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Exergoeconomic performance optimization for a steady-flow endoreversible refrigeration model including six typical cycles

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Abstract
The operation of a universal steady flow endoreversible refrigeration cycle model consisting of a constant thermal-capacity heating branch, two constant thermal-capacity cooling branches and two adiabatic branches is viewed as a production process with exergy as its output. The finite time exergoeconomic performance optimization of the refrigeration cycle is investigated by taking profit rate optimization criterion as the objective. The relations between the profit rate and the temperature ratio of working fluid, between the COP (coefficient of performance) and the temperature ratio of working fluid, as well as the optimal relation between profit rate and the COP of the cycle are derived**.** The focus of this paper is to search the compromised optimization between economics (profit rate) and the utilization factor (COP) for endoreversible refrigeration cycles, by searching the optimum COP at maximum profit, which is termed as the finite-time exergoeconomic performance bound. Moreover, performance analysis and optimization of the model are carried out in order to investigate the effect of cycle process on the performance of the cycles using numerical example. The results obtained herein include the performance characteristics of endoreversible Carnot, Diesel, Otto, Atkinson, Dual and Brayton refrigeration cycles.

Keywords: Finite-time thermodynamics; Endoreversible refrigeration cycle; Exergoeconomic performance

1. Introduction
Recently, the analysis and optimization of thermodynamic cycles for different optimization objectives has made tremendous progress by using finite-time thermodynamic theory [1-14]. Finite-time thermodynamics is a powerful tool for the performance analysis and optimization of various cycles. For refrigeration cycles, the performance analysis and optimization have been carried out by taking cooling load, coefficient of performance (COP), specific cooling load, cooling load density, exergy destruction, exergy output, exergy efficiency, and ecological criteria as the optimization objectives in much work, and many meaningful results have been obtained [15-27].
A relatively new method that combines exergy with conventional concepts from long-run engineering economic optimization to evaluate and optimize the design and performance of energy systems is exergoeconomic (or thermoeconomic) analysis [28, 29]. Salamon and Nitzan's work [30] combined the endoreversible model with exergoeconomic analysis. It was termed as finite time exergoeconomic analysis [31-45] to distinguish it from the endoreversible analysis with pure thermodynamic objectives and the exergoeconomic analysis with long-run economic optimization. Similarly, the performance

bound at maximum profit was termed as finite time exergoeconomic performance bound to distinguish it from the finite time thermodynamic performance bound at maximum thermodynamic output.
There have been some papers concerning finite time exergoeconomic optimization for refrigeration cycles [31, 33, 38]. A further step in this paper is to build a universal endoreversible steady flow refrigeration cycle model consisting of a constant thermal-capacity heating branch, two constant thermal-capacity cooling branches and two adiabatic branches with the consideration of heat resistance loss. The finite time exergoeconomic performance of the universal endoreversible refrigeration cycles is studied. The relations between the profit rate and the temperature ratio of working fluid, between the COP and the temperature ratio of working fluid, as well as the optimal relation between profit rate and the COP of the cycle are derived. The focus of this paper is to search the compromise optimization between economics (profit rate) and the energy utilization factor (COP) for the endoreversible refrigeration cycles. Moreover, performance analysis and optimization of the model are carried out in order to investigate the effect of cycle process on the performance of the cycles using numerical examples. The results obtained herein include the performance characteristics of endoreversible Carnot, Diesel, Otto, Atkinson, Dual and Brayton refrigeration cycles.

2. Cycle model
An endoreversible steady flow referigeration cycle operating between an infinite heat sink at temperature T_H and an infinite heat source at temperature T_L is shown in Figure 1. In this T-s diagram, the processes between 2 and 3 , as well as between 5 and 1 are two adiabatic branches; the process between 1 and 2 is a heating branch with constant thermal capacity (mass flow rate and specific heat product) C_{in}; the processes between 3 and 4, and 4 and 5 are two cooling branches with constant thermal capacity C_{out1} and C_{out2}. In addition, the heat conductances (heat transfer coefficient-area product) of the hot- and cold-side heat exchangers are U_{H1}, U_{H2}, and U_L, respectively. The heat exchanger inventory is taken as a constant, that is $U_{H1}+U_{H2}+U_L=U_T$. This cycle model is more generalized. If C_{in}, C_{out1} and C_{out2} have different values, the model can become various special endoreversible refrigeration cycle models.

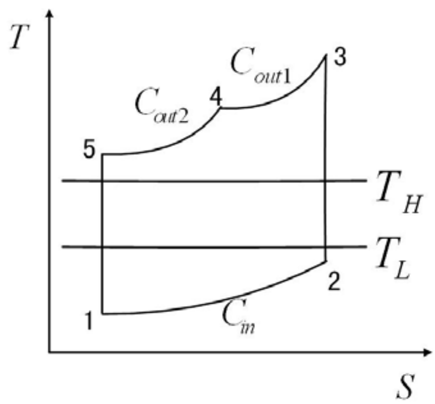

Figure 1. T-s diagram for universal endoreversible cycle model

3. Performance analysis
According to the properties of working fluid and the theory of heat exchangers, the rate of heat transfer Q_{H1} and Q_{H2} released to the heat sink and the rate of heat transfer Q_L (i. e. the cooling load R) supplied by heat source are given, respectively, by

$$Q_H = Q_{H1} + Q_{H2} \tag{1}$$

$$Q_{H1} = \dot{m}C_{out1}(T_3 - T_4) = \dot{m}C_{out1}E_{H1}(T_3 - T_H) \tag{2}$$

$$Q_{H2} = \dot{m}C_{out2}(T_4 - T_5) = \dot{m}C_{out2}E_{H2}(T_4 - T_H) \tag{3}$$

$$R = Q_L = \dot{m}C_{in}(T_2 - T_1) = \dot{m}C_{in}E_L(T_L - T_1) \tag{4}$$

where $\dot{m}$ is mass flow rate of the working fluid, E_{H1}, E_{H2} and E_L are the effectivenesses of the hot- and cold-side heat exchangers, and are defined as

$$E_{H1} = 1 - \exp(-N_{H1}),\ E_{H2} = 1 - \exp(-N_{H2}),\ E_L = 1 - \exp(-N_L) \tag{5}$$

where N_{H1}, N_{H2} and N_L are the numbers of heat transfer units of the hot- and cold-side heat exchangers, and are defined as

$$N_{H1} = U_{H1}/(\dot{m}C_{out1}),\ N_{H2} = U_{H2}/(\dot{m}C_{out2}),\ N_L = U_L/(\dot{m}C_{in}) \tag{6}$$

where U_{H1}, U_{H2} and U_L are the heat conductance, that is, the product of heat transfer coefficient α and heat transfer surface area F

$$U_{H1} = \alpha_{H1}F_{H1},\ U_{H2} = \alpha_{H2}F_{H2},\ U_L = \alpha_L F_L \tag{7}$$

The COP ε of the cycle is

$$\varepsilon = (Q_H/Q_L - 1)^{-1} = \left[(Q_{H1} + Q_{H2})/Q_L - 1\right]^{-1} \tag{8}$$

Combining equations (1) - (3) and (8) gives

$$T_4 = E_{H1}T_H + (1 - E_{H1})xT_L \tag{9}$$

$$T_5 = (1 - E_{H1})(1 - E_{H2})xT_L + \left(E_{H1} + E_{H2} - E_{H1}E_{H2}\right)T_H \tag{10}$$

$$T_1 = T_L - a(xT_L - T_H)\big/\left(1 + \varepsilon^{-1}\right) \tag{11}$$

$$T_2 = E_L T_L + (1 - E_L)T_1 = T_L - a\left(1 - E_L\right)(xT_L - T_H)\big/\left(1 + \varepsilon^{-1}\right) \tag{12}$$

where $a = \left[C_{out1}E_{H1} + C_{out2}E_{H2}\left(1 - E_{H1}\right)\right]\big/\ (C_{in}E_L)$, $x = T_3/T_L$

Consider the endoreversible cycle $1-2-3-4-5-1$. Applying the second law of thermodynamics gives

$$\Delta S = C_{in}\ln\left(T_2/T_1\right) - C_{out1}\ln\left(T_3/T_4\right) - C_{out2}\ln\left(T_4/T_5\right) = 0 \tag{13}$$

From the equation (13), one has

$$T_2 T_4^{\frac{C_{out1} - C_{out2}}{C_{in}}} T_5^{\frac{C_{out2}}{C_{in}}} - T_1 T_3^{\frac{C_{out1}}{C_{in}}} = 0 \tag{14}$$

Combining equations (9) - (14) gives

$$\varepsilon = \frac{T_L a_1 - T_L\left(xT_L\right)^{C_{out1}/C_{in}}}{a\left(xT_L - T_H\right)\left[\left(1 - E_L\right)a_1 - \left(xT_L\right)^{C_{out1}/C_{in}}\right] - T_L a_1 + T_L\left(xT_L\right)^{C_{out1}/C_{in}}} \tag{15}$$

$$R = Q_L = \dot{m}C_{in}E_L \frac{T_L a_1 - T_L\left(xT_L\right)^{C_{out1}/C_{in}}}{\left(1 - E_L\right)a_1 - \left(xT_L\right)^{C_{out1}/C_{in}}} \tag{16}$$

where $a_1 = \left[\left(1 - E_{H1}\right)xT_L + E_{H1}T_H\right]^{(C_{out1} - C_{out2})/C_{in}}\left[(1 - E_{H1})(1 - E_{H2})xT_L + \left(E_{H1} + E_{H2} - E_{H1}E_{H2}\right)T_H\right]^{C_{out2}/C_{in}}$

The required power input P of the cycle is

$$P = Q_H - Q_L = \dot{m}C_{in}E_L\left[a\left(xT_L - T_H\right) - \left(T_L a_1 - T_L\left(xT_L\right)^{C_{out1}/C_{in}}\right)\Big/\left[\left(1 - E_L\right)a_1 - \left(xT_L\right)^{C_{out1}/C_{in}}\right]\right] \tag{17}$$

Assuming the environment temperature is T_0, the rate of exergy output of the refrigeration cycle is:

$$A = Q_L(T_0/T_L - 1) - Q_H(T_0/T_H - 1) = Q_L\eta_1 - Q_H\eta_2 \tag{18}$$

where η_i is the Carnot coefficient of the reservoir *i* $(i = 1,2)$.
So the rate of exergy output of the refrigeration cycle is

$$A = \dot{m}C_{in}E_L\left\{\eta_1\left[T_La_1 - T_L(xT_L)^{C_{out1}/C_{in}}\right]\Big/\left[(1-E_L)a_1 - (xT_L)^{C_{out1}/C_{in}}\right] - \eta_2 a(xT_L - T_H)\right\} \tag{19}$$

Assuming that the prices of exergy output and the work input be ψ_1 and ψ_2, the profit rate of the refrigeration cycle is:

$$\pi = \psi_1 A - \psi_2 P \tag{20}$$

Substituting equations (17) and (19) into equation (20) yields

$$\pi = \dot{m}C_{in}E_L\left\{(\psi_1\eta_1 + \psi_2)\left[T_La_1 - T_L(xT_L)^{C_{out1}/C_{in}}\right]\Big/\left[(1-E_L)a_1 - (xT_L)^{C_{out1}/C_{in}}\right] - a(\psi_1\eta_2 + \psi_2)(xT_L - T_H)\right\} \tag{21}$$

4. Discussions

Equations (15) and (21) are universal relations governing the profit rate function and the COP of the steady flow refrigeration cycle with considerations of heat transfer loss. They include the finite time exergoeconomic performance characteristic of many kinds of refrigeration cycles.
When $C_{in} = C_{out1} = C_{out2} = C$ (C_V or C_P), equations (15) and (21) become:

$$\varepsilon = \frac{T_LE_L(T_H - xT_L)}{(xT_L - T_H)\left[(E_{H2}E_L - E_{H2} - E_L)xT_L + E_{H2}(1-E_L)T_H\right] - T_LE_L(T_H - xT_L)} \tag{22}$$

$$\pi = \dot{m}CE_{H2}(xT_L - T_H)\left[(\psi_1\eta_1 + \psi_2)\big/(1+\varepsilon^{-1}) - (\psi_1\eta_2 + \psi_2)\right] \tag{23}$$

Equations (22) and (23) are the finite time exergoeconomic performance characteristic of a steady flow endoreversible Otto ($C = C_V$) or Brayton ($C = C_P$) refrigeration cycle.
When $C_{out1} = C_{out2} = C_v$ and $C_{in} = C_p$, $E_{H1} = 0$, and equations (15) and (21) become:

$$\varepsilon = \frac{T_La_1' - T_L(xT_L)^{1/k}}{a'(xT_L - T_H)\left[(1-E_L)a_1' - (xT_L)^{1/k}\right] - T_La_1' + T_L(xT_L)^{1/k}} \tag{24}$$

$$\pi = \dot{m}C_pE_L\left\{(\psi_1\eta_1 + \psi_2)\left[T_La_1' - T_L(xT_L)^{1/k}\right]\Big/\left[(1-E_L)a_1' - (xT_L)^{1/k}\right] - a'(\psi_1\eta_2 + \psi_2)(xT_L - T_H)\right\} \tag{25}$$

where $a' = E_{H2}/(kE_L)$, $a_1' = xT_L\left[(1-E_{H2})xT_L + E_{H2}T_H\right]^{1/k}$.
Equations (24) and (25) are the finite time exergoeconomic performance characteristic of a steady flow endoreversible Atkinson refrigeration cycle.
When $C_{out1} = C_{out2} = C_p$ and $C_{in} = C_v$, $E_{H1} = 0$, and equations (15) and (21) become:

$$\varepsilon = \frac{T_La_1'' - T_L(xT_L)^k}{a''(xT_L - T_H)\left[(1-E_L)a_1'' - (xT_L)^{1/k}\right] - T_La_1'' + T_L(xT_L)^k} \tag{26}$$

$$\pi = \dot{m}C_vE_L\left\{(\psi_1\eta_1 + \psi_2)\left[T_La_1'' - T_L(xT_L)^k\right]\Big/\left[(1-E_L)a_1'' - (xT_L)^k\right] - a''(\psi_1\eta_2 + \psi_2)(xT_L - T_H)\right\} \tag{27}$$

where $a'' = kE_{H2}/E_L$, $a_1'' = xT_L\left[(1-E_{H2})xT_L + E_{H2}T_H\right]^k$.

Equations (26) and (27) are the finite time exergoeconomic performance characteristic of a steady flow endoreversible Diesel refrigeration cycle.
When $C_{out1} = C_p$, $C_{out2} = C_v$ and $C_{in} = C_v$, equations (15) and (21) become:

$$\varepsilon = \frac{T_L a_1''' - T_L (xT_L)^k}{a'''(xT_L - T_H)\left[(1-E_L)a_1''' - (xT_L)^{1/k}\right] - T_L a_1''' + T_L (xT_L)^k} \tag{28}$$

$$\pi = \dot{m}C_v E_L \left\{(\psi_1\eta_1 + \psi_2)\left[T_L a_1''' - T_L (xT_L)^k\right] \Big/ \left[(1-E_L)a_1''' - (xT_L)^k\right] - a'''(\psi_1\eta_2 + \psi_2)(xT_L - T_H)\right\} \tag{29}$$

where $a''' = \left[kE_{H1} + E_{H2}(1-E_{H1})\right]/E_L$, $a_1''' = \left[(1-E_{H1})xT_L + E_{H1}T_H\right]^{(k-1)}\left[(1-E_{H1})(1-E_{H2})xT_L + (E_{H1} + E_{H2} - E_{H1}E_{H2})T_H\right]$.
Equations (28) and (29) are the finite time exergoeconomic performance characteristic of a steady flow endoreversible Dual refrigeration cycle.
When $C_{in} = C_{out1} = C_{out2} \to \infty$, equations (15) and (21) are the finite time exergoeconomic performance characteristic of the endoreversible Carnot refrigeration cycle [31, 38].
Equations (15) and (21) are the major performance relations for the endoreversible refrigeration cycle coupled to two constant-temperature reservoirs. They determine the relations between the COP and the temperature ratio of the working fluid, between the profit rate and the temperature ratio of the working fluid, as well as between the profit rate and the COP. Finding the optimum f ($f = U_L/U_H$) with the constraint of $U_{H1} + U_{H2} + U_L = U_H + U_L = U_T$, one may obtain the optimal profit rate (π_{opt}) and the optimal COP for the fixed temperature ratio of the working fluid. The optimal COP is a monotonically increasing function of the temperature ratio of the working fluid, while there exists a maximum profit rate for an optimal temperature ratio of the working fluid. Maximizing π_{opt} with respect to x by setting $\partial\pi_{opt}/\partial x = 0$ in Eq. (21) yields the maximum profit rate $\pi_{\max}$ and the optimal temperature ratio of the working fluid x_{opt}. Furthermore, substituting x_{opt} into equation (15) after optimizing U_L/U_H yields ε_m, which is the finite-time thermodynamic exergoeconomic bound.
The idea mentioned above may be applied to various endoreversible cycles, including Brayton cycle by setting $C_{in} = C_{out} = C_P$ or Otto cycle by setting $C_{in} = C_{out} = C_v$. For the endoreversible Brayton or Otto refrigeration cycle, when $U_H = U_L = U_T/2$, the profit rate approaches its optimum value for a given COP. The relation between the optimal profit rate and COP is:

$$\pi_{opt} = \dot{m}C\left[(1+\varepsilon^{-1})T_L - T_H\right]\left[\frac{1}{1+\varepsilon^{-1}}(\psi_1\eta_1 + \psi_2) - (\psi_1\eta_2 + \psi_2)\right]\{\exp[U_T/(2\dot{m}C)] - 1\}/\{\exp[U_T/(2\dot{m}C)] + 1\} \tag{30}$$

Maximizing π_{opt} with respect to ε by setting $\partial\pi_{opt}/\partial\varepsilon = 0$ in Eq. (25) directly yields the maximum profit rate and the corresponding optimal COP ε_m, that is, the finite-time thermodynamic exergoeconomic bound:

$$\pi_{\max} = \dot{m}C\left\{\left[T_H T_L(\psi_1\eta_1 + \psi_2)/(\psi_1\eta_2 + \psi_2)\right]^{0.5} - T_H\right\}\left\{\left[T_L(\psi_1\eta_1 + \psi_2)(\psi_1\eta_2 + \psi_2)/T_H\right]^{0.5} - (\psi_1\eta_2 + \psi_2)\right\}$$
$$\{\exp[U_T/(2\dot{m}C)] - 1\}/\{\exp[U_T/(2\dot{m}C)] + 1\} \tag{31}$$

$$\varepsilon_m = \left\{\left\{T_H(\psi_1\eta_1 + \psi_2)/\left[(\psi_1\eta_2 + \psi_2)T_L\right]\right\}^{0.5} - 1\right\}^{-1} \tag{32}$$

The finite-time thermodynamic exergoeconomic bound (ε_m) is different from the classical reversible bound and the finite-time thermodynamic bound at the maximum cooling load output. It is dependent on T_H, T_L, T_0 and ψ_2/ψ_1.
Note that for the process to be potential profitable, the following relationship must exist: $0 < \psi_2/\psi_1 < 1$, because one unit of work input must give rise to at least one unit of exergy output. As the price of exergy output becomes very large compared with the price of the work input, i.e. $\psi_2/\psi_1 \to 0$, equation (21) becomes

$$\pi = \psi_1 A \tag{33}$$

That is the profit rate maximization approaches the exergy output maximization, where A is the rate of exergy output of the universal endoreversible refrigeration cycle.
On the other hand, as the price of exergy output approaches the price of the work input, i.e. $\psi_2/\psi_1 \to 1$, equation (21) becomes

$$\pi = -\psi_1 T_0 \sigma \tag{34}$$

where σ is the rate of entropy production of the universal endoreversible refrigeration cycle. That is the profit rate maximization approaches the entropy production rate minimization, in other word, the minimum waste of exergy. Equation (34) indicates that the refrigerator is not profitable regardless of the COP is at which the refrigerator is operating. Only the refrigerator is operating reversibly ($\varepsilon = \varepsilon_C$) will the revenue equal the cost, and then the maximum profit rate will equal zero. The corresponding rate of entropy production is also zero.

5. Numerical examples
To illustrate the preceding analysis, numerical examples are provided. In the calculations, it is set that $T_0 = 298.15K$, $T_H = T_0$, $\dot{m} = 1.1165kg/s$, $E_H = E_L = 0.9$, $c_v = 0.7166kJ/(kg \cdot K)$, $c_p = 1.0032kJ/(kg \cdot K)$, and $\tau = T_H/T_L = 1.4$. A dimensionless profit rate is defined as $\Pi = \pi/(T_L E_L C_V \psi_2)$.
Figures 2-6 show the effects of the price ratio on the dimensionless profit rate versus temperature ratio of the working fluid and the COP versus temperature ratio of the working fluid for Otto, Diesel, Atkinson, Dual and Brayton refrigeration cycles. Figure 7 shows the effects of the price ratio on the dimensionless profit rate versus the COP for five cycles.
From the Figures 2-5, one can see that the COP decreases monotonically when x increases for any one of the five cycles, while the profit rate versus x is parabolic-like one. When $\psi_1/\psi_2 = 1.0$, the profit rate maximization approaches zero, this means that the refrigerator is not profitable in any case. From Figure 7, one can see that , when $\psi_1/\psi_2 = 1.0$, the profit rate approaches to zero as the COP increases; when $\psi_1/\psi_2 > 1$, the curves of the dimensionless profit rate versus the COP are parabolic-like ones. The COP at the maximum profit rate is the finite-time exergoeconomic performance bound. Therefore, from the above analysis, one can find that the effect of the price ratio ψ_1/ψ_2 on the finite-time exergoeconomic performance bound is larger: when $\psi_1/\psi_2 = 1.0$, the profit rate approaches to zero as the COP increases; when $\psi_1/\psi_2 \gg 1$, the finite-time exergoeconomic performance bound of the endoreversible refrigerator approaches to the finite-time thermodynamic performance bound. Therefore, the finite-time exergoeconomic performance bound (ε_π) lies between the finite-time thermodynamic performance bound and the reversible performance bound. ε_π is related to the latter two through the price ratio, and the associated COP bounds are the upper and lower limits of ε_π.

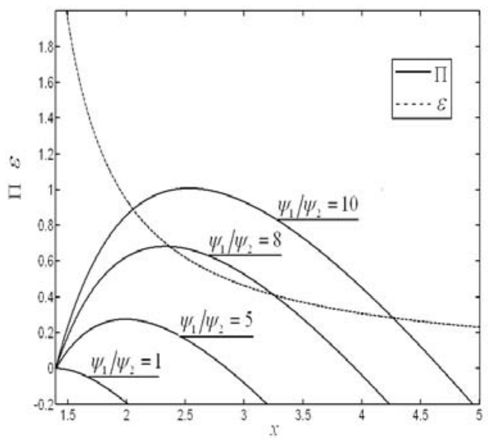

Figure 2. Dimensionless profit rate and the COP characteristic for Otto cycle

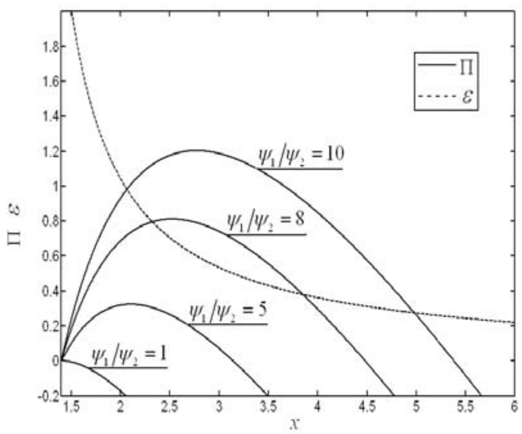

Figure 3. Dimensionless profit rate and the COP characteristic for Atkinson cycle

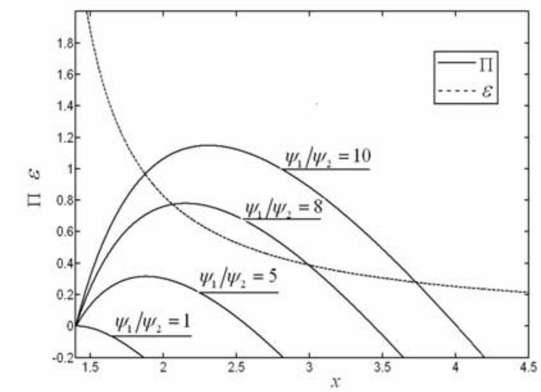

Figure 4. Dimensionless profit rate and the COP characteristic for Diesel cycle

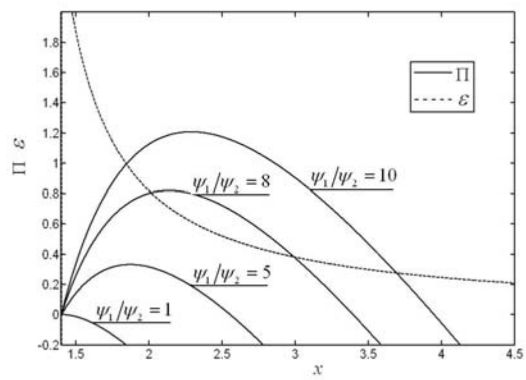

Figure 5. Dimensionless profit rate and the COP characteristic for Dual cycle

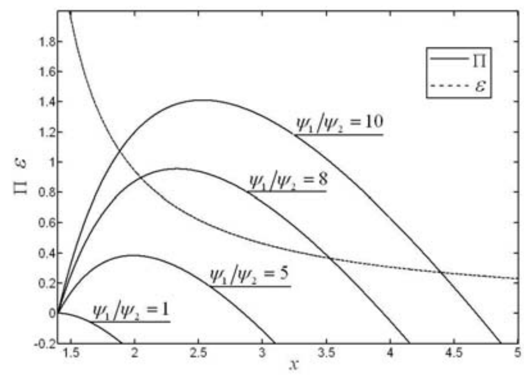

Figure 6. Dimensionless profit rate and the COP characteristic for Brayton cycle

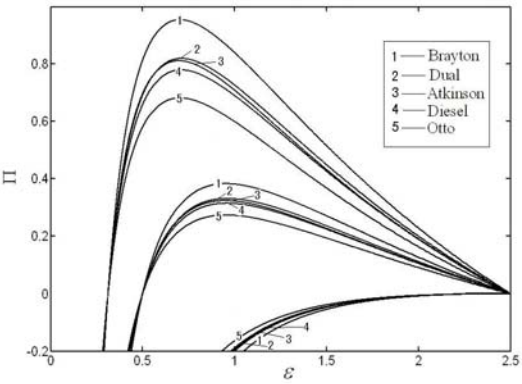

Figure 7. Dimensionless profit rate versus the COP characteristic for five cycles

6. Conclusion

Economics plays a major role in the thermal power and cryogenics industry. This paper combines finite time thermodynamics with exergoeconomics to form a new analysis of universal endoreversible refrigeration cycle model. One seeks the economic optimization objective function instead of pure thermodynamic parameters by viewing the refrigerator as a production process. It is shown that the economic and thermodynamic optimization converged in the limits $\psi_1/\psi_2 \to 0$ and $\psi_1/\psi_2 \to 1$. Analysis and optimization of the model are carried out in order to investigate the effect of cycle process on the performance of the cycles using numerical examples. The results obtained herein include the performance characteristics of endoreversible Carnot, Diesel, Otto, Atkinson, Dual and Brayton refrigeration cycles.

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References

[1] Andresen B. Finite-Time Thermodynamics. Physics Laboratory , University of Copenhagen, 1983.
[2] Andresen B, Berry R S, Ondrechen M J, Salamon P. Thermodynamics for processes in finite time. Acc. Chem. Res. 1984, 17(8): 266-271.
[3] Sieniutycz S, Salamon P. Advances in Thermodynamics. Volume 4: Finite Time Thermodynamics and Thermoeconomics. New York: Taylor & Francis, 1990.
[4] De Vos A. Endoreversible Thermodynamics of Solar Energy Conversion. Oxford: Oxford University, 1992.
[5] Sieniutycz S, Shiner J S. Thermodynamics of irreversible processes and its relation to chemical engineering: Second law analyses and finite time thermodynamics. J. Non-Equilib. Thermodyn., 1994, 19(4): 303-348.
[6] Bejan A. Entropy generation minimization: The new thermodynamics of finite-size device and finite-time processes. J. Appl. Phys., 1996, 79(3): 1191-1218.
[7] Hoffmann K H, Burzler J M, Schubert S. Endoreversible thermodynamics. J. Non- Equilib. Thermodyn., 1997, 22(4): 311-355.
[8] Berry R S, Kazakov V A, Sieniutycz S, Szwast Z, Tsirlin A M. Thermodynamic Optimization of Finite Time Processes. Chichester: Wiley, 1999.
[9] Chen L, Wu C, Sun F. Finite time thermodynamic optimization or entropy generation minimization of energy systems. J. Non-Equilib. Thermodyn., 1999, 24(4): 327-359.
[10] Durmayaz A, Sogut O S, Sahin B and Yavuz H. Optimization of thermal systems based on finite-time thermodynamics and thermoeconomics. Progress Energy & Combustion Science, 2004, 30(2): 175-217.

[11] Chen L, Sun F. Advances in Finite Time Thermodynamics Analysis and Optimization . New York: Nova. Science Publishers, 2004.
[12] Chen L. Finite-Time Thermodynamic Analysis of Irreversible Processes and Cycles. Higher Education Press, Beijing, 2005.
[13] Sieniutycz S, Jezowski J. Energy Optimization in Process Systems. Oxford: Elsevier; 2009.
[14] Li J, Chen L, Sun F. Ecological performance of a generalized irreversible Carnot heat engine with complex heat transfer law. Int. J. Energy and Environment, 2011, 2(1): 57-70.
[15] Goth Y, Feidt M. Optimum COP for endoreversible heat pump or refrigerating machine. C R Acad. Sc. Paris, 1986, 303(1):19-24.
[16] Feidt M. Sur une systematique des cycles imparfaits. Entropie, 1997, 205(1): 53-61.
[17] Klein S A. Design considerations for refrigeration cycles. Int. J. Refrig., 1992, 15(3): 181-185.
[18] Wu C. Maximum obtainable specific cooling load of a refrigerator. Energy Convers. Mgnt., 1995, 36(1): 7-10.
[19] Chiou J S, Liu C J, Chen C K. The performance of an irreversible Carnot refrigeration cycle. J. Phys. D: Appl. Phys., 1995, 28(7): 1314-1318.
[20] Ait-Ali M A. The maximum coefficient of performance of internally irreversible refrigerators and heat pumps. J. Phys. D: Appl. Phys., 1996, 29(4): 975-980.
[21] Radcenco V, Vargas J V C, Bejan A, Lim J S. Two design aspects of defrosting refrigerators. Int. J. Refrig., 1995, 18(2):76-86.
[22] Chen C K, Su Y F. Exergetic efficiency optimization for an irreversible Brayton refrigeration cycle. Int. J. Thermal Sci., 2005, 44(3): 303-310.
[23] Su Y F, Chen C K. Exergetic efficiency optimization of a refrigeration system with multi-irreversibilities. Proceedings of the IMECH E Part C, Journal of Mechanical Engineering Science, 2006, 220(8): 1179-1188.
[24] Zhou S, Chen L, Sun F, Wu C. Theoretical optimization of a regenerated air refrigerator. J. Phys. D: Appl. Phys., 2003, 36(18): 2304-2311.
[25] Wu F, Chen L, Sun F, Wu C, Guo F. Optimal performance of an irreversible quantum Brayton refrigerator with ideal Bose gases. Physica Scripta, 2006, 73(5): 452-457.
[26] Chen L, Zhu X, Sun F, Wu C. Ecological optimization for generalized irreversible Carnot refrigerators. J. Phys. D: Appl. Phys., 2005, 38(1): 113-118.
[27] Tu Y, Chen L, Sun F, Wu C. Exergy-based ecological optimization for an endoreversible Brayton refrigeration cycle. Int. J. Exergy, 2006, 3(2): 191-201.
[28] Tsatsaronts G. Thermoeconomic analysis and optimization of energy systems. Progress in Energy and Combustion Science, 1993, 19(3): 227-257.
[29] El-Sayed M. The Thermoeconomics of Energy Conversion. London: Elsevier, 2003.
[30] Salamon P, Nitzan A. Finite time optimizations of a Newton's law Carnot cycle. J. Chem. Phys., 1981, 74(6): 3546-3560.
[31] Chen L, Sun F, Chen W. Finite time exergoeconomic performance bound and optimization criteria for two-heat-reservoir refrigerators. Chinese Sci. Bull., 1991, 36(2): 156-157.
[32] Wu C, Chen L, Sun F. Effect of heat transfer law on finite time exergoeconomic performance of heat engines. Energy, The Int. J., 1996, 21(12): 1127-1134.
[33] Chen L, Sun F, Wu C. Maximum profit performance of an absorption refrigerator. Int. J. Energy, Environment, Economics, 1996, 4(1): 1-7.
[34] Chen L, Sun F, Wu C. Exergoeconomic performance bound and optimization criteria for heat engines. Int. J. Ambient Energy, 1997, 18(4): 216-218.
[35] Wu C, Chen L and Sun F. Effect of heat transfer law on finite time exergoeconomic performance of a Carnot heat pump. Energy Conves. Management, 1998, 39(7): 579-588.
[36] Chen L, Wu C, Sun F, Cao S. Maximum profit performance of a three- heat-reservoir heat pump. Int. J. Energy Research, 1999, 23(9): 773-777.
[37] Wu F, Chen L, Sun F, Wu C. Finite-time exergoeconomic performance bound for a quantum Stirling engine. Int. J. Engineering Science, 2000, 38(2): 239-247.
[38] Chen L, Wu C, Sun F. Effect of heat transfer law on finite time exergoeconomic performance of a Carnot refrigerator. Exergy, An Int. J., 2001, 1(4): 295-302.
[39] Chen L, Sun F, Wu C. Maximum profit performance for generalized irreversible Carnot engines. Appl. Energy, 2004, 79(1): 15-25.

[40] Zheng Z, Chen L, Sun F, Wu C. Maximum profit performance for a class of universal steady flow endoreversible heat engine cycles. Int. J. Ambient Energy, 2006, 27(1): 29-36.
[41] Feng H, Chen L, Sun F. Finite time exergoeconomic performance optimization for an irreversible universal steady flow variable-temperature heat reservoir heat pump cycle model. Int. J. Energy and Environment, 2010, 1(6): 969-986.
[42] Kan X, Chen L, Sun F, Wu F. Finite time exergoeconomic performance optimization of a thermoacoustic heat engine. Int. J. Energy and Environment, 2011, 2(1): 85-98.
[43] Li J, Chen L, Sun F. Finite-time exergoeconomic performance of an endoreversible Carnot heat engine with complex heat transfer law. Int. J. Energy and Environment, 2011, 2(1): 171-178.
[44] Chen L, Yang B, Sun F. Exergoeconomic performance optimization of an endoreversible intercooled regenerated Brayton cogeneration plant. Part 1: thermodynamic model and parameter analyses. Int. J. Energy and Environment, 2011, 2(2): 199-210.
[45] Yang B, Chen L, Sun F. Exergoeconomic performance optimization of an endoreversible intercooled regenerated Brayton cogeneration plant. Part 2: heat conductance allocation and pressure ratio optimization. Int. J. Energy and Environment, 2011, 2(2): 211-218.

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