MODELS FOR OPTIMUM THERMO-ECOLOGICAL CRITERIA OF ACTUAL THERMAL CYCLES

by

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In this study, the ecological optimization point of irreversible thermal cycles (refrigerator, heat pump, and power cycles) was investigated. The importance of ecological optimization is to propose a way to use fuels and energy source more efficiently because of an increasing energy need and environmental pollution. It provides this by maximizing obtained (or minimizing supplied) work and minimizing entropy generation for irreversible (actual) thermal cycles. In this research, ecological optimization was defined for all basic irreversible thermal cycles, by using the first and second laws of thermodynamics. Finally, the ecological optimization was defined in thermodynamic cycles and results were given to show the effects of the cycles' ecological optimization point, efficiency, coefficient of performance, and power output (or input), and exergy destruction.

Key words: *ecological optimization, irreversibility, refrigeration, heat pump, power cycles*

Introduction

In order to decrease environmental problems occurring in the world and to meet the increasing energy need, more efficient and environment friendly cycles must be designed. Finite time thermodynamics have been widely used from the 1970's onwards in order to find the performance limits of thermal systems and detect optimum values of the system. Angulo-Brown [1] on the other hand recommended ecological criteria, which are another optimization method. Ecological criteria are expressed as $\dot{E} = \dot{W} - T_\mathrm{L}\dot{\sigma}$ for finite time Camot engines, and here T_L is the cold reservoir temperature and $\dot{\sigma}$ is the entropy production of the system. Yan [2] on the other hand, recommended a more reasonable principle $\dot{E} = \dot{W} - T_\mathrm{o}\dot{\sigma}$); here T_o is the environmental temperature. This principle enables us to obtain minimum exergy destruction and maximum power output (minimum power input for heat pump and refrigeration cycles). In the literature, many studies exist in which ecological criteria are applied [3-29]. Yan and Lin [6] evaluated heat changer surface fields for an irreversible refrigerator with three heat sources; while Huang *et al*. [7] did the same for a refrigerator with four heat level irreversible absorption for coefficient of performance (*COP*) and ecological criteria. Ecological criteria used in those areas are energy based and shown as $\dot{E}^* = \dot{Q}_\mathrm{L} - \varphi_\mathrm{C} T_\mathrm{L}\dot{\sigma}$. φ_C, T_L, $\dot{Q}_\mathrm{L}$, and $\dot{\sigma}$ terms are Carnot *COP*, cold reservoir temperature, refrigeration load, and entropy production, respectively. Ust *et al*. [30-32] recommended a new principle, which they called ecological performance coefficient (ECOP). This principle is shown as ECOP = $\dot{W}/T_0\dot{\sigma}$ and they examined various thermodynamic cycles us-

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ing this principle. In addition to those, optimizations for many cycles including Stirling--Ericsson heat machines, internal combustion engines, and refrigeration cycles were made [33-39].

In this study, all basic thermodynamic cycles were optimized using ecological criteria and while doing this, with an aim to choosing simple and easily applicable methods. Existing ecological optimization methods were enhanced and facilitated. Contrary to other studies, here, an internal irreversibility parameter was included in the analysis, instead of accepting it as a different parameter. Researchers are expected to design ecological systems more easily by using these methods.

Thermodynamic analysis

By using the first law of thermodynamics, power output in ecological function was defined as the difference between the heat entering the system and the heat thrown out ($\dot{W} = \dot{Q}_{\mathrm{H}} - \dot{Q}_{\mathrm{L}}$) and was generalized in the form of eq. (1); this equation contains power output and exergy destruction terms. After, eq. (1) was applied to all basic thermodynamic cycles in next two sections:

$$\dot{E} = \dot{Q}_{\mathrm{H}} - \dot{Q}_{\mathrm{L}} - T_{\mathrm{o}}\dot{\sigma} \tag{1}$$

Design parameters: selected as evaporator temperature for irreversible Rankine cycle, condenser temperature for irreversible heat pump and refrigeration cycles, non-dimensional compression parameter for SI, CI, Stirling cycles and non-dimensional pressure parameter for irreversible Brayton and Ericsson cycles. Recognitions made for cycles are:

for irreversible Rankine, heat pump and refrigeration cycles

– all processes are irreversible,
– systems follow a continuous pattern, and
– heat exchangers' dimensions are limited and convection coefficients are constant, and

for irreversible SI, CI, Stirling-Ericsson and Brayton cycles

– all processes are irreversible,
– specific heats are constant and average specific heats are used,
– piston friction is neglected,
– working fluid is air, which is the ideal gas, and
– polytrophic coefficients are constant and an average polytrophic coefficient is identified for each cycle.

Environmental temperature (T_{o}) is accepted as 298.15 K for all cycles.

An internal irreversibility parameter was defined for all cycles by using the second law of thermodynamics. The second law of thermodynamics can be expressed as:

$$\frac{\dot{Q}_{\mathrm{H}}}{T_{\mathrm{H,WF}}} - \frac{\dot{Q}_{\mathrm{L}}}{T_{\mathrm{L,WF}}} \leq 0 \tag{2}$$

In order to make this inequality an equality, a coefficient must be defined. This coefficient is an internal irreversibility parameter. With the addition of the internal irreversibility parameter, the the equality in eq. (2), transforms into the form:

$$I\frac{\dot{Q}_{\mathrm{H}}}{T_{\mathrm{H,WF}}} = \frac{\dot{Q}_{\mathrm{L}}}{T_{\mathrm{L,WF}}} \tag{3}$$

Here I is the internal irreversibility parameter of the system and can be expressed as by using eq. (3):

$$I = \frac{\dot{Q}_L T_{H,WF}}{\dot{Q}_L T_{L,WF}} \tag{4}$$

As seen in the following sections, the function, which will be optimized, can be simplified by writing an internal irreversibility parameter, as seen in eq. (4), and among the optimum values found, only heat and heat conductivity parameters will remain. The internal irreversibility parameter is greater than one in all irreversible (actual) cycles ($I > 1$), while in reversible cycles it will be equal to zero ($I = 0$).

Power generation cycles

Thermodynamic analysis of irreversible Rankine cycle

Recently, Rankine cycle has gained importance thanks to an increase in the search for new energy technologies and more efficient systems, because this cycle can work with organic working fluids (hydrocarbons, siloxanes or refrigerants) that enable the use of energy sources at low temperatures. In addition to this, facilities which work with a conventional Rankine cycle that uses coal as fuel are still widely used. Emissions that are sent out by these facilities to the environment and entropy production must be reduced to a minimum. In fig. 1, the *T-s* diagram of an irreversible Rankine cycle is seen.

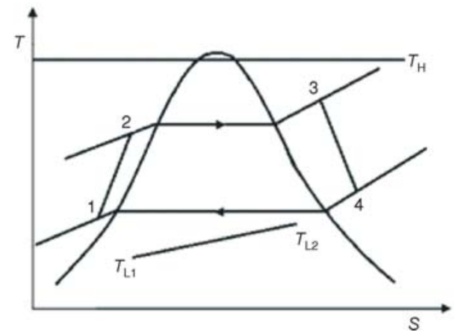

Figure 1. *T-s* diagram of irreversible Rankine cycle

From the first law of thermodynamics:

$$\dot{Q}_{H,R} - \dot{Q}_{L,R} = \dot{W}_{net,R} \tag{5}$$

Here $\dot{Q}_{H,R}$ and $\dot{Q}_{L,R}$ can be obtained similar to the finite time thermodynamic used in refs. [40, 41]:

$$\dot{Q}_{H,R} = \frac{Q_{H,R}}{t_{1,R}} = \phi_{E,R}(T_{H,R} - T_{E,R}) \quad \text{and} \quad \dot{Q}_{L,R} = \frac{Q_{L,R}}{t_{2,R}} = \phi_{C,R}(T_{C,R} - T_{L,R}) \tag{6}$$

$$t_R = t_{1,R} + t_{2,R} \tag{7}$$

Here, $T_{H,R}$ is the combustion temperature and $T_{L,R}$:

$$T_{L,R} = \frac{T_{L1,R} + T_{L2,R}}{2} \tag{8}$$

From the second law of thermodynamics:

$$\frac{\dot{Q}_{H,R}}{T_{E,R}} - \frac{\dot{Q}_{L,R}}{T_{C,R}} \le 0 \quad \text{or} \quad I_R \frac{\dot{Q}_{H,R}}{T_{E,R}} = \frac{\dot{Q}_{L,R}}{T_{C,R}} \tag{9}$$

Here, I_R is the internal irreversibility coefficient of the Rankine cycle. I_R and $\dot{Q}_{H,R}$ can be obtained, respectively, by using eqs. (4) and (5)-(9):

$$I_R = \frac{\phi_{C,R} T_{E,R}(T_{C,R} - T_{L,R})}{\phi_{E,R} T_{C,R}(T_{H,R} - T_{E,R})} \tag{10}$$

$$\dot{Q}_{H,R} = \frac{1}{\dfrac{1}{\phi_{\mathrm{E,R}}(T_{\mathrm{H,R}} - T_{\mathrm{E,R}})} + \dfrac{I_{\mathrm{R}}\dfrac{T_{\mathrm{C,R}}}{T_{\mathrm{E,R}}}}{\phi_{\mathrm{C,R}}(T_{\mathrm{L,R}} - T_{\mathrm{C,R}})}} \tag{11}$$

By using eqs. (1) and (5)-(11), $\dot{E}_{\mathrm{R}}$ is obtained as:

$$\dot{E}_R = \frac{\phi_{\mathrm{C,R}} T_{\mathrm{H,R}} (T_{\mathrm{L,R}} - T_{\mathrm{C,R}})(T_0 - T_{\mathrm{L,R}}) + \phi_{\mathrm{E,R}} T_{\mathrm{L,R}} (T_{\mathrm{H,R}} - T_{\mathrm{E,R}})(T_{\mathrm{o}} - T_{\mathrm{H,R}})}{T_{\mathrm{H,R}}(1 + T_{\mathrm{E,HR}}^2 - T_{\mathrm{E,HR}} T_{\mathrm{H,R}} + T_{\mathrm{HR}}^2) T_{\mathrm{L,R}}} \tag{12}$$

Here, evaporator temperature is selected as design parameter and in order to detect the optimum evaporator temperature, a derivative of the Rankine cycle according to the evaporator temperature is equalized to zero ($\partial\dot{E}_{\mathrm{R}}/\partial T_{\mathrm{E,R}} = 0$). The corresponding value is:

$$T_{op,E,R} = \frac{T_{\mathrm{L,R}} T_{\mathrm{H,R}} \Delta - \sqrt{\phi_{\mathrm{E,R}}^2 T_{\mathrm{L,R}}^2 (T_{\mathrm{H,R}} - 2T_{\mathrm{o}})^2 + \phi_{\mathrm{C,R}}^2 T_{\mathrm{H,R}} (T_{\mathrm{C,R}} - T_{\mathrm{L,R}})^2 (T_{\mathrm{L,R}} - T_{\mathrm{o}})^2 T_{\mathrm{H,R}} T_{\mathrm{o}} \nabla}}{\phi_{\mathrm{E,R}} T_{\mathrm{L,R}} (T_{\mathrm{H,R}} - 2T_{\mathrm{o}})} \tag{13}$$

$$\Delta = (T_{\mathrm{H,R}} \phi_{\mathrm{E,R}} + T_{\mathrm{L,R}} K_{\mathrm{C,R}} - T_{\mathrm{C,R}} \phi_{\mathrm{C,R}}) \tag{14}$$

$$\nabla = (T_{\mathrm{C,R}} \phi_{\mathrm{C,R}} - T_{\mathrm{L,R}} \phi_{\mathrm{C,R}} - T_{\mathrm{L,R}} \phi_{\mathrm{C,R}} - T_{\mathrm{L,R}} \phi_{\mathrm{E,R}}) \tag{15}$$

Thermodynamic analysis of irreversible SI and CI cycles

Internal combustion engines are the most widely used power cycles. When the emissions released by these cycles are taken into consideration, they have serious negative effects on the environment. Although it is estimated that fossil fuels will deplete soon, it can be said that internal combustible engines could be used in power production, by considering the studies that are carried out on fuels which can be produced in a laboratory. If we take into account the harmful effects on the environment and relatively low efficiencies, the importance of designing internal combustible engines in an environmental friendly way becomes apparent. In this section, SI and CI engines were optimized by taking ecological criteria into consideration. Figure 2 show the *T-s* diagram of analyzed engines.

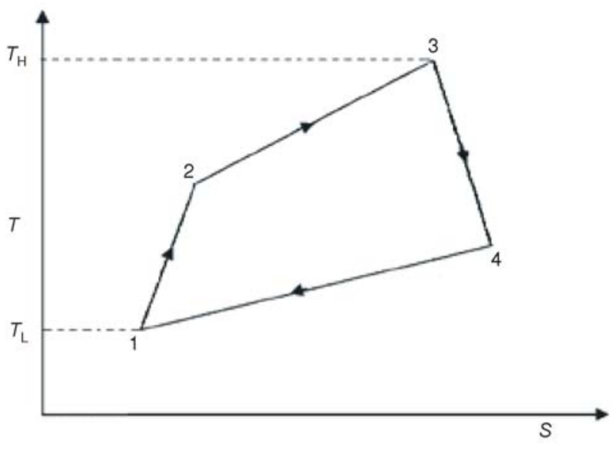

Figure 2. *T-s* diagram of irreversible SI, CI, and Brayton cycles [42, 43]

– Irreversible SI engine

From the first law of thermodynamics:

$$\dot{Q}_{\mathrm{H,SI}} - \dot{Q}_{\mathrm{L,SI}} = \dot{W}_{\mathrm{net,SI}} \tag{16}$$

$$\dot{Q}_{\mathrm{H,SI}} = \frac{\dot{m}_{\mathrm{SI}} k_{\mathrm{SI}}}{1 + k_{\mathrm{SI}}} (T_{3,\mathrm{SI}} - T_{2,\mathrm{SI}}), \quad \dot{Q}_{\mathrm{L,SI}} = \frac{\dot{m}_{\mathrm{SI}} K_{\mathrm{SI}}}{1 + k_{\mathrm{SI}}} (T_{4,\mathrm{SI}} - T_{1,\mathrm{SI}}) \tag{17}$$

For SI engine:

$$\varepsilon_{\mathrm{SI}} = \frac{V_{1,\mathrm{SI}}}{V_{2,\mathrm{SI}}} = \frac{V_{4,\mathrm{SI}}}{V_{3,\mathrm{SI}}}, \quad \frac{T_{2,\mathrm{SI}}}{T_{1,\mathrm{SI}}} = \frac{T_{3,\mathrm{SI}}}{T_{4,\mathrm{SI}}} = x_{\mathrm{SI}}, \quad x_{\mathrm{SI}} = (\varepsilon_{\mathrm{SI}})^{n_{SI}-1}, \quad \frac{T_{3,\mathrm{SI}}}{T_{1,\mathrm{SI}}} = \alpha_{\mathrm{SI}},$$
$$T_{2,\mathrm{SI}} = T_{1,\mathrm{SI}} x_{\mathrm{SI}}, \quad T_{4,\mathrm{SI}} = \frac{T_{3,\mathrm{SI}}}{x_{\mathrm{SI}}}, \quad K_{SI} = c_{\mathrm{v,SI,2\text{-}3}} + c_{\mathrm{v,SI,4\text{-}1}}, \quad k_{\mathrm{SI}} = \frac{c_{\mathrm{v,2\text{-}3,SI}}}{c_{\mathrm{v,4\text{-}1,SI}}} \tag{18}$$

From the second law of thermodynamics:

$$\frac{\dot{Q}_{\mathrm{H,SI}}}{T_{3,\mathrm{SI}}} - \frac{\dot{Q}_{\mathrm{L,SI}}}{T_{1,\mathrm{SI}}} \le 0 \quad \text{or} \quad I_{SI} \frac{\dot{Q}_{\mathrm{H,SI}}}{T_{3,\mathrm{SI}}} = \frac{\dot{Q}_{\mathrm{L,SI}}}{T_{1,\mathrm{SI}}} \tag{19}$$

Here, I_{SI} is the internal irreversibility coefficient of SI engine. I_{SI} and $\dot{Q}_{\mathrm{L,SI}}$ can be obtained as follows by using eqs. (4) and (16)-(19):

$$I_{\mathrm{SI}} = \frac{\alpha_{\mathrm{SI}}}{k_{\mathrm{SI}} x_{\mathrm{SI}}} \tag{20}$$

$$\dot{Q}_{\mathrm{L,SI}} = \frac{\dot{m}_{\mathrm{SI}} k_{\mathrm{SI}} T_{1,\mathrm{SI}} (\alpha_{\mathrm{SI}} - x_{\mathrm{SI}})}{1 + k_{\mathrm{SI}}} \tag{21}$$

By using eqs. (1) and eqs. (16)-(21), $\dot{E}_{\mathrm{SI}}$ is obtained as:

$$\dot{E}_{\mathrm{SI}} = \frac{\dot{m}_{\mathrm{SI}} K_{\mathrm{SI}} (\alpha_{\mathrm{SI}} - x_{\mathrm{SI}})[k_{\mathrm{SI}} T_0 x_{\mathrm{SI}} + \alpha_{\mathrm{SI}} (k_{\mathrm{SI}} T_{1,\mathrm{SI}} x_{\mathrm{SI}} - T_{1,\mathrm{SI}} - T_{\mathrm{o}})]}{\alpha_{\mathrm{SI}} x_{\mathrm{SI}} (1 + k_{\mathrm{SI}})} \tag{22}$$

Here, in order to detect optimum compression rate, a derivative of the SI engine according to the compression rate of its ecological function is equalized to zero ($\partial \dot{E}_{\mathrm{SI}} / \partial x_{\mathrm{SI}} = 0$). The corresponding value is:

$$x_{\mathrm{op,SI}} = \sqrt{\frac{\alpha_{\mathrm{SI}}^2 (T_{1,\mathrm{SI}} + T_{\mathrm{o}})}{k_{\mathrm{SI}} (\alpha_{\mathrm{SI}} T_{1,\mathrm{SI}} + T_{\mathrm{o}})}} \tag{23}$$

– Irreversible CI engine

From the first law of thermodynamics:

$$\dot{Q}_{\mathrm{H,CI}} - \dot{Q}_{\mathrm{L,CI}} = \dot{W}_{\mathrm{net,CI}} \tag{24}$$

$$\dot{Q}_{\mathrm{H,CI}} = \frac{\dot{m}_{\mathrm{CI}} K_{\mathrm{CI}} k_{\mathrm{CI}}}{1 + k_{\mathrm{CI}}} (T_{3,\mathrm{CI}} - T_{2,\mathrm{CI}}), \quad \dot{Q}_{\mathrm{L,CI}} = \frac{\dot{m}_{\mathrm{CI}} K_{\mathrm{CI}}}{1 + k_{\mathrm{CI}}} (T_{4,\mathrm{CI}} - T_{1,\mathrm{CI}}) \tag{25}$$

For CI engine:

$$\varepsilon_{\mathrm{CI}} = \frac{V_{1,\mathrm{CI}}}{V_{2,\mathrm{CI}}}, \quad \frac{T_{2,\mathrm{CI}}}{T_{1,\mathrm{CI}}} = x_{\mathrm{CI}}, \quad \frac{T_{4,\mathrm{CI}}}{T_{3,\mathrm{CI}}} = \frac{w_{\mathrm{CI}}}{x_{\mathrm{CI}}}, \quad x_{\mathrm{CI}} = (\varepsilon_{CI})^{n_{CI}-1}, \quad \frac{T_{3,\mathrm{CI}}}{T_{1,\mathrm{CI}}} = \alpha_{\mathrm{CI}},$$
$$T_{2,\mathrm{CI}} = T_{1,\mathrm{CI}} x_{\mathrm{CI}}, \quad T_{4,\mathrm{CI}} = \frac{T_{3,\mathrm{CI}} w_{\mathrm{CI}}}{x_{\mathrm{CI}}}, \quad w_{\mathrm{CI}} = r^{n_{CI}-1}, \quad K_{\mathrm{CI}} = c_{\mathrm{p,2\text{-}3,CI}} + c_{\mathrm{v,4\text{-}1,CI}}, \quad k_{CI} = \frac{c_{\mathrm{p,2\text{-}3,CI}}}{c_{\mathrm{v,4\text{-}1,CI}}} \tag{26}$$

From the second law of thermodynamics:

$$\frac{\dot{Q}_{\mathrm{H,CI}}}{T_{3,\mathrm{CI}}} - \frac{\dot{Q}_{\mathrm{L,CI}}}{T_{1,\mathrm{CI}}} \le 0 \quad \text{or} \quad I_{\mathrm{CI}} \frac{\dot{Q}_{\mathrm{H,CI}}}{T_{3,\mathrm{CI}}} = \frac{\dot{Q}_{\mathrm{L,CI}}}{T_{1,\mathrm{CI}}} \tag{27}$$

Here, I_{CI} is the internal irreversibility coefficient of CI engine. I_{CI} and $\dot{Q}_{\mathrm{L,CI}}$ can be obtained by using eqs. (4) and (24)-(27), respectively:

$$I_{\mathrm{CI}} = \frac{\alpha_{\mathrm{CI}}(\alpha_{\mathrm{CI}} w_{\mathrm{CI}} - x_{\mathrm{CI}})}{k_{\mathrm{CI}} x_{\mathrm{CI}}(\alpha_{\mathrm{CI}} - x_{\mathrm{CI}})} \tag{28}$$

$$\dot{Q}_{\mathrm{L,CI}} = \frac{\dot{m}_{\mathrm{CI}} K_{\mathrm{CI}} T_{1,\mathrm{CI}}(\alpha_{\mathrm{CI}} w_{\mathrm{CI}} - x_{\mathrm{CI}})}{k_{\mathrm{CI}}(1+k_{\mathrm{CI}}) x_{\mathrm{CI}}} \tag{29}$$

By using eqs. (1) and (24)-(29), $\dot{E}_{\mathrm{CI}}$ is obtained as:

$$\dot{E}_{\mathrm{CI}} = \frac{K_{CI}\, m_{CI}\, \{\alpha_{\mathrm{CI}} x_{\mathrm{CI}}(T_{1,\mathrm{CI}} + T_o + k_{\mathrm{CI}} T_o - k_{\mathrm{CI}} T_{1,\mathrm{CI}}) - \alpha_{\mathrm{CI}}^2 [w_{\mathrm{CI}} T_{\mathrm{o}} + T_{1,\mathrm{CI}}][w_{\mathrm{CI}} - k_{\mathrm{CI}} T_{\mathrm{o}} x_{\mathrm{CI}})]\}}{\alpha_{\mathrm{CI}} x_{\mathrm{CI}}(1+k_{\mathrm{CI}})} \tag{30}$$

Here, in order to detect the optimum compression rate, a derivative of CI engine according to the compression rate of its ecological function is equalized to zero ($\partial \dot{E}_{\mathrm{CI}} / \partial x_{\mathrm{CI}} = 0$). The corresponding value is:

$$x_{\mathrm{op,CI}} = \sqrt{\frac{\alpha_{\mathrm{CI}}^2 (T_{1,\mathrm{CI}} + T_{\mathrm{o}}) w_{\mathrm{CI}}}{k_{\mathrm{CI}}(\alpha_{\mathrm{CI}} T_{1,\mathrm{CI}} + T_{\mathrm{o}})}} \tag{31}$$

Thermodynamic analysis of irreversible Brayton cycles

Since the Brayton cycle generates an ideal model for continuous combustion gas turbines, it has an important place in thermodynamic optimization studies. There are various studies examining parameters such as maximum power, maximum power density, and thermal efficiency of the Brayton cycle under finite time, finite area, and finite rate restrictions and which determine its performance limits [44-55]. Figure 2 shows the *T-s* diagram of the irreversible Brayton cycle.

From the first law of thermodynamics:

$$\dot{Q}_{\mathrm{H,B}} - \dot{Q}_{\mathrm{L,B}} = \dot{W}_{\mathrm{net,B}} \tag{32}$$

$$\dot{Q}_{\mathrm{H,B}} = \frac{\dot{m}_{\mathrm{B}} K_{\mathrm{B}} k_{\mathrm{B}}}{1+k_{\mathrm{B}}}(T_{3,\mathrm{B}} - T_{2,\mathrm{B}}), \quad \dot{Q}_{\mathrm{L,B}} = \frac{\dot{m}_{\mathrm{B}} K_{\mathrm{B}}}{1+k_{\mathrm{B}}}(T_{4,\mathrm{B}} - T_{1,\mathrm{SI}}) \tag{33}$$

For Brayton cycles:

$$v_{\mathrm{B}} = \frac{P_{2,\mathrm{B}}}{V_{1,\mathrm{B}}} = \frac{P_{3,\mathrm{B}}}{P_{4,\mathrm{B}}}, \quad \frac{T_{2,\mathrm{B}}}{T_{1,\mathrm{B}}} = \frac{T_{3,\mathrm{B}}}{T_{4,\mathrm{B}}} = x_{\mathrm{B}}, \quad x_{\mathrm{B}} = (v_{\mathrm{B}})^{\frac{(n_B - 1)}{n_B}}, \quad \frac{T_{3,\mathrm{B}}}{T_{1,\mathrm{B}}} = \alpha_{\mathrm{B}},$$
$$T_{2,\mathrm{B}} = T_{1,\mathrm{B}} x_{\mathrm{B}}, \quad T_{4,\mathrm{B}} = \frac{T_{3,\mathrm{B}}}{x_{\mathrm{B}}}, \quad K_{\mathrm{B}} = c_{\mathrm{p,2\text{-}3,B}} + c_{\mathrm{p2,4\text{-}1,B}}, \quad k_{\mathrm{B}} = \frac{c_{\mathrm{p,2\text{-}3,B}}}{c_{\mathrm{p,4\text{-}1,B}}} \tag{34}$$

From the second law of thermodynamics:

$$\frac{\dot{Q}_{\mathrm{H,B}}}{T_{3,\mathrm{B}}} - \frac{\dot{Q}_{\mathrm{L,B}}}{T_{1,\mathrm{B}}} \le 0 \quad \text{or} \quad I_B \frac{\dot{Q}_{\mathrm{H,B}}}{T_{3,\mathrm{CI}}} = \frac{\dot{Q}_{\mathrm{L,B}}}{T_{1,\mathrm{B}}} \tag{35}$$

Here, I_{B} is the internal irreversibility coefficient of the Brayton cycle. I_{B} and $\dot{Q}_{\mathrm{L,B}}$ can be obtained by using eqs. (4) and (32)-(35), respectively:

$$I_{B} = \frac{\alpha_{B}}{k_{B} x_{B}} \tag{36}$$

$$\dot{Q}_{L,SI} = \frac{\dot{m}_{B} K_{B} T_{B} (\alpha_{B} - x_{B})}{1 + k_{B}} \tag{37}$$

By using eqs. (1) and (32)-(37), E_B is obtained as:

$$\dot{E}_{B} = \frac{\dot{m}_{B} K_{B} (\alpha_{B} - x_{B})[k_{B} T_{o} x_{B} + \alpha_{B} (k_{B} x_{B} - 2)]}{\alpha_{B} x_{B} (1 + k_{B})} \tag{38}$$

Here, in order to detect optimum pressure rate, a derivative of the Brayton cycle according to the compression rate of its ecological function is equalized to zero ($\partial \dot{E}_B/\partial x_B = 0$). The corresponding value is:

$$x_{op,B} = \sqrt{\frac{2}{k_{B}(\alpha_{B} + 1)}\alpha_{SI}} \tag{39}$$

Thermodynamic analysis of irreversible Stirling and Ericsson cycles

Ericsson and Stirling engines have attracted the attention of several generations of engineers and physicists due to their theoretical potential to provide high conversion efficiency that approached those of the Carnot cycle. However, use of these engines did not prove to be successful due to relatively poor material technologies available at that time. As, the world community has become much more environmentally conscious, further attention in these engines has been again received because these engines are inherently clean and thermally more efficient. Moreover, as a result of advances in material technology, these engines are currently being considered for variety of applications due to their many advantages like low noise, less pollution and their flexibility as an external combustion engine to utilize a variety of energy sources or fuels. These engines are also under research and development for their use as heat pumps, replacing systems that are not ecological friendly and environmentally acceptable. Nowadays, the popularity of these engines is growing rapidly due to their many advantages like being more efficient, less pollution levels and their flexibility as external combustion engines to utilize different energy sources such as solar energy.

The central receiver and parabolic dish solar systems based Stirling/Ericsson heat engine is more efficient and suitable for both the terrestrial [33, 56] and non-terrestrial [57, 58] solar installations. In the literature, Erbay and Yavuz [59-62], investigated irreversible Stirling and Ericsson engines. In this study their model was adopted for the analyzing of the Stirling-Ericsson engines (fig. 3).

From the first law of thermodynamics:

$$\dot{Q}_{H,SE} - \dot{Q}_{L,SE} = \dot{W}_{net,SE} \tag{40}$$

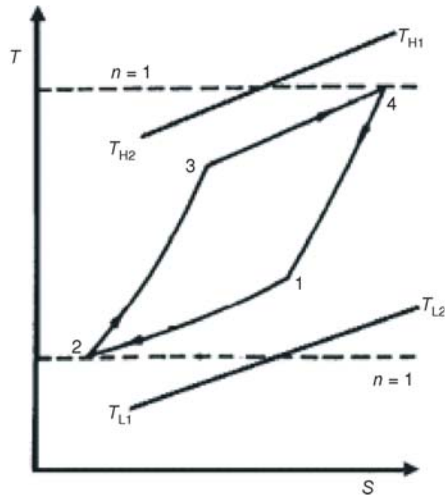

Figure 3. *T-s* diagram of irreversible Stirling-Ericsson cycle [32-35]

$$\dot{Q}_{\mathrm{H,SE}} = \dot{m}_{\mathrm{SE}} T_{4,\mathrm{SE}} R(1 - x_{\mathrm{SEr}}) \frac{\lambda - n}{(1-n)(\lambda - 1)} \quad \dot{Q}_{\mathrm{L}} = \dot{m}_{\mathrm{SE}} T_{2,\mathrm{SEr}} R\left(\frac{1}{x_{\mathrm{SEr}}} - 1\right) \frac{\lambda - n}{(1-n)(\lambda - 1)} \tag{41}$$

$$x_{\mathrm{SE}} = (\varepsilon_{\mathrm{S}})^{n_S - 1} = \left(\frac{1}{\nu_{\mathrm{Er}}}\right)^{\frac{n_{Er}-1}{n_{Er}}}, \quad \varepsilon_{\mathrm{S}} = \frac{V_{1,\mathrm{S}}}{V_{2,\mathrm{S}}}, \quad \nu_{Er} = \frac{P_{2,\mathrm{Er}}}{P_{1,\mathrm{Er}}},$$

$$\frac{T_{4,\mathrm{SEr}}}{T_{2,\mathrm{SEr}}} = \alpha_{\mathrm{SEr}}, \quad T_{1,\mathrm{SEr}} = \frac{T_{2,\mathrm{SEr}}}{x_{\mathrm{SEr}}}, \quad T_{3,\mathrm{SE}} = T_{4,\mathrm{SEr}} x_{\mathrm{SEr}}, \tag{42}$$

$$\theta_{\mathrm{SEr}} = \frac{\lambda - n}{(1-n)(\lambda - 1)}, \quad T_{\mathrm{H,SEr}} = \frac{T_{\mathrm{H1,SEr}} + T_{\mathrm{H2,SEr}}}{2}, \quad T_{\mathrm{L,SEr}} = \frac{T_{\mathrm{L1,SEr}} + T_{\mathrm{L2,SEr}}}{2}$$

as defined in [30-33].

From the second law of thermodynamics:

$$\frac{\dot{Q}_{\mathrm{H,SEr}}}{T_4} - \frac{\dot{Q}_{\mathrm{L,SEr}}}{T_2} \le 0 \quad \text{or} \quad I_{\mathrm{SEr}} \frac{\dot{Q}_{\mathrm{H,SEr}}}{T_4} = \frac{\dot{Q}_{\mathrm{L,SEr}}}{T_2} \tag{43}$$

Here, I_{SEr} is the internal irreversibility coefficient of Stirling and Ericsson cycles. I_{SEr} and $\dot{Q}_{\mathrm{L,SEr}}$ can be obtained by using eqs. (4) and (40)-(43), respectively:

$$I_{\mathrm{SEr}} = \frac{1}{x_{\mathrm{SEr}}} \tag{44}$$

$$\dot{Q}_{\mathrm{L,SEr}} = \alpha_{\mathrm{SEr}} \theta_{\mathrm{SEr}} \dot{m}_{\mathrm{SEr}} \mathrm{R} T_{2,\mathrm{SEr}} T_{\mathrm{L,SEr}} \left(\frac{1 - x_{\mathrm{SEr}}}{T_{\mathrm{H,SEr}} x_{\mathrm{SEr}}}\right) \tag{45}$$

By using eqs. (1) and (40)-(45), $\dot{E}_{\mathrm{SEr}}$ is obtained as:

$$\dot{E}_{\mathrm{SEr}} = \alpha_{\mathrm{SEr}} \theta_{\mathrm{SEr}} \dot{m}_{\mathrm{SEr}} \mathrm{R} T_{2,\mathrm{SEr}} \left[\frac{(T_{\mathrm{L,SEr}} + T_o - T_{\mathrm{H,SEr}} x_{\mathrm{SEr}} - T_o x_{\mathrm{SEr}})(1 - x_{\mathrm{SEr}})}{T_{\mathrm{H,SEr}} x_{\mathrm{SEr}}}\right] \tag{46}$$

Here, in order to detect an optimum pressure rate for the Ericsson cycle or an optimum compression rate for the Stirling cycle, a derivative of cycle of Stirling-Ericsson cycles according to the compression of their ecological function is equalized to zero ($\partial \dot{E}_{\mathrm{SEr}} / \partial x_{\mathrm{SEr}} = 0$). The corresponding value is:

$$x_{\mathrm{op,SEr}} = \sqrt{\frac{T_{\mathrm{L,SEr}} + T_{\mathrm{o}}}{T_{\mathrm{H,SEr}} + T_{\mathrm{o}}}} \tag{47}$$

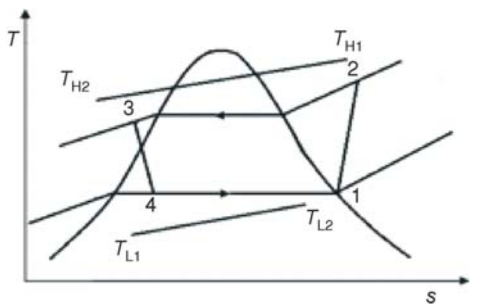

Figure 4. ***T-s*** **diagram of irreversible heat pump and refrigerators cycles**

Heat pump and refrigeration cycles

Heat pump and refrigeration cycles work with the same cycle requiring power from the environment. Cycles which are more efficient and harm the environment less must be designed by reducing the power obtained from the environment and entropy production. In this section, limits of these cycles were determined by using ecological criteria. Figure 4 shows the *T-s* diagram of irreversible heat pump and refrigerator cycles.

Thermodynamic analysis of general irreversible heat pump and irreversible refrigerator systems

From the first law of thermodynamics, power input (exergy input) for heat pump and refrigeration cycles is equal to the difference between the heat entering the system and the heat leaving the system. This expression can be obtained as follows:

$$\dot{Q}_{\mathrm{H,\,hr}} - \dot{Q}_{\mathrm{L,\,hr}} = \dot{W}_{\mathrm{net,\,hr}} \tag{48}$$

Here, $\dot{Q}_{\mathrm{H,R}}$ and $\dot{Q}_{\mathrm{L,R}}$ can be obtained, similar to the finite time thermodynamic used in references [40-41]:

$$\dot{Q}_{\mathrm{H,\,hr}} = \frac{Q_{\mathrm{H,\,hr}}}{t_{1,\,\mathrm{hr}}} = \phi_{\mathrm{C,\,hr}}(T_{\mathrm{H,\,hr}} - T_{\mathrm{E,\,hr}}) \quad \text{and} \quad \dot{Q}_{\mathrm{L,\,hr}} = \frac{Q_{\mathrm{L,\,hr}}}{t_{2,\,\mathrm{hr}}} = \phi_{\mathrm{E,\,hr}}(T_{\mathrm{C,\,hr}} - T_{\mathrm{L,\,hr}}) \tag{49}$$

$$t_{\mathrm{hr}} = t_{1,\mathrm{hr}} + t_{2,\mathrm{hr}} \tag{50}$$

Here, $T_{\mathrm{H,hr}}$ and $T_{\mathrm{L,hr}}$ are:

$$T_{\mathrm{H,\,hr}} = \frac{T_{\mathrm{H1,\,hr}} + T_{\mathrm{H2,\,hr}}}{2}, \quad T_{\mathrm{L,\,hr}} = \frac{T_{\mathrm{L1,\,hr}} + T_{\mathrm{L2,\,hr}}}{2} \tag{51}$$

From the second law of thermodynamics:

$$\frac{\dot{Q}_{\mathrm{H,\,hr}}}{T_{\mathrm{E,\,hr}}} - \frac{\dot{Q}_{\mathrm{L,\,hr}}}{\mathrm{T}_{\mathrm{C,\,hr}}} \le 0 \quad \text{or} \quad \frac{\dot{Q}_{\mathrm{H,\,hr}}}{T_{\mathrm{E,\,hr}}} = I_{\mathrm{H,\,hr}} \frac{\dot{Q}_{\mathrm{L,\,hr}}}{T_{\mathrm{C,\,hr}}} \tag{52}$$

Here, I_{hr} is the internal irreversible coefficient of heat pump and refrigeration cycles. I_{hr} and $\dot{Q}_{\mathrm{H,\,hr}}$ can be obtained by using eqs. (4) and (49)-(52), respectively:

$$I_{\mathrm{hr}} = \frac{\phi_{\mathrm{C,\,hr}} T_{\mathrm{E,\,hr}} (T_{\mathrm{C,\,hr}} - T_{\mathrm{H,\,hr}})}{\phi_{\mathrm{E,\,hr}} T_{\mathrm{C,\,hr}} (T_{\mathrm{L,\,hr}} - T_{\mathrm{E,\,hr}})} \tag{53}$$

$$\dot{Q}_{\mathrm{H,\,hr}} = \frac{1}{\left[\dfrac{1}{\phi_{\mathrm{C,\,hr}}}(T_{\mathrm{C,\,hr}} - T_{\mathrm{H,\,hr}})\right] + \left[\dfrac{\dfrac{T_{\mathrm{E,\,hr}}}{T_{\mathrm{C,\,hr}}}}{I_{\mathrm{hr}} \phi_{\mathrm{E,\,hr}} (T_{\mathrm{L,\,hr}} - T_{\mathrm{E,\,hr}})}\right]} \tag{54}$$

By using eqs. (1) and (50)-(55), $\dot{E}_{\mathrm{hr}}$ is obtained as:

$$\dot{E}_{\mathrm{hr}} = \frac{\phi_{\mathrm{E,\,hr}} T_{\mathrm{H,\,hr}} (T_{\mathrm{E,\,hr}} - T_{\mathrm{L,\,hr}})(T_{\mathrm{L,\,hr}} - T_{\mathrm{o}}) + \phi_{\mathrm{C,\,hr}} T_{\mathrm{L,\,hr}} (T_{\mathrm{C,\,hr}} - T_{\mathrm{H,\,hr}})(T_{\mathrm{H,\,hr}} - T_{\mathrm{o}})}{T_{\mathrm{H,\,hr}} (1 + T_{\mathrm{C,\,hr}}^2 - T_{\mathrm{C,\,hr}} T_{\mathrm{H,\,hr}} + T_{\mathrm{H,\,r}}^2) T_{\mathrm{L,\,hr}}} \tag{55}$$

Here, condenser temperature is selected as design parameter and in order to detect optimum condenser temperature, the derivative of the heat pump – refrigerator cycle according to condenser temperature is equalized to zero ($\partial \dot{E}_{\mathrm{hr}} / \partial T_{\mathrm{C,hr}} = 0$). The corresponding value is:

$$T_{\mathrm{op,C,\,hr}} = \frac{T_{\mathrm{L,\,hr}} T_{\mathrm{H,\,hr}} J_1 + \sqrt{\begin{matrix}\phi_{\mathrm{C,\,hr}}^2 T_{\mathrm{L,\,hr}}^2 (T_{\mathrm{H,\,hr}} - T_{\mathrm{o}})^2 + \\ + \phi_{\mathrm{E,\,hr}}^2 T_{\mathrm{H,\,hr}}^2 (T_{\mathrm{E,\,hr}} - T_{\mathrm{L,\,hr}})^2 (T_{\mathrm{L,\,hr}} - T_{\mathrm{o}})^2\end{matrix}} \, T_{\mathrm{H,\,hr}} T_{\mathrm{o}} J_2}{\phi_{\mathrm{C,\,hr}} T_{\mathrm{L,\,hr}} (T_{\mathrm{H,Hhr}} - T_{\mathrm{o}})} \tag{56}$$

$$J_1 = T_{\mathrm{L,hr}}\phi_{\mathrm{E,hr}} + T_{\mathrm{H,hr}}\phi_{\mathrm{C,hr}} - \phi_{\mathrm{E,hr}}T_{\mathrm{E,hr}} \tag{57}$$

$$J_2 = T_{\mathrm{E,hr}}\phi_{\mathrm{E,hr}} - T_{\mathrm{L,hr}}\phi_{\mathrm{C,hr}} - T_{\mathrm{L,hr}}\phi_{\mathrm{E,hr}} \tag{58}$$

Results and numerical examples

In this section, optimum points for all basic thermodynamic cycles were determined and they were discussed in detail for each cycle separately. Numerical examples were selected in a way that would be close to the values met in practice and by using references [3-29, 56-62].

Irreversible Rankine cycle

For numerical calculations, the working parameters of Rankine cycle are: $T_{\mathrm{H,R}} = 1000$ K, $T_{\mathrm{C,R}} = 350$ K, $T_{\mathrm{L,R}} = 300$ K, $\phi_{\mathrm{E,R}} = 15$ kW/K, and $\phi_{\mathrm{C,R}} = 0.25$ kW/K. Figure 5 shows the variation of power output ($\dot{W}$), thermal efficiency (η), ecological function ($\dot{E}$), exergy destruction (ExD), and the internal irreversibility parameter (I) of the Rankine cycle with regard to evaporator temperature. Evaporator temperature for the Rankine cycle was detected as 998.957 K. It is seen that with the increase of evaporator temperature, ecological function rises up to the optimum evaporator temperature, while power output and thermal efficiency decrease as ExD increase. The reason for the decrease in power and efficiency is the increase of irreversibility and exergy destruction. By examining eqs. (6) and (10), it can be clearly seen that internal irreversibility and entropy production (exergy destruction) will increase with the increase of evaporator temperature.

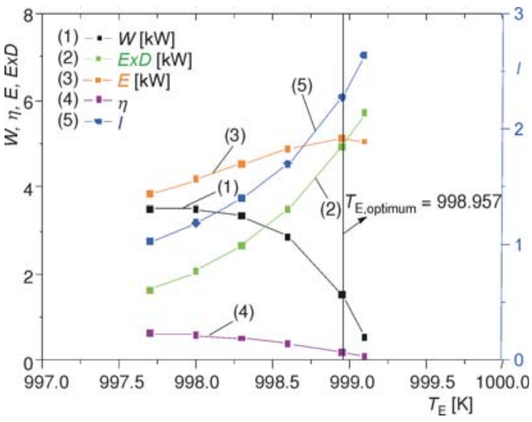

Figure 5. Effect off T_{E} on *E*, *W*, *I*, *ExD*, and η for irreversible Rankine cycle

Irreversible SI engines

For numerical calculations, the working parameters of the SI engine are: $T_{1,\mathrm{SI}} = 350$ K, $\alpha_{\mathrm{SI}} = 4$, $\dot{m}_{\mathrm{SI}} = 0.5$ kg/s, $c_{\mathrm{vS,I,2\text{-}3}} = 0.8$ kJ / kgK, $n_{\mathrm{SI}} = 1.37$. Figures 6 and 7 show the variation of power output ($\dot{W}$), thermal efficiency (η), ecological function ($\dot{E}$), exergy destruction (ExD), and the internal irreversibility parameter (I) of the SI engine with regard to compression rate. The optimum compression rate for the SI engine was detected as 12.25. It is seen that with the increase of the compression rate, ecological function rises up to the optimum compression rate, then starts to decrease and thermal efficiency increases, while power output, ExD, and the internal irreversibility parameter decrease. The reason for the decrease in power and irreversibility parameter can be

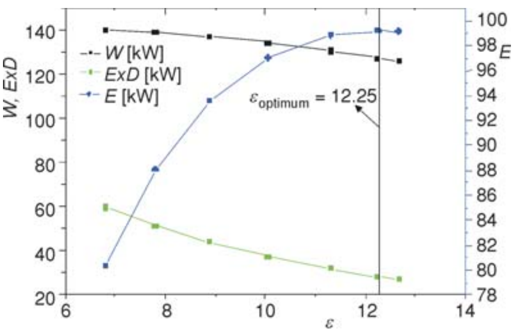

Figure 6. Effect of ε on the *E*, *W*, and *ExD* for irreversible SI engine

seen by looking at eqs. (20)-(22). The decrease of ExD and the direct irreversibility parameter are directly proportional since entropy production is the result of irreversibilities.

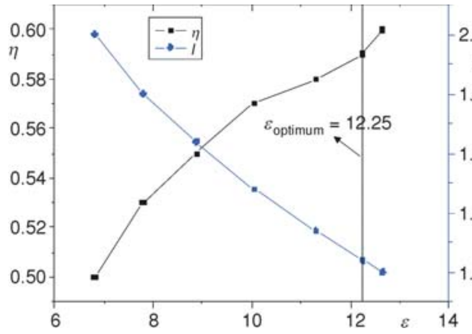

Figure 7. Effect of ε on the η and I for irreversible SI engine

Irreversible CI engines

For numerical calculations, the working parameters of the CI engine are: $T_{1,\mathrm{CI}} = 350$ K, $\alpha_{\mathrm{SI}} = 6$, $\dot{m}_{\mathrm{SI}} = 0.5$ kg/s, $k_{\mathrm{CI}} = 1.1$, $K_{\mathrm{CI}} = 1.4$ kJ/kgK, $w_{\mathrm{CI}} = 1.23$, and $n_{\mathrm{CI}} = 1.37$. Figures 8 and 9 show the variation of power output ($\dot{W}$), thermal efficiency (η), ecological function ($\dot{E}$), energy destruction (ExD) and the internal irreversibility parameter (I) of the CI engine with regard to compression rate. The optimum compression rate for the CI engine was detected as 18.52. It is seen that with the increase of compression rate, ecological function rises up to the optimum compression rate, then starts to decrease and thermal efficiency increases, while power output, exergy destruction and the internal irreversibility parameter decrease. The reason for the decrease in power and irreversibility parameter can be seen by looking at eqs. (28)-(30). A decrease of exergy destruction and the direct irreversibility parameter are directly proportional since entropy production is the result of irreversibilities.

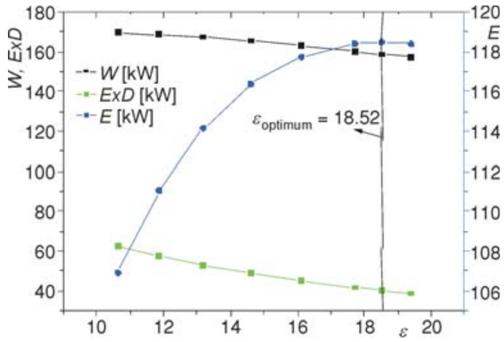

Figure 8. Effect of ε on the E, W, and ExD for irreversible CI engine

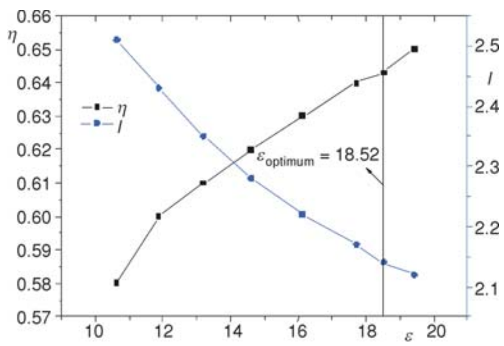

Figure 9. Effect of ε on the η and I for irreversible CI engine

Irreversible Brayton cycles

For numerical calculations, the working parameters of the Brayton cycle are: $T_{1,\mathrm{B}} = 300$ K, $\alpha_{\mathrm{B}} = 3.5$, $\dot{m}_{\mathrm{B}} = 0.5$ kg/s, $c_{\mathrm{p,B,2\text{-}3}} = 1.1$ kJ/kgK, and $n_{\mathrm{B}} = 1.37$. Figures 10 and 11 show the variation of power output ($\dot{W}$), thermal efficiency (η), ecological function ($\dot{E}$), exergy destruction (ExD), and the internal irreversibility parameter (I) of the Brayton cycle with regard to pressure rate. The optimum pressure rate of the Brayton cycle was detected as 25.12. It is seen that with the increase of pressure rate, ecological function rises up to the optimum pressure rate, then starts to decrease and thermal efficiency increases, while power output, exergy destruction and the internal irreversibility parameter decrease. The reason for the decrease in power and irreversibility parameter can be seen by looking at eqs. (36)-(38). A decrease of exergy destruction and direct irreversibility parameter are directly proportional since entropy production is the result of irreversibilities.

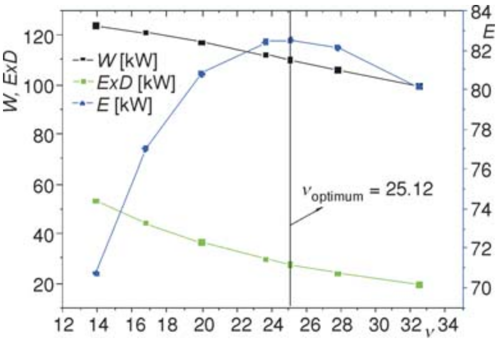

Figure 10. Effect of ν on E, W, and ExD for irreversible Brayton cycle

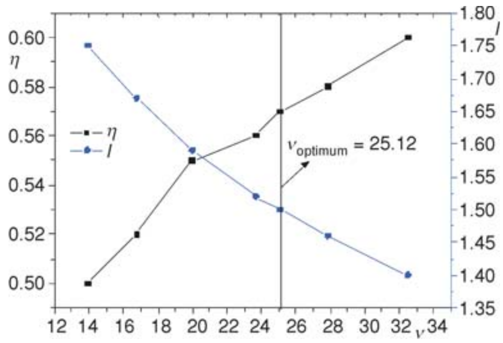

Figure 11. Effect of ν on η and I for irreversible Brayton cycle

Irreversible Stirling-Ericsson engine

For numerical calculations, the working parameters of the Stirling-Ericsson cycle are: $T_{1,\mathrm{SEr}} = 350$ K, $\alpha_{\mathrm{SEr}} = 3$, $\dot{m}_s = 0.5$ kg/s, $\theta = 5.34$, $R = 0.287$ kJ/kgK, $T_{\mathrm{H,SEr}} = 1000$ K, and $T_{\mathrm{L,SEr}} = 300$ K. Figures 12 and 13 show the variation of power output ($\dot{W}$), thermal efficiency (η), ecological function ($\dot{E}$), exergy destruction (ExD) and the internal irreversibility parameter (I) of Stirling-Ericsson cycle with regard to compression rate. The optimum compression and pressure rates of the Stirling-Ericsson cycle were detected as 2.49. It is seen that with the increase of pressure and compression rates, ecological function rises up to the optimum compression rate, then starts to decrease and thermal efficiency increases, power output, and exergy destruction increase, while in contrast, thermal efficiency and the internal irreversibility parameter decrease. The reason for the increase in power and decrease in irreversibility parameter can be seen by looking at eqs. (44)-(46).

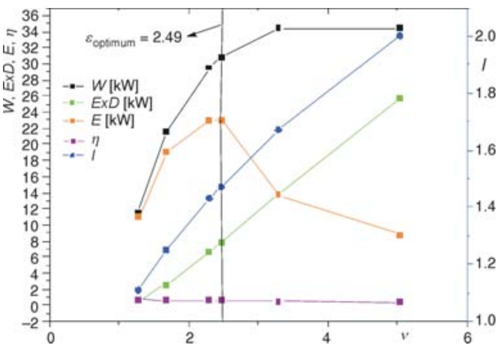

Figure 12. Effect of ε on E, η, I, W, and ExD for irreversible Sitirling engine

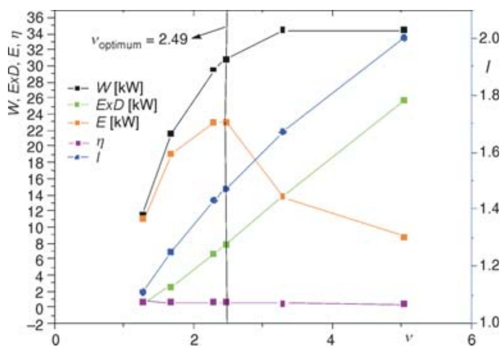

Figure 13. Effect of ν on E, η, I, W, and ExD for irreversible Ericsson engine

Irreversible heat pump and refrigeration cycles

For numerical calculations, the working parameters of the heat pump and refrigeration cycle are: $T_{\mathrm{H,hr}} = 300$ K, $T_{\mathrm{E,hr}} = 250$ K, $T_{\mathrm{L,hr}} = 260$ K, $\phi_{\mathrm{E,hr}} = 0.1$ kW/K, and $\phi_{\mathrm{C,hr}} = 8$ kW/K. Figures 14 and 15 show the variation of power output ($\dot{W}$), thermal efficiency (η), ecological func-

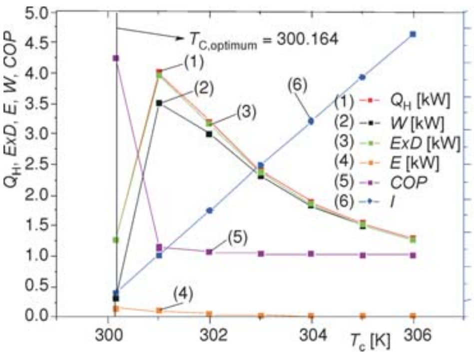

Figure 14. Effect of T_C on Q_H (heating load), *W*, *I*, *E*, *ExD*, and *COP* for irreversible heat pump cycle

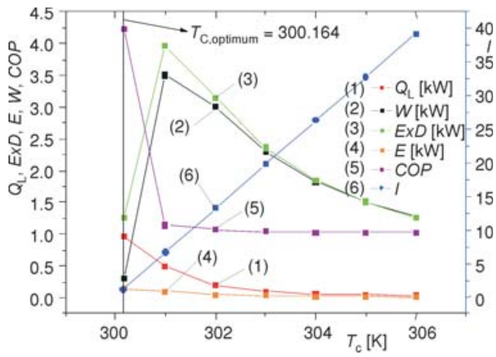

Figure 15. Effect of T_C on Q_L (cooling load), *W*, *I*, *E*, *ExD*, and *COP* for irreversible refrigerator

tion ($\dot{E}$), exergy destruction (*ExD*) and the internal irreversibility parameter (*I*) of heat pump and refrigeration cycles with regard to condenser temperature. The optimum condenser temperature for the heat pump and refrigeration cycle was detected as 300.164 K. With the increase in condenser temperature *ExD*, heat load in the heat pump cycle and refrigeration load in the refrigeration cycle dramatically increased until reaching a value of 301 K and started to decrease after that value, while the *COP* value dramatically decreased until reaching 301 K and decreasing speed dropped after this value. Ecological function on the other hand decreases regularly after the optimum condenser temperature.

Conclusions

In this study, ecological optimization was made for all basic thermodynamic irreversible cycles and the results were discussed. All cycles were evaluated by taking exergy destruction, thermal efficiency, *COP*, power output or power input, and internal irreversibility parameters into account. By regarding the results presented within the scope of this study, the actual thermodynamic cycles to be used were aimed to be designed with the least exergy destruction and as environmental friendly as possible.

Nomenclature

c_p – specific heat, [kJkg^{-1}K]
COP – coefficient of performance
E – ecological funcion
ExD – exergy destruction [kW]
I – internal irreversibility parameter
K – sum of specific heats, [kJkg^{-1}K]
k – ratio of specific heats
$\dot{m}$ – mass flow, [kgs^{-1}]
n – polytrophic coefficient
R – ideal gas constant, [kJkg^{-1}K]
r – cut-off ratio
T – temperature, [kW]
t – time, [s]
Q – heat, [kJ]
$\dot{Q}$ – heat, [kW]
W – work, [kW]
WF – working fluid
w – dimensionless compression ratio parameter
x – dimensionless compression ratio or dimensionless pressure ratio parameter

Greek symbols

α – ratio of the highest and the lowest temperature of the cycle
ε – compression ratio
η – efficiency
θ – a coefficient for Ericsson and Stirling engines
λ – isentropic coefficient
ν – pressure ratio
σ – entropy generation, [kWK^{-1}]
ϕ – heat conductance, [kWK^{-1}]

Subscripts

B – Brayton
C – condenser

CI – compression injection engine
E – evaporator
H – high
hr – heat pump or refrigeration
L – low
R – Rankine
r – refrigerator
S – Stirling
SEr – Stirling or Ericsson engine
SI – spark injection engine
o – environmental conditions
op – optimum

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