lambda_c > \lambda_r$ , which leads to $\psi(T_r) < \psi(T_c)$.

Fig. 5 illustrates the shape of the function $\psi$ for different values of the intercooler temperature. The curve corresponding to the Carnot cycle limit is also shown. This figure shows that the optimum temperature $T_e$ is strongly dependent on the coefficient of performance of the TDC, for fixed value of the ratio of the electric to the fuel gas cost, $\lambda_c$.

## 3. SECOND LAW ANALYSIS

The optimization carried out in the previous sections is performed regardless to the second law of thermodynamics. However in many circumstances, the relationship between economical figures of merit and the thermodynamic efficiency of the components of the system may be of practical interest.

The entropy generation of the whole cycle shown in Fig.1 is expressed as follows

$$S_{gen} = \frac{Q_g}{T_g} + \frac{Q_r}{T_r} - \frac{Q_c}{T_c} \tag{28}$$

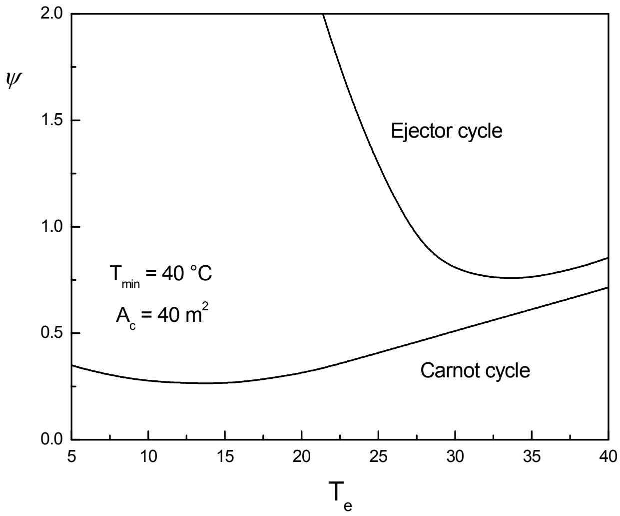


Figure 5. Typical plots of the cost function $\psi$ $(T_e)$ for $\alpha_E = 5.675$, $\alpha_F = 3.03$ $(\lambda_c = 1.873)$.

where

$$Q_c = Q_r + Q_g + W_m \tag{29}$$

From equations (2) and (3), and equation (29), equation (29) can be expressed as follows

$$S_{gen} = Q_r \left\{ \left( \frac{1+COP_m}{COP_m} \right) \left( \frac{1}{T_c} - \frac{1}{T_g} \right) \frac{1}{COP_{th}} + \frac{1}{T_c} - \frac{1}{T_r} + \frac{1}{T_c\, COP_m} \right\} \tag{30}$$

From the definition of irreversibility, referred to some thermodynamic environment at temperature $T_o$ it follows,

$$I = T_o Q_r \left\{ \left( \frac{1+COP_m}{COP_m} \right) \left( \frac{1}{T_c} - \frac{1}{T_g} \right) \frac{1}{COP_{th}} + \frac{1}{T_c} - \frac{1}{T_r} + \frac{1}{T_c\, COP_m} \right\} \tag{31}$$

The exergetic efficiency of the whole cycle is defined as

$$\eta_{ex} = 1 - I / \left( \sum E_{Qin} + W_m \right) \tag{32}$$

where $E_{Qin}$ is the exergy of the heat input (Kotas, 1995). The denominator of the above equation can be expressed as follows

$$\zeta = \left( \sum E_{Qin} + W_m \right) / Q_r = \left( \frac{T_g - T_o}{T_g} \right) / COP + \left( \frac{T_r - T_o}{T_r} \right) + 1/COP_m \tag{33}$$

Fig. 6 illustrates the shape of $\phi$ and $\eta_{ex}$ for the particular cases corresponding to figures 3 and 5. This figure shows a possibility of existing optimum values for the exergetic efficiency as a function of $T_e$.

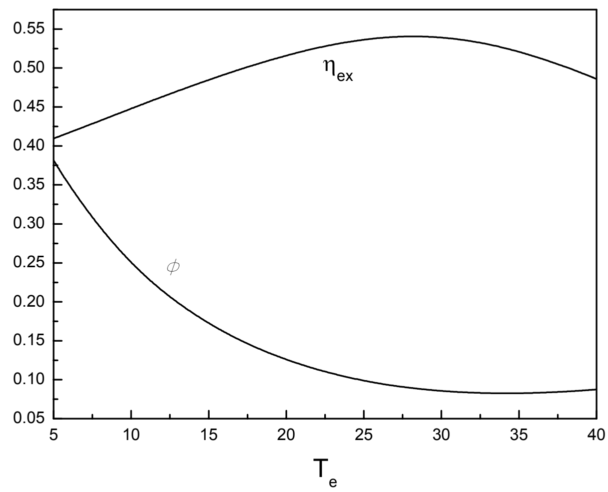


Figure 6. Irreversibility ( $\phi$ ) and exergetic efficiency ( $\eta_{ex}$ ) for $A_c$ = 40 m$^2$

### 3.1 Constrained optimization

The irreversibility given by equation (31) is a function of the intercooler temperature $T_e$. For each specified value of $I$, a value for $T_e$ can be found by solving equation (31) in terms of $T_e$.

The constrained optimum for $\ell$ in terms of $a_c$ and $T_e$ can be determined by the method of Lagrange multipliers as follows.

Let the auxiliary function $L$ defined by

$$L = \ell + \lambda_\ell (\phi - \phi_o) \tag{34}$$

where $\phi = I / Q_r$ and $\phi_o = I_o / Q_r$, for a given value of $I_o$. By taking the partial derivative of equation (34) with respect to $a_c$ to vanish, equation (16) is found, while equation (18) holds for the special case of $\ell = 0$.

By taking the partial derivative of $L$ with respect to $T_e$ to vanish, the following equation holds,

$$\frac{\partial \ell}{\partial T_e} + \lambda_\ell \frac{\partial \phi}{\partial T_e} = 0 \tag{35}$$

or in terms of $\psi$ it follows

$$-\frac{\partial \psi}{\partial T_e} + \lambda_\ell \frac{\partial \phi}{\partial T_e} = 0 \qquad (36)$$

Since $\eta_{ex} = 1 - \phi / \zeta$, for each value of $\eta_{ex}$, $T_e$ can be obtained from equations (31), (32), and (33). With $T_e$ thus found, $a_c$ can be found from equations (16) or (18), as the case might be. With the values of $a_c$ and $T_e$ thus found, the Lagrange multiplier $\lambda_\ell$ is found from equation (37) as follows,

$$\lambda_\ell = \frac{\partial \psi}{\partial T_e} \Big/ \frac{\partial \phi}{\partial T_e} \qquad (37)$$

The derivative of $\phi$ with respect to $T_e$ is given by

$$\frac{\partial \phi}{\partial T_e} = -T_o \left( \frac{1}{T_g} - \frac{1}{T_c} \right) \left[ \left( 1 + COP_m \right) \frac{\partial COP_{th}}{\partial T_e} / COP_{th} + \frac{\partial COP_m}{\partial T_e} / COP_m \right] / COP_m \; COP_{th} - \frac{T_o}{T_c} \frac{1}{COP_m^{\;2}} \frac{\partial COP_m}{\partial T_e} \qquad (38)$$

Curves for constant value of $\ell$ can be plotted as a function of $a_c$ and $\eta_{ex}$ (or $T_e$).

The loci of a curve of $\ell = 0$ can also be plotted as a function of $\alpha_F$ and $\alpha_E$, for fixed values of $\eta_{ex}$.

Once the Lagrange multiplier $\lambda_\ell$ is found from equation (37), for each value of $\eta_{ex}$, the sensibility coefficient of optimum $\ell$ with respect to either $\phi_o$ or $\eta_{ex}$ can be found by the following

$$\frac{\partial \ell_{opt}}{\partial \phi_o} = \lambda_\ell \qquad (39)$$

## 4. CONCLUSIONS

The present paper reports a basic economical analysis of a double stage enhanced cooling cycle assisted by solar energy. The approach presented here may be useful to determine the condition under which, a double stage solar assisted cooling cycle can be economically competitive with a single stage mechanical compression cooling cycles, for fixed condenser and evaporator temperatures.

The life cost savings technique, i.e., the $P_1$ - $P_2$ method of (Klein and Beckman, 1979) was used to set down the objective function for the economical analysis. The second law analysis was performed in order to relate the life cost savings to the exergetic efficiency of the whole cycle. The analysis presented here is appropriate to make a straightforward calculation in order to determine also the bounds for the economical feasibility region, in terms of the electric energy cost, the auxiliary energy cost, and the difference among the capital costs considered. The design method $f - \overline{\phi}$ chart is shown here to be a convenient tool to determine, prior to any full simulation, the conditions under which a thermally driven cycle can be economically more attractive than a mechanically driven cycle. It can also be used to determine the optimum intercooler temperature, for the case of conjugation of the MDC and the TDC in a double stage cooling cycle, for given energy costs and capital costs. It is also shown that the ratio of the electricity to the auxiliary heat cost is a meaningful parameter, which can be used to determine the conditions under which optimum solutions exist. The numerical example presented here shows that the optimum solution derived from the life cycle cost savings function does not correspond to the optimum solution derived from the function of the exergetic efficiency. This is expected, since an optimum economical design of a thermal system, is in general not equivalent to an optimum thermodynamic design, as is pointed out in (Kotas, 1995).

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