We are IntechOpen,
the world's leading publisher of
Open Access books
Built by scientists, for scientists

6,900
Open access books available

185,000
International authors and editors

200M
Downloads

154
Countries delivered to

Our authors are among the

TOP 1%
most cited scientists

12.2%
Contributors from top 500 universities

CLARIVATE ANALYTICS
BOOK CITATION INDEX
INDEXED

WEB OF SCIENCE™

Selection of our books indexed in the Book Citation Index
in Web of Science™ Core Collection (BKCI)

Interested in publishing with us?
Contact book.department@intechopen.com

Numbers displayed above are based on latest data collected.
For more information visit www.intechopen.com

ECOP Criterion for Irreversible Three-Heat-Source Absorption Refrigerators

Paiguy Armand Ngouateu Wouagfack and Réné Tchinda

Additional information is available at the end of the chapter

1. Introduction

In this chapter, the optimization analysis based on the new thermo-ecological criterion (ECOP) first performed by Ust et al. [1] for the heat engines is extended to an irreversible three-heat-source absorption refrigerator. The thermo-ecological objective function ECOP is optimized with respect to the temperatures of the working fluid. The maximum ECOP and the corresponding optimal temperatures of the working fluid, coefficient of performance, specific cooling load, specific entropy generation rate and heat-transfer surface areas in the exchangers are then derived analytically. Comparative analysis with the COP criterion is carried out to prove the utility of the ecological coefficient of performance criterion.

2. Thermodynamics analysis

The main components of an absorption refrigeration system are a generator, an absorber, a condenser and an evaporator as shown schematically in Fig. 1 [2]. In the shown model, $\dot{Q}_H$ is the rate of absorbed heat from the heat source at temperature T_H to generator, $\dot{Q}_C$ and $\dot{Q}_A$ are, respectively, the heat rejection rates from the condenser and absorber to the heat sinks at temperatures T_C and T_A and $\dot{Q}_L$ is the heat input rate from the cooling space at temperature T_L to the evaporator. In absorption refrigeration systems, usually NH₃/H₂O and LiBr/H₂O are used as the working substances, and these substances abide by ozone depletion regulations, since they do not consist of chlorofluorocarbons. In Fig. 1, the liquid rich solution at state 1 is pressurized to state 1' with a pump. In the generator, the working fluid is concentrated to state 3 by evaporating the working medium by means of $\dot{Q}_H$ heat rate input. The weak solution at state 2 passes through the expansion valve into the absorber with a pressure reduction (2–2'). In the condenser, the working fluid at

state 3 is condensed to state 4 by removing $\dot{Q}_C$ heat rate. The condensed working fluid at state 4 is then throttled by a valve and enters the evaporator at state 4′. The liquid working fluid is evaporated due to heat transfer rate $\dot{Q}_L$ from the cooling space to the working fluid (4′–5). Finally, the vaporized working fluid is absorbed by the weak solution in the absorber, and by means of $\dot{Q}_A$ heat rate release in the absorber, state 1 is reached.

Work input required by the solution pump in the system is negligible relative to the energy input to the generator and is often neglected for the purpose of analysis. Under such assumption, the equation for the first law of thermodynamics is written as:

$$\dot{Q}_H + \dot{Q}_L - \dot{Q}_C - \dot{Q}_A = 0 \tag{1}$$

Absorption refrigeration systems operate between three temperature levels, if $T_A = T_C$, or four temperature levels when $T_A \neq T_C$. In this chapter, by taking $T_A = T_C$, the cycle of the working fluid consists of three irreversible isothermal and three irreversible adiabatic processes. The temperatures of the working fluid in the three isothermal processes are different from those of the external heat reservoirs so that heat is transferred under a finite temperature difference, as shown in Fig. 2 where

$$\dot{Q}_O = \dot{Q}_C + \dot{Q}_A \tag{2}$$

T_1 and T_2 are, respectively, the temperatures of the working fluid in the generator and evaporator. It is assumed that the working fluid in the condenser and absorber has the same temperature T_3 [2]. $\dot{Q}_{LC}$ is the heat leak from the heat sink to the cooled space.

The heat exchanges between the working fluid and heat reservoirs obey a linear heat transfer law, so that the heat-transfer equations in the generator, evaporator, condenser and absorber are, respectively, expressed as follows:

$$\dot{Q}_H = U_H A_H \left(T_H - T_1 \right) \tag{3}$$

$$\dot{Q}_L = U_L A_L \left(T_L - T_2 \right) \tag{4}$$

$$\dot{Q}_O = U_O \left(A_A + A_C \right) \left(T_3 - T_O \right) \tag{5}$$

where A_H, A_L, A_C and A_A are, respectively, the heat-transfer areas of the generator, evaporator, condenser and absorber, U_H and U_L are, respectively, the overall heat-transfer coefficients of the generator and evaporator, and it is assumed that the condenser and absorber have the same overall heat-transfer coefficient U_O [2].
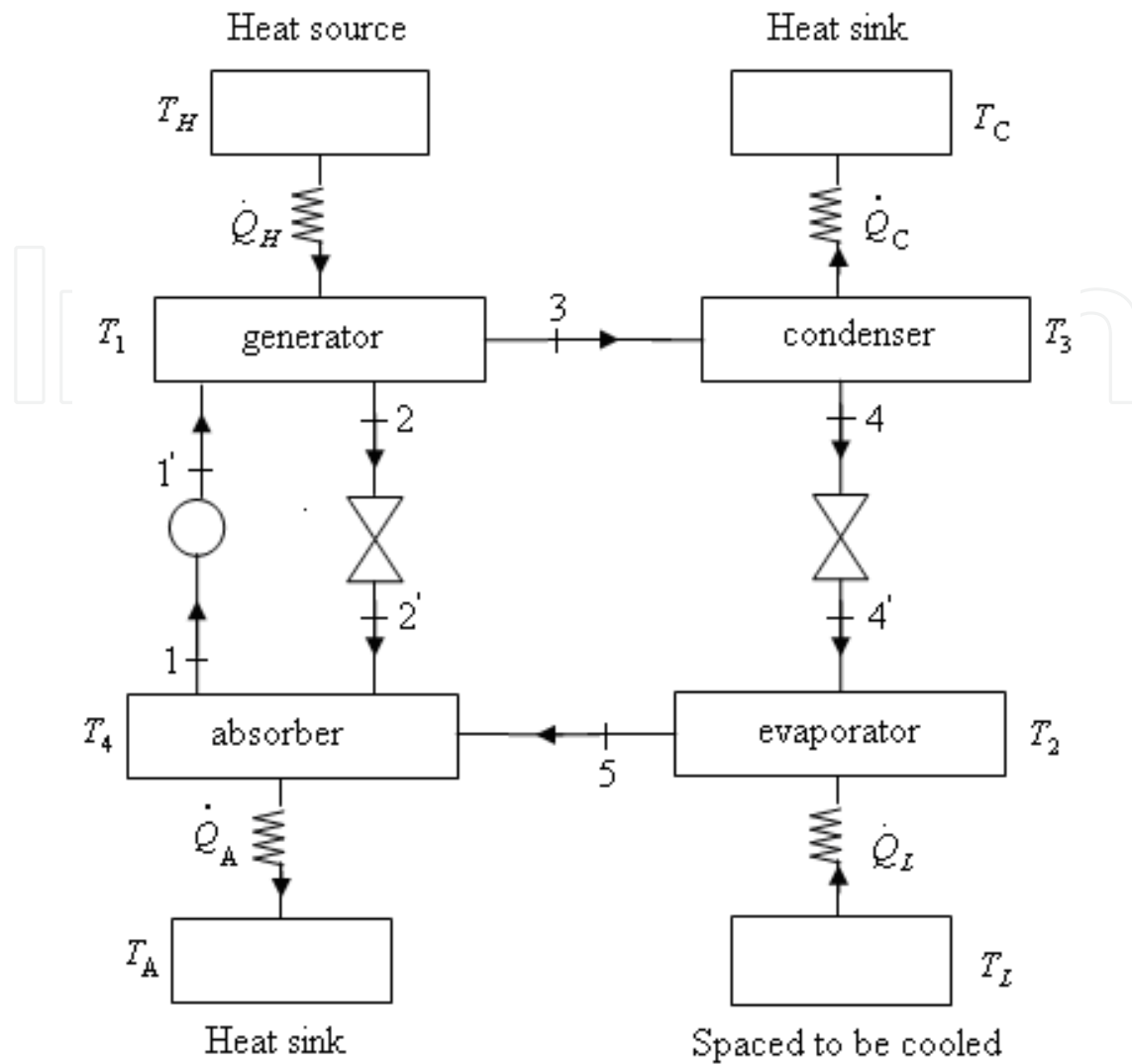
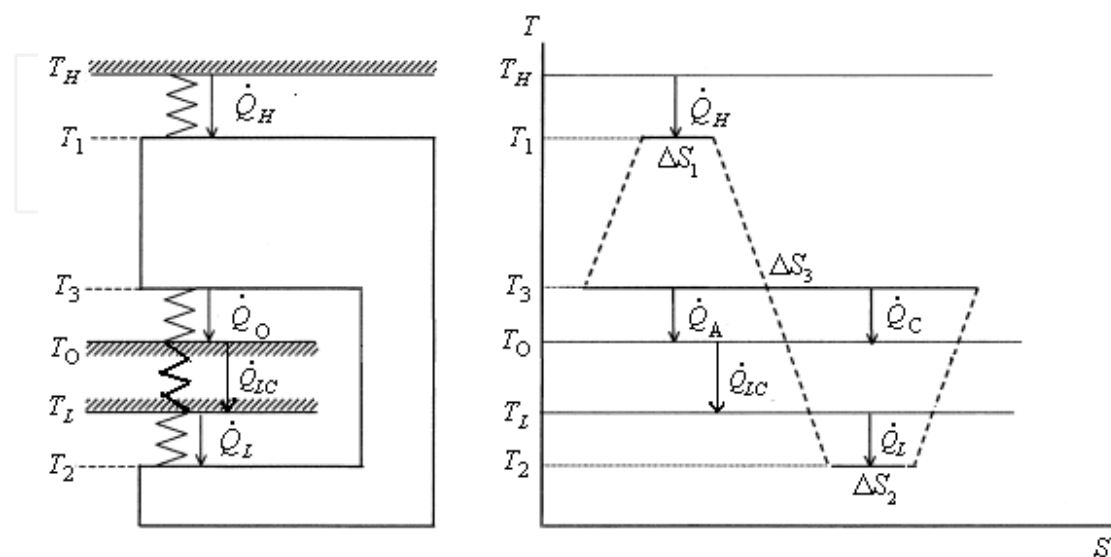**Figure 1.** Schematic diagram of absorption refrigeration system [2]

Figure 2. Considered irreversible absorption refrigeration model and its T–S diagram.

The absorption refrigeration system does not exchange heat with other external reservoirs except for the three heat reservoirs at temperatures T_H, T_L and T_O, so the total heat-transfer area between the cycle system and the external heat reservoirs is given by the relationships:

$$A = A_H + A_L + A_O \qquad (6)$$

where

$$A_O = A_C + A_A. \qquad (7)$$

The rate of heat leakage $\dot{Q}_{LC}$ from the heat sink at temperature T_O to the cold reservoir at temperature T_L was first provided by Bejan [3] and it is given as:

$$\dot{Q}_{LC} = K_{LC}(T_O - T_L) \qquad (8)$$

where K_{LC} is the heat leak coefficient.

Real absorption refrigerators are complex devices and suffer from a series of irreversibilities. Besides the irreversibility of finite rate heat transfer which is considered in the endoreversible cycle models and the heat leak from the heat sink to the cooled space, there also exist other sources of irreversibility. The internal irreversibilities that result from friction, mass transfer and other working fluid dissipations are an another main source of irreversibility, which can decrease the coefficient of performance and the cooling load of absorption refrigerators. The total effect of the internal irreversibilities on the working fluid can be characterized in terms of entropy production. An irreversibility factor is introduced to describe these internal irreversibilities:

$$I = \frac{\Delta S_3}{\Delta S_1 + \Delta S_2} \qquad (9)$$

On the basis of the second law of thermodynamics, $\Delta S_3 > \Delta S_1 + \Delta S_2$ for an internally irreversible cycle, so that $I > 1$. If the internal irreversibility is neglected, the cycle is endoreversible and so $I = 1$. The second law of thermodynamics for an irreversible three-heat-source cycle requires that:

$$\oint \frac{\delta \dot{Q}}{T} = \frac{\dot{Q}_H}{T_1} + \frac{\dot{Q}_L}{T_2} - \frac{\dot{Q}_O}{T_3} \leq 0 \qquad (10)$$

From Eq. (9), the inequality in Eq. (10) is written as:

$$\frac{\dot{Q}_H}{T_1} + \frac{\dot{Q}_L}{T_2} - \frac{\dot{Q}_O}{IT_3} = 0 \qquad (11)$$

The coefficient of performance of the irreversible three-heat-source absorption refrigerator is:

$$COP = \frac{\dot{Q}_L - \dot{Q}_{LC}}{\dot{Q}_H} = \frac{\dot{Q}_L}{\dot{Q}_H}\left(1 - \frac{\dot{Q}_{LC}}{\dot{Q}_L}\right) \tag{12}$$

From Eq. (6), it is expressed as:

$$A_L = \frac{A}{1 + \frac{A_H}{A_L} + \frac{A_O}{A_L}} \tag{13}$$

Using Eqs. (3)-(5), Eq. (13) is rewritten as:

$$A_L = \frac{A}{1 + \frac{\dot{Q}_H}{\dot{Q}_L}\frac{U_L(T_L - T_2)}{U_H(T_H - T_1)} + \frac{\dot{Q}_O}{\dot{Q}_L}\frac{U_L(T_L - T_2)}{U_O(T_3 - T_O)}} \tag{14}$$

Combining Eqs. (1) and (11), the following ratios are derived:

$$\frac{\dot{Q}_L}{\dot{Q}_H} = \frac{T_2(T_1 - IT_3)}{T_1(IT_3 - T_2)} \tag{15}$$

$$\frac{\dot{Q}_O}{\dot{Q}_L} = \frac{IT_3(T_1 - T_2)}{(T_1 - IT_3)T_2} \tag{16}$$

The first is the coefficient of performance of the irreversible three-heat-source absorption refrigeration cycle without heat leak losses.

Substituting Eqs. (15) and (16) into Eq. (14), the heat-transfer area of the evaporator is expressed as a function of T_1, T_2 and T_3 for a given total heat-transfer areas :

$$A_L = \frac{A}{1 + \frac{U_L T_1(IT_3 - T_2)(T_L - T_2)}{U_H(T_H - T_1)(T_1 - IT_3)T_2} + \frac{U_L IT_3(T_1 - T_2)(T_L - T_2)}{U_O(T_3 - T_O)(T_1 - IT_3)T_2}} \tag{17}$$

By investigating similar reasoning, the heat-transfer areas of the generator and of condenser and absorber are given respectively by:

$$A_H = \frac{A}{1 + \frac{U_H T_2(T_1 - IT_3)(T_H - T_1)}{U_L T_1(IT_3 - T_2)(T_L - T_2)} + \frac{U_H IT_3(T_1 - T_2)(T_H - T_1)}{U_O T_1(IT_3 - T_2)(T_3 - T_O)}} \tag{18}$$

and

$$A_O = \frac{A}{1 + \dfrac{U_O T_1 (IT_3 - T_2)(T_3 - T_O)}{U_H IT_3 (T_1 - T_2)(T_H - T_1)} + \dfrac{U_O T_2 (T_1 - IT_3)(T_3 - T_O)}{U_L IT_3 (T_1 - T_2)(T_L - T_2)}} \qquad (19)$$

Substituting Eq. (17) into Eq. (4):

$$\dot{Q}_L = \frac{A}{\dfrac{1}{U_L(T_L - T_2)} + \dfrac{T_1(IT_3 - T_2)}{U_H(T_H - T_1)(T_1 - IT_3)T_2} + \dfrac{IT_3(T_1 - T_2)}{U_O(T_3 - T_O)(T_1 - IT_3)T_2}} \qquad (20)$$

Combining Eqs. (8), (12), (15) and (20), the coefficient of performance of the irreversible three-heat-source refrigerator as a function of the temperatures T_1, T_2 and T_3 of the working fluid in the generator, evaporator, condenser and absorber is obtained:

$$COP = \frac{T_2(T_1 - IT_3)}{T_1(IT_3 - T_2)} \left\{ 1 - \xi(T_O - T_L) \left[\frac{1}{U_L(T_L - T_2)} + \frac{T_1(IT_3 - T_2)}{U_H(T_H - T_1)(T_1 - IT_3)T_2} + \frac{IT_3(T_1 - T_2)}{U_O(T_3 - T_O)(T_1 - IT_3)T_2} \right] \right\} \qquad (21)$$

where the parameter

$$\xi = \frac{K_{LC}}{A} \qquad (22)$$

represents the heat leakage coefficient and its dimension is w/(Km²)

The specific cooling load of the irreversible three-heat-source refrigerator is deduced as:

$$r = \frac{\dot{Q}_L - \dot{Q}_{LC}}{A} = \left[\frac{1}{U_L(T_L - T_2)} + \frac{T_1(IT_3 - T_2)}{U_H(T_H - T_1)(T_1 - IT_3)T_2} + \frac{IT_3(T_1 - T_2)}{U_O(T_3 - T_O)(T_1 - IT_3)T_2} \right]^{-1} - \xi(T_O - T_L) \qquad (23)$$

The specific entropy production rate of the irreversible three-heat-source absorption refrigerator is:

$$s = \frac{\dot{\sigma}}{A} = \frac{\dfrac{\dot{Q}_O - \dot{Q}_{LC}}{T_O} - \dfrac{\dot{Q}_H}{T_H} - \dfrac{\dot{Q}_L - \dot{Q}_{LC}}{T_L}}{A} \qquad (24)$$

Using Eq. (1) s is rewritten as:

$$s = \left(\frac{1}{T_L} - \frac{1}{T_O}\right)\frac{\dot{Q}_{LC}}{A} + \left(\frac{1}{T_O} - \frac{1}{T_H}\right)\frac{\dot{Q}_H}{\dot{Q}_L}\frac{\dot{Q}_L}{A} + \left(\frac{1}{T_O} - \frac{1}{T_L}\right)\frac{\dot{Q}_L}{A} \tag{25}$$

or

$$s = \left(\frac{1}{T_O} - \frac{1}{T_H}\right)\frac{\dot{Q}_H}{A} + \left(\frac{1}{T_O} - \frac{1}{T_L}\right)\frac{\dot{Q}_L - \dot{Q}_{LC}}{A} \tag{26}$$

Substituting Eqs.(8), (15) and (20) into Eq. (25), the specific entropy production rate as a function of T_1, T_2 and T_3 is given by :

$$s = \left(\frac{1}{T_L} - \frac{1}{T_O}\right)\left\{\xi(T_O - T_L) - \left[1 - \frac{\varepsilon_r T_1(IT_3 - T_2)}{T_2(T_1 - IT_3)}\right]\left[\frac{1}{U_L(T_L - T_2)} + \frac{T_1(IT_3 - T_2)}{U_H(T_H - T_1)(T_1 - IT_3)T_2} + \frac{IT_3(T_1 - T_2)}{U_O(T_3 - T_O)(T_1 - IT_3)T_2}\right]^{-1}\right\} \tag{27}$$

where

$$\varepsilon_r = \frac{\left(1 - \dfrac{T_O}{T_H}\right)}{\left(\dfrac{T_O}{T_L} - 1\right)} \tag{28}$$

is the coefficient of performance for reversible three-heat-source refrigerator.

According to the definition of the general thermo-ecological criterion function for different heat engine models [4-9], a two-heat-source refrigerator [10, 11] and three-heat-source absorption refrigerator [2], the new thermo-ecological objective function called ecological coefficient of performance (*ECOP)* of an absorption refrigerator is defined as:

$$ECOP = \frac{\dot{Q}_L - \dot{Q}_{LC}}{T_{env}\dot{\sigma}} = \frac{\dot{Q}_L - \dot{Q}_{LC}}{AT_{env}s} \tag{29}$$

Putting Eq.(26) into Eq. (29):

$$ECOP = \frac{1}{T_{env}\left[\dfrac{1}{T_O} - \dfrac{1}{T_H} + \left(\dfrac{1}{T_O} - \dfrac{1}{T_L}\right)\dfrac{1}{COP}\right]} \tag{30}$$

When Eq. (21) is put in Eq. (30), the ecological coefficient of performance of the irreversible three-heat-source absorption refrigerator as a function of T_1, T_2 and T_3 is derived as :

$$ECOP = \cfrac{\cfrac{1}{T_{env}\left(T_O^{-1} - T_L^{-1}\right)}}{1 - \cfrac{\varepsilon_r T_1\left(IT_3 - T_2\right)}{T_2\left(T_1 - IT_3\right)}\left\{1 - \xi\left(T_O - T_L\right)\left[\cfrac{1}{U_L\left(T_L - T_2\right)} + \cfrac{T_1\left(IT_3 - T_2\right)}{U_H\left(T_H - T_1\right)\left(T_1 - IT_3\right)T_2} + \cfrac{IT_3\left(T_1 - T_2\right)}{U_O\left(T_3 - T_O\right)\left(T_1 - IT_3\right)T_2}\right]\right\}^{-1}}$$

$$(31)$$

where T_{env} is the temperature in the environment conditions.

3. Performance optimization for a three-heat-source irreversible absorption refrigerator based on ECOP criterion

The ECOP function given in Eq. (31) is plotted with respect to the working fluid temperatures (T_1, T_2 and T_3) for different internal irreversibility parameters as shown in Fig. 3(a), (b) and (c). As it can be seen from the figure, there exists a specific T_1, T_2 and T_3 that maximize the ECOP function for given I and ξ values. Therefore, Eq. (31) can be maximized (or optimized) with respect to T_1, T_2 and T_3. The optimization is carried out analytically.
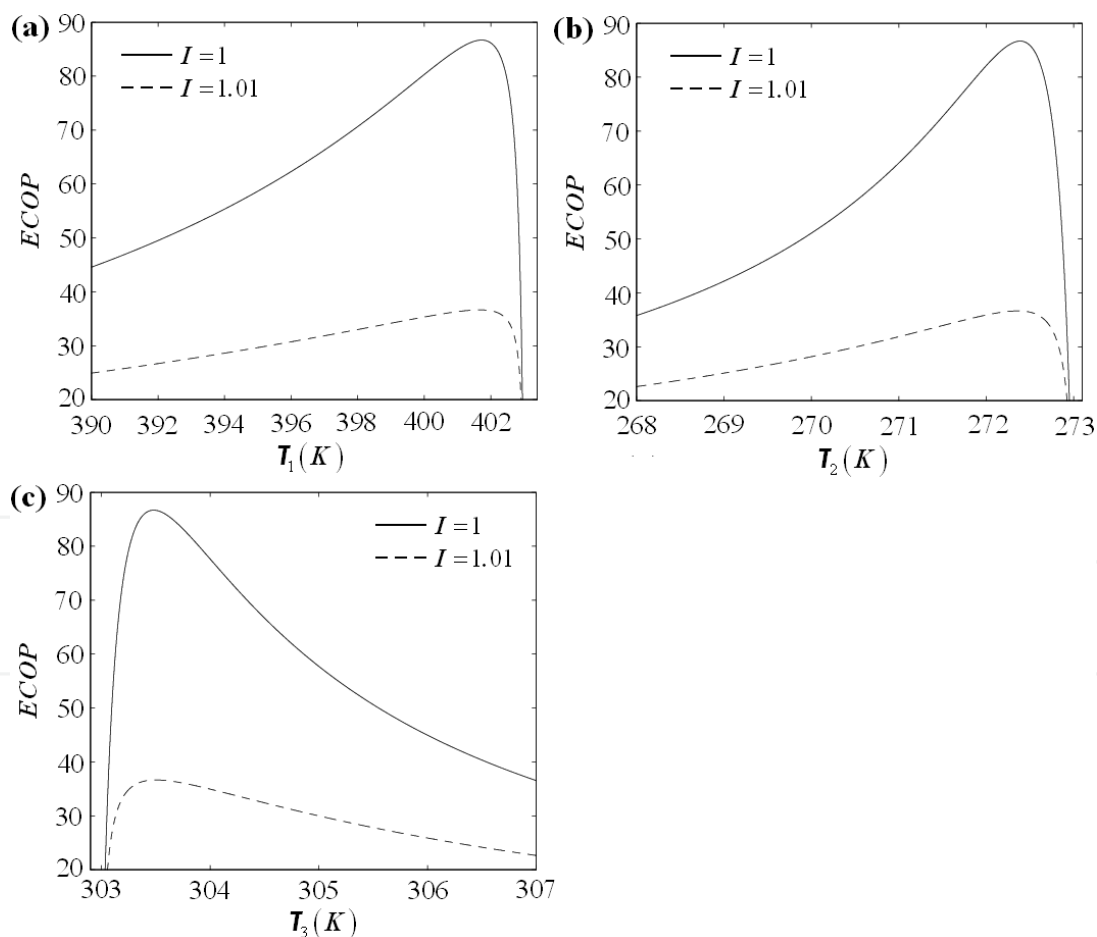

Figure 3. Variation of the ECOP objective function with respect to T_1 (a), T_2 (b) and T_3 (c) for different I values ($T_G = 403K$, $T_L = 273K$, $T_O = 303K$, $T_{env} = 290K$, $U_G = 1163\ W/m^2K$, $U_E = 2326\ W/m^2K$, $U_O = 4650\ W/m^2K$, $K_L = 1082\ W/K$, $A = 1100\ m^2$)

For the sake of convenience, let

$$x = \frac{IT_3}{T_1} \tag{32}$$

$$y = \frac{IT_3}{T_2} \tag{33}$$

$$z = IT_3 \tag{34}$$

Then Eq. (31) is rewritten as:

$$ECOP = \frac{\dfrac{1}{T_{env}\left(T_O^{-1} - T_L^{-1}\right)}}{1 - \dfrac{\varepsilon_r(y-1)}{1-x}\left\{1 - \xi(T_O - T_E)\left[\dfrac{y}{U_L(T_L y - z)} + \dfrac{x(y-1)}{U_H(T_H x - z)(1-x)} + \dfrac{y-x}{U(z-T)(1-x)}\right]\right\}^{-1}} \tag{35}$$

where

$$T = IT_O \tag{36}$$

and

$$U = \frac{U_O}{I}. \tag{37}$$

Starting from Eq. (35), the extremal conditions:

$$\frac{\partial ECOP}{\partial x} = 0 \tag{38}$$

$$\frac{\partial ECOP}{\partial y} = 0 \tag{39}$$

$$\frac{\partial ECOP}{\partial z} = 0 \tag{40}$$

give respectively:

$$\frac{1}{\xi(T_O - T_E)} - \frac{y}{U_L(T_L y - z)} - \frac{1}{U(z-T)} - \frac{z(y-1)}{U_H(T_H x - z)^2} = 0 \tag{41}$$

$$\frac{1}{\xi(T_O - T_E)} - \frac{y}{U_L(T_L y - z)} - \frac{1}{U(z - T)} - \frac{z(y - 1)}{U_L(T_L y - z)^2} = 0 \tag{42}$$

$$\frac{y(1 - x)}{U_L(T_L y - z)^2} + \frac{x(y - 1)}{U_H(T_H x - z)^2} - \frac{y - x}{U(z - T)^2} = 0 \tag{43}$$

Combining Eqs (41)-(43), the following general relation is found:

$$\sqrt{U_H}(T_H x - z) = \sqrt{U_L}(T_L y - z) = \sqrt{U}(z - T) \tag{44}$$

From Eqs (44), it is derived as:

$$x = \frac{(1 + b_1)z}{T_H} - \frac{b_1 T}{T_H} \tag{45}$$

$$y = \frac{(1 + b_2)z}{T_L} - \frac{b_2 T}{T_L} \tag{46}$$

where

$$b_1 = \sqrt{\frac{U}{U_H}} \tag{47}$$

$$b_2 = \sqrt{\frac{U}{U_L}} \tag{48}$$

When Eqs. (45) and (46) are substituted into Eq. (43):

$$z = T\frac{D + b_2}{1 + b_2} \tag{49}$$

where

$$D = \frac{1 + \sqrt{d_1\left[1 - \frac{T_L(1 - d_1)}{T}\right]}}{1 - d_1} \tag{50}$$

$$d_1 = \xi \frac{(1+b_2)^2}{U}\left(\frac{T_O}{T_L} - 1\right) \tag{51}$$

Therefore Eqs. (45) and (46) are rewritten as:

$$x = \frac{T}{T_H} B_1 (D + B) \tag{52}$$

$$y = \frac{T}{T_L} D \tag{53}$$

where

$$B = \frac{b_2 - b_1}{1 + b_1} \tag{54}$$

$$B_1 = \frac{1 + b_1}{1 + b_2} \tag{55}$$

Using Eqs. (49), (52) and (53) with Eqs.(32)-(34), the corresponding optimal temperatures of the working fluid in the three isothermal processes when the ecological coefficient of performance is a maximum, are, respectively, determined by:

$$T_1^* = T_H \frac{D + b_2}{(1 + b_1)(D + B)} \tag{56}$$

$$T_2^* = T_L \frac{D + b_2}{D(1 + b_2)} \tag{57}$$

$$T_3^* = T_O \frac{D + b_2}{1 + b_2} \tag{58}$$

Substituting Eqs. (56)-(58) into Eqs. (21), (23), (27) and (31) the maximum *ECOP* function and the corresponding optimal coefficient of performance, optimal specific cooling load and optimal specific entropy generation rate are derived, respectively, as:

$$ECOP_{max} = \frac{1}{T_{env}\left(T_O^{-1} - T_L^{-1}\right)}$$

$$\times \frac{1}{1 - \dfrac{\varepsilon_r T_H (TD - T_L)}{\left[T_H - B_1 (D+B)T \right] T_L} \left\{ 1 - \xi(T_O - T_L) \dfrac{T_H D - B_1^{\ 2}(D+B)(T_L + BDT)}{U^* B_1^{\ 2}(D-1)\left[T_H - B_1(D+B)T \right] T_L} \right\}^{-1}} \tag{59}$$

$$COP^* = \frac{\left[T_H - B_1(D+B)T \right] T_L}{T_H (TD - T_L)} \left\{ 1 - \xi(T_O - T_L) \frac{T_H D - B_1^{\ 2}(D+B)(T_L + BDT)}{U^* B_1^{\ 2}(D-1)\left[T_H - B_1(D+B)T \right] T_L} \right\} \tag{60}$$

$$r^* = \frac{U^* B_1^{\ 2}(D-1)\left[T_H - B_1(D+B)T \right] T_H}{T_H D - B_1^{\ 2}(D+B)(T_L + BDT)} - \xi(T_O - T_L) \tag{61}$$

$$s^* = \left(\frac{1}{T_L} - \frac{1}{T_O} \right) \left\{ \xi(T_O - T_L) - \left[1 - \frac{\varepsilon_r T_H (TD - T_L)}{\left[T_H - B_1(D+B)T \right] T_L} \right] \left[\frac{U^* B_1^{\ 2}(D-1)\left[T_H - B_1(D+B)T \right] T_L}{T_H D - B_1^{\ 2}(D+B)(T_L + BDT)} \right] \right\} \tag{62}$$

where

$$U^* = \frac{U}{\left(1 + b_1\right)^2} \tag{63}$$

From Eqs. (17)-(19) and (56)-(58), it is found that , when the three-heat-source absorption refrigerator is operated in the state of maximum ecological coefficient of performance, the relations between the heat-transfer areas of the heat exchangers and the total heat-transfer area are determined by:

$$A_H^{\ *} = A \frac{b_1}{1 + b_1} \frac{B_1^{\ 2}(D+B)(TD - T_L)}{T_H D - B_1^{\ 2}(D+B)(T_L + BDT)} \tag{64}$$

$$A_L^{\ *} = A \frac{b_2}{1 + b_1} \frac{DB_1 \left[T_H - B_1 T(D+B) \right]}{T_H D - B_1^{\ 2}(D+B)(T_L + BDT)} \tag{65}$$

$$A_O^{\ *} = A \frac{1}{1 + b_1} \frac{B_1 \left[T_H D - B_1 T_L (D+B) \right]}{T_H D - B_1^{\ 2}(D+B)(T_L + BDT)} \tag{66}$$

From Equations (64)-(66), a concise optimum relation for the distribution of the heat-transfer areas is obtained as:

$$\sqrt{U_H} A_H^{\ *} + \sqrt{U_L} A_L^{\ *} = \sqrt{U} A_O^{\ *} \tag{67}$$

Obviously, this relation is independent of the heat leak and the temperatures of the external heat reservoirs.

4. Comparison with COP criterion

In Fig.4, the variation of the normalized ECOP ($\overline{ECOP} = \dfrac{ECOP}{ECOP_{max}}$), normalized COP ($\overline{COP} = \dfrac{COP}{COP_{max}}$) and the specific cooling load (r) with respect to the specific entropy generation rate (s) are demonstrated. One interesting observation from this figure is that maximum of the ECOP and COP coincides although their functional forms are different: the coefficient of performance gives information about the necessary heat rate input in order to produce certain amount of cooling load and the ecological coefficient of performance gives information about the entropy generation rate or loss rate of availability in order to produce certain amount of cooling load. The maximum ECOP and COP conditions give the same amount of cooling load and entropy generation rate. It is also seen analytically that the performance parameters T_1^*, T_2^*, T_3^*, A_1^*, A_2^*, A_3^*, r^*, s^* and $COP^* = COP_{max}$ at the maximum ECOP and maximum COP are same. Getting the same performance at maximum ECOP and COP conditions is an expected and logical result. Since, for a certain cooling load the maximum COP results from minimum heat consumption so that minimum environmental pollution. The minimum environmental pollution is also achieved by maximizing the $ECOP$. Although the optimal performance conditions $ECOP$ and COP criteria are same, their impact on the system design performance is different. The coefficient of performance is used to evaluate the performance and the efficiency of systems. This method only takes into account the first law of thermodynamics which is concerned only with the conversion of energy, and therefore, can not show how or where irreversibilities in a system or process occur. Also, when different sources and forms of energy are involved within a system, the COP criterion of a system doesn't describe its performance from the view point of the energy quality involved. This factor is taken into account by the second law of thermodynamics characterized by the entropy production which appears in the ecological coefficient of performance criterion ($ECOP$). This aspect is of major importance today since that with the requirement of a rigorous management of our energy resources, one should have brought to be interested more and more in the second principle of thermodynamics, because degradations of energy, in other words the entropy productions, are equivalent to consumption of energy resources. For this important reason, the $ECOP$ criterion can enhance the system performance of the absorption refrigerators by reducing the irreversible losses in the system. A better understanding of the second law of thermodynamics reveals that the ecological coefficient of performance optimization is an important technique in achieving better operating conditions.
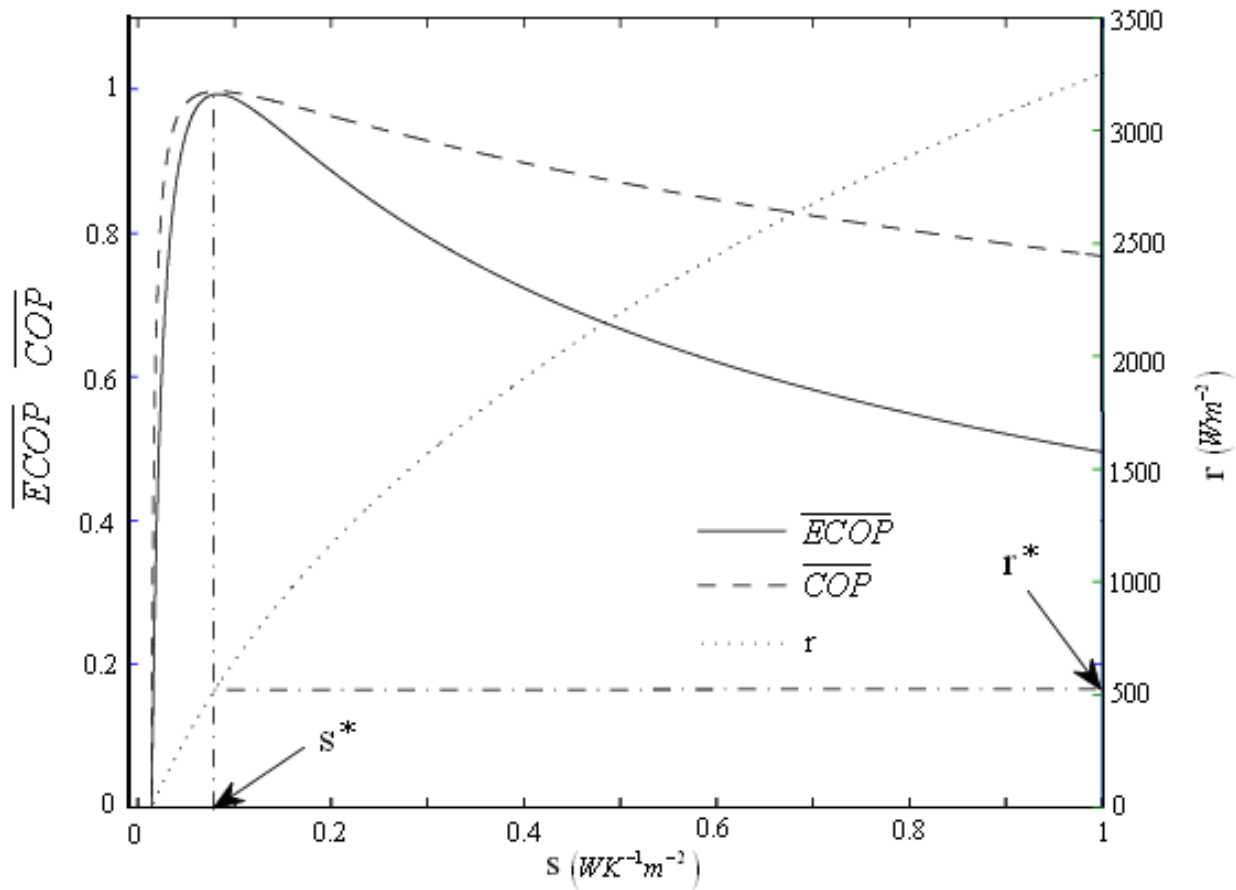
Figure 4. Variation of the normalized $ECOP$, normalized COP and the specific cooling load with respect to the specific entropy generation rate ($T_G = 403K$, $T_L = 273K$, $T_O = 303K$, $T_{env} = 290K$, $U_G = 1163 \ W / m^2 K$, $U_E = 2326 \ W / m^2 K$, $U_O = 4650 \ W / m^2 K$, $K_L = 1082 \ W / K$, $A = 1100 \ m^2$)

5. Conclusion

This chapter presented an analytical method developed to achieve the performance optimization of irreversible three-heat-source absorption refrigeration models having finite-rate of heat transfer, heat leakage and internal irreversibility based on an objective function named ecological coefficient of performance (ECOP). The optimization procedure consists in defining the objective function ECOP in term of the temperatures of the working fluid in the generator, evaporator, condenser and absorber and using extremal conditions to determine analytically the maximum ECOP and the corresponding optimal design parameters. It also established comparative analyses with the COP criterion and shown that the performance parameters at the maximum ECOP and maximum COP are same. The three-heat-source absorption refrigerator cycles are the simplified models of the absorption refrigerators, but the four-heat-source absorption refrigerators cycles are closer to the real absorption refrigerators.

Author details

Paiguy Armand Ngouateu Wouagfack
L2MSP, Department of Physics, University of Dschang, Dschang , Cameroon

Réné Tchinda
LISIE, University Institute of Technology Fotso Victor, University of Dschang, Bandjoun, Cameroon

6. References

[1] Ust, Y.: Ecological performance analysis and optimization of power-generation systems, Ph.D. Thesis Progress Report, Yildiz Technical University, Istanbul. (2004).

[2] Ngouateu, Wouagfack, P. A., Tchinda, R.: Performance optimization of three-heat-source irreversible refrigerator based on a new thermo-ecological criterion. International Journal of Refrigeration. 34, 1008-1015 (2011).

[3] Bejan, A.: Theory of heat transfer-irreversible refrigeration plant. International Journal of Heat Transfer. 32, 1631-1639 (1989).

[4] Ust, Y., Sahin, B., Sogut, O. S.: Performance analysis and optimization of an irreversible Dual cycle based on ecological coefficient of performance (ECOP) criterion. Applied Energy. 82 (1), 23–39 (2005).

[5] Ust, Y., Sahin, Kodal, A.: Ecological coefficient of performance (ECOP) optimization for generalized irreversible Carnot heat engines. Journal of the Energy Institute. 78 (3), 145–151 (2005).

[6] Ust, Y., Sahin,B., Kodal, A.: Performance analysis of an irreversible Brayton heat engine based on ecological coefficient of performance criterion. International Journal of Thermal Science. 45 (1), 94–101 (2006).

[7] Ust, Y., Sogut, O. S., Sahin, B., Durmayaz, A.: Ecological coefficient of performance (ECOP) optimization for an irreversible Brayton heat engine with variable-temperature thermal reservoirs. Journal of the Energy Institute. 79 (1), 47–52 (2006).

[8] Ust, Y., Sahin, B., Kodal, A., Akcay, I. H.: Ecological coefficient of performance analysis and optimization of an irreversible regenerative Brayton heat engine. Applied Energy. 83 (6), 558–572 (2006).

[9] Sogut, O. S., Ust, Y., Sahin, B.: The effects of intercooling and regeneration on the thermo-ecological performance analysis of an irreversible-closed Brayton heat engine with variable-temperature thermal reservoirs. Journal of Physics D: Applied Physics. 39, 4713–4721 (2006).

[10] Ust, Y., Sahin, B.: Performance optimization of irreversible refrigerators based on a new thermo-ecological criterion. International Journal of Refrigeration. 30, 527–534 (2007).

[11] Ust, Y.: Performance analysis and optimization of irreversible air refrigeration cycles based on ecological coefficient of performance criterion. Applied Thermal Engineering. 29, 47–55 (2009).