OPTIMIZATION OF AN IRREVERSIBLE OTTO AND DIESEL CYCLES BASED ON ECOLOGICAL FUNCTION

by

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Original scientific paper
https://doi.org/10.2298/TSCI170613190M

In this work, a mathematical model is presented for the irreversible Otto and Diesel cycles using finite time thermodynamics. The cycle is analyzed between two reservoirs with infinite thermal capacitances, where the processes of heat exchange occur in the heat exchangers between the working fluid and the thermal reservoirs at constant temperatures. The irreversibilities follow from the heat exchange processes occurring in finite time, the leakage of heat from the hot source to the cold source and the non-isentropic compression and expansion processes. The ecological optimization criterion represents the best compromise between power output of an engine and the environment that surrounds it. The results are presented through the power curves and ecological criteria, efficiency and ecological criteria and entropy generation rate and ecological criteria. Analysis is conducted to behavior of power, thermal efficiency and entropy generation rate ecologically optimized through which are evaluated the influences of some parameters on their behavior. Finally, maximum and ecological criteria are compared graphically. The analysis shows that the ecological optimizations present the best compromise between power and environment. The results can be used as an important criterion in developing projects of internal combustion engines.

Key words: *Otto cycle, Diesel cycle, finite time thermodynamics, optimization, ecological function*

Introduction

The internal combustion engines are present in the daily modern life. The environmental laws increasingly stringent and the necessity of responsible use of fuels causes the current researches are focused primarily in power quality, where an ecological compromise between power output and a loss in power consumption fuels is a necessity. One of the tools to improve these engines is finite-time thermodynamics (FTT) [1-3], thus, the FTT was applied to Otto and Diesel engines [4, 5].

The ecological optimizations had its beginning with Angulo-Brown [6], until then many authors often sought to maximize the power [7-14] or thermal efficiency [9-11] or minimizing the entropy generation rate [15, 16]. Some of these optimizations have roots in endorreversible model as quoted by Chen [9], where it is considered that the engine operates

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between two heat reservoirs at different temperatures. Thus, heat transfers are performed by heat exchangers between the reservoirs and the engine, and irreversibility are due to the thermal resistance of the heat exchangers. Using this model, Curzon and Alborn [17] has shown that for many endorreversible cycles under conditions of maximum net power, it's thermal efficiency is equal to $\eta = 1 - (T_L / T_H)^{1/2}$, which T_L is the low temperature reservoir and T_H is the high temperature reservoir, and the author also showed that in real installations this expression is more appropriate than the classical expression of Carnot efficiency. However, there was still a need to understand the thermodynamic interactions between the heat engine and the environment, *i. e.*, it was necessary to create a function that relates power and environment, so that one could study a good ratio between these two factors.

Thus, the concept of an ecological function [6] came up, $\dot{E} = \dot{W} - T_L \dot{S}_g$ which is named for presenting a ratio between net power output and the rate of exergy destruction (availability loss). Yan [18], did a review of the work of Angulo-Brown and found it would be more appropriate to use the temperature T_0 (temperature of the engine is inserted) instead of the low temperature reservoir T_L. Therefore, the ecological function $\dot{E} = \dot{W} - T_0 \dot{S}_g$ is most comprehensive and generalizes the one proposed by Angulo-Brown function. The ecological function can be interpreted as a compromise between power output and *loss of power*, that loss is accounted by the entropy generation rate of the engine itself and the system that surrounds it [19].

In this work will be analyzed irreversible cycles Otto and Diesel where the irreversibility of processes originates in the processes of heat exchange occurring in finite time, non-isentropic compression and expansion processes and the heat leakage between the reservoirs, like the work of [20-24]. For a better understanding of the ecological function, comparisons are made with the maximum operating conditions, showing that the operation of Otto and Diesel engines under ecological conditions present a good compromise between power and availability loss.

Mathematical formulation

Irreversible Otto and Diesel cycle modeling

Models presented in fig. 1(a) and 1(b) are represented in a temperature-entropy diagram for the irreversible Otto and Diesel cycle, respectively, where the processes of heat exchange occur in heat exchangers between the working fluid and heat reservoirs at constant temperatures.

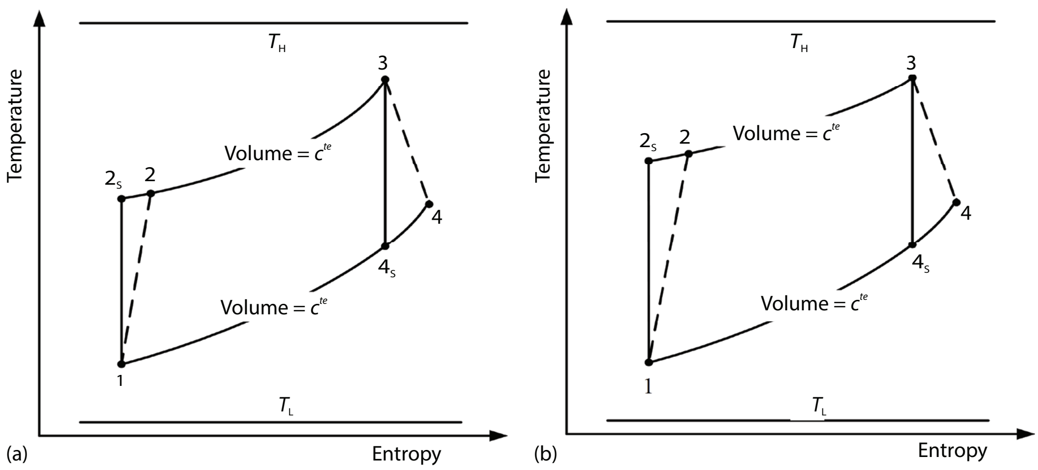

Figure 1. Temperature-entropy diagrams for Otto (a) and Diesel (b) irreversible cycles

In addition, heat process, the working fluid exchanges heat with the temperature reservoir that remains constant temperature, T_H. Similarly, in the heat rejection process, the working fluid exchanges heat with the low temperature reservoir that remains constant temperature, T_F.

The rate of heat transfer that occurs in heat exchangers can be written both in accordance with the variation of the thermal energy of the working fluid and by the heat exchangers analysis, considered countercurrent heat exchangers:

$$\dot{Q}_{HC} = U_H A_H \frac{(T_H - T_3) - (T_H - T_{2S})}{\ln\left(\dfrac{T_H - T_3}{T_H - T_{2S}}\right)} = \dot{C}_W \varepsilon_H (T_H - T_{2S}) = \dot{C}_W (T_3 - T_{2S}) \tag{1}$$

$$\dot{Q}_{LC} = U_L A_L \frac{(T_{4S} - T_L) - (T_1 - T_L)}{\ln\left(\dfrac{T_{4S} - T_L}{T_1 - T_L}\right)} = \dot{C}_W \varepsilon_F (T_{4S} - T_L) = \dot{C}_W (T_{4S} - T_1) \tag{2}$$

where $U_H A_H$ and $U_L A_L$ is the product of the overall heat transfer coefficient and heat transfer area, and effectivities ε_H and ε_L are given by the definition of the number of the transfer units NTU_H and NTU_L for countercurrent heat exchangers:

$$\varepsilon_H = 1 - \exp(-NTU_H) \tag{3}$$

$$\varepsilon_L = 1 - \exp(-NTU_L) \tag{4}$$

From equations of heat transfer rates of the cycle, eqs. (1) and (2), it obtains expressions for the temperatures T_3 and T_1:

$$T_3 = \varepsilon_H (T_O - T_{2S}) + T_{2S} \tag{5}$$

$$T_1 = -\varepsilon_L (T_{4S} - T_L) + T_{4S} \tag{6}$$

Two sources of irreversibility are used in this work, which are arising from the finite heat transfer that occurs between the heat reservoirs and the working fluid. In a study realized by Bejan [25], analyzing power plants, it was proposed a third source of irreversibilities occurring in the form of a rate of heat leakage, $\dot{Q}_I$, that crosses the plant directly from the heat exchanger of high temperature for heat exchanger of low temperature without any power output. Figure 2 shows that only part of the heat that flows from the high temperature reservoir is transferred to the cycle, which is denominate, $\dot{Q}_H$. Similarly, the heat rejection from the cycle to the low temperature reservoir is denominate $\dot{Q}_L$ and the heat leakage is given by eq. (7).

$$\dot{Q}_I = \dot{C}_I (T_H - T_L) \tag{7}$$

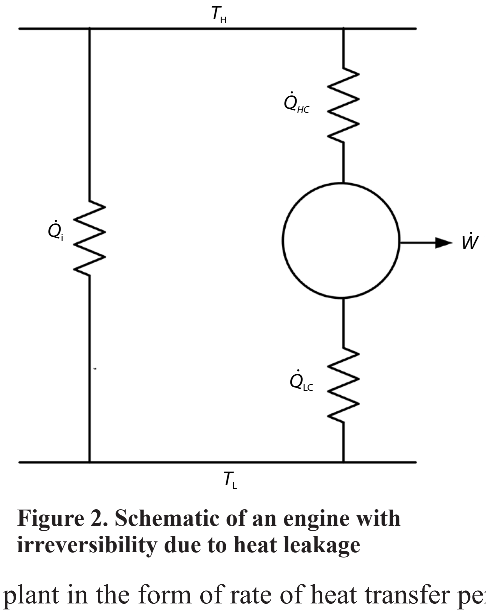

Figure 2. Schematic of an engine with irreversibility due to heat leakage

where $\dot{C}_I$ is the rate of internal conductance of the plant in the form of rate of heat transfer per unit temperature.

The rate of total heat transfer supplied by the reservoirs of high and low temperature can then be calculated by eqs. (8) and (9), respectively:

$$\dot{Q}_{\mathrm{HT}} = \dot{Q}_{\mathrm{HC}} + \dot{Q}_{\mathrm{I}} \tag{8}$$

$$\dot{Q}_{\mathrm{LT}} = \dot{Q}_{\mathrm{LC}} + \dot{Q}_{\mathrm{I}} \tag{9}$$

Applying the First law of thermodynamics for closed systems for both irreversible cycles, 1-2-3-4-1 of figs. 1(a) and 1(b), it is verified that net power output is equal to the difference between the rates of total heat transfer that entering and leaving the cycles, as shown in the following expression:

$$\dot{W} = \dot{Q}_{\mathrm{HT}} - \dot{Q}_{\mathrm{LT}} \tag{10}$$

Substituting eqs. (8) and (9) in eq. (10), the net power can be rewritten:

$$\dot{W} = \dot{Q}_{\mathrm{HC}} - \dot{Q}_{\mathrm{LC}} \tag{11}$$

Substituting eqs. (1) and (2) in eq. (11) and making explicit T_4, it can obtain the following expression:

$$T_4 = \frac{-\dot{W} + \dot{C}_{\mathrm{W}} \varepsilon_{\mathrm{H}} (T_{\mathrm{H}} - T_2)}{(\dot{C}_{\mathrm{W}} \varepsilon_{\mathrm{L}}) + T_{\mathrm{L}}} \tag{12}$$

The values of thermal efficiency, η, are obtained through the ratio between the net power and the rate of total heat transfer of thermal reservoirs, providing the following eq. (13):

$$\eta = \frac{\dot{W}}{\dot{Q}_{\mathrm{HC}} + \dot{Q}_{\mathrm{I}}} \tag{13}$$

Applying the Second law of thermodynamics and in the Otto and Diesel reversible cycles, it obtains eq. (14):

$$\frac{\dot{Q}_{\mathrm{LC}}}{T_{\mathrm{L,Av}}} - \frac{\dot{Q}_{\mathrm{HC}}}{T_{\mathrm{H,Av}}} = 0 \tag{14}$$

where

$$T_{\mathrm{H,Av}} = \frac{T_{2\mathrm{S}} + T_3}{2} \tag{15}$$

$$T_{\mathrm{L,Av}} = \frac{T_{4\mathrm{S}} + T_1}{2} \tag{16}$$

Substituting eqs. (15) and (16) in eq. (14) it obtains:

$$\frac{\dot{Q}_{\mathrm{LC}}}{\dfrac{T_{4\mathrm{S}} + T_1}{2}} - \frac{\dot{Q}_{\mathrm{HC}}}{\dfrac{T_{2\mathrm{S}} + T_3}{2}} = 0 \tag{17}$$

Knowing that the thermodynamic Otto and Diesel cycles are irreversible, eq. (14), which defines a reversible process can be rewritten in the form of an inequality, then setting an irreversible process:

$$\frac{2\dot{Q}_{\mathrm{LC}}}{T_4 + T_1} - \frac{2\dot{Q}_{\mathrm{HC}}}{T_2 + T_3} > 0 \tag{18}$$

Using the irreversibilities parameter, R, proposed by Wu *et al*. [2] it can rewrite eq. (18) as equality:

$$\frac{2\dot{Q}_{\mathrm{LC}}}{T_4+T_1}-R\frac{2\dot{Q}_{\mathrm{HC}}}{T_2+T_3}=0 \tag{19}$$

Otto cycle

The Otto cycle can be characterized by the processes of addition and heat rejection occurring at constant volume. Using a modeling of cold air-standard, where the working fluid is air which behaves as an ideal gas and considering that the rate of specific heats of the working fluid are constant, the heat capacity rate of the fluid is given by the rate of thermal capacity of the fluid at constant volume. Then, eqs. (1) and (2) can be rewritten:

$$\dot{Q}_{\mathrm{HC}}=\dot{C}_v\varepsilon_{\mathrm{H}}\left(T_{\mathrm{H}}-T_2\right)=\dot{C}_v\left(T_3-T_2\right) \tag{20}$$

$$\dot{Q}_{\mathrm{LC}}=\dot{C}_v\varepsilon_{\mathrm{L}}\left(T_4-T_{\mathrm{L}}\right)=\dot{C}_v\left(T_4-T_1\right) \tag{21}$$

Substituting eqs. (20) and (21), the rate of heat transfer related to effectivities of each heat exchanger and eqs. (5), (6), and (12) with respect to temperature T_3, T_1, and $T_{4\mathrm{m}}$ in the eq. (19), it obtains an algebraic equation for the T_2 temperature:

$$A_{1,\mathrm{O}}T_2^2+A_{2,\mathrm{O}}T_2+A_{3,\mathrm{O}}=0 \tag{22}$$

where the coefficients $A_{1,\mathrm{O}}$, $A_{2,\mathrm{O}}$, and $A_{3,\mathrm{O}}$ are expressed:

$$A_{1,\mathrm{O}}=a_{1,\mathrm{O}} \tag{23}$$

$$A_{1,\mathrm{O}}=a_{2,\mathrm{O}}\dot{W}+a_{3,\mathrm{O}} \tag{24}$$

$$A_{3,\mathrm{O}}=a_{4,\mathrm{O}}\dot{W}+a_{5,\mathrm{O}} \tag{25}$$

where the coefficients $a_{1,\mathrm{O}}$, $a_{2,\mathrm{O}}$, $a_{3,\mathrm{O}}$, $a_{4,\mathrm{O}}$ and $a_{5,\mathrm{O}}$ are expressed:

$$a_{1,\mathrm{O}}=\dot{C}_v\varepsilon_{\mathrm{H}}+\frac{1}{2}R\dot{C}_v\varepsilon_{\mathrm{H}}^2-\frac{R\dot{C}_v\varepsilon_{\mathrm{H}}^2}{\varepsilon_{\mathrm{L}}}+\frac{1}{2}\dot{C}_v\varepsilon_{\mathrm{H}}^2 \tag{26}$$

$$a_{2,\mathrm{O}}=-1+\frac{1}{2}R\varepsilon_{\mathrm{H}}+\frac{1}{2}\varepsilon_{\mathrm{H}}-\frac{R\varepsilon_{\mathrm{H}}}{\varepsilon_{\mathrm{L}}} \tag{27}$$

$$a_{3,\mathrm{O}}=R\dot{C}_v\varepsilon_{\mathrm{H}}T_{\mathrm{L}}+\dot{C}_v\varepsilon_{\mathrm{H}}T_{\mathrm{H}}-R\dot{C}_v\varepsilon_{\mathrm{H}}^2T_{\mathrm{H}}-\dot{C}_v\varepsilon_{\mathrm{H}}^2T_{\mathrm{H}}+\frac{2\dot{C}_v\varepsilon_{\mathrm{H}}^2T_{\mathrm{H}}}{\varepsilon_{\mathrm{L}}} \tag{28}$$

$$a_{4,\mathrm{O}}=-\frac{1}{2}R\varepsilon_{\mathrm{H}}T_{\mathrm{H}}-\frac{1}{2}\varepsilon_{\mathrm{H}}T_{\mathrm{H}}+\frac{R\varepsilon_{\mathrm{H}}T_{\mathrm{H}}}{\varepsilon_{\mathrm{L}}} \tag{29}$$

$$a_{5,\mathrm{O}}=\frac{1}{2}R\dot{C}_v\varepsilon_{\mathrm{H}}^2T_{\mathrm{H}}^2-R\dot{C}_v\varepsilon_{\mathrm{H}}T_{\mathrm{H}}T_{\mathrm{L}}+\frac{1}{2}\dot{C}_v\varepsilon_{\mathrm{H}}^2T_{\mathrm{H}}-\frac{R\dot{C}_v\varepsilon_{\mathrm{H}}^2T_{\mathrm{H}}^2}{\varepsilon_{\mathrm{L}}} \tag{30}$$

Solving eq. (22), it can obtain an equation for, T_2, temperature as a function of net power output of the Otto cycle:

$$T_{2,\mathrm{O}}=\frac{-A_{2,\mathrm{O}}\pm\sqrt{A_{2,\mathrm{O}}^2-4A_{1,\mathrm{O}}A_{3,\mathrm{O}}}}{2A_{1,\mathrm{O}}} \tag{31}$$

Diesel cycle

Different from the Otto cycle, the Diesel cycle can be characterized by the addition of heat process occurring at constant pressure, while the process of heat rejection occurs at constant volume, as in the Otto cycle. The same model used for the Otto cycle can be used for the Diesel cycle, whereas the heat capacity rate of the fluid in the process of heat addition is the rate of heat capacity at constant pressure fluid. Therefore, eq. (1) can be rewritten:

$$\dot{Q}_{\text{HC}} = \dot{C}_p \varepsilon_{\text{H}} \left(T_{\text{H}} - T_2\right) = \dot{C}_p \left(T_3 - T_2\right) \tag{32}$$

Realizing the same substitution process performed for the Otto cycle, eqs. (32) and (21), the heat transfer rates related to the effectivities of each heat exchanger and eqs. (5), (6), and (12), relative to the temperatures T_3, T_1, and T_4 in eq. (19), it obtains an algebraic equation for T_2 temperature:

$$A_{1,\text{D}} T_2^2 + A_{2,\text{D}} T_2 + A_{3,\text{D}} = 0 \tag{33}$$

where the coefficients $A_{1,\text{D}}$, $A_{2,\text{D}}$, and $A_{3,\text{D}}$ are expressed:

$$A_{1,\text{D}} = a_{1,\text{D}} \tag{34}$$

$$A_{2,\text{D}} = a_{2,\text{D}} \dot{W} + a_{3,\text{D}} \tag{35}$$

$$A_{3,\text{D}} = a_{4,\text{D}} \dot{W} + a_{5,\text{D}} \tag{36}$$

where the coefficients $a_{1,\text{D}}$, $a_{2,\text{D}}$, $a_{3,\text{D}}$, $a_{4,D}$, and $a_{5,\text{D}}$ are expressed:

$$a_{1,\text{D}} = \dot{C}_p \varepsilon_{\text{H}} + \frac{1}{2} Rk \dot{C}_p \varepsilon_{\text{H}}^2 - \frac{Rk \dot{C}_p \varepsilon_{\text{H}}^2}{\varepsilon_{\text{L}}} + \frac{1}{2} \dot{C}_p \varepsilon_{\text{H}}^2 \tag{37}$$

$$a_{2,\text{D}} = -1 + \frac{1}{2} Rk \varepsilon_{\text{H}} + \frac{1}{2} \varepsilon_{\text{H}} - \frac{Rk \varepsilon_{\text{H}}}{\varepsilon_{\text{L}}} \tag{38}$$

$$a_{3,\text{D}} = R \dot{C}_P \varepsilon_{\text{H}} T_{\text{L}} + \dot{C}_P \varepsilon_{\text{H}} T_{\text{H}} - Rk \dot{C}_P \varepsilon_{\text{H}}^2 T_{\text{H}} - \dot{C}_P \varepsilon_{\text{H}}^2 T_{\text{H}} + \frac{2Rk \dot{C}_P \varepsilon_{\text{H}}^2 T_{\text{H}}}{\varepsilon_{\text{L}}} \tag{39}$$

$$a_{4,\text{D}} = -\frac{1}{2} Rk \varepsilon_{\text{H}} T_{\text{H}} - \frac{1}{2} \varepsilon_{\text{H}} T_{\text{H}} + \frac{Rk \varepsilon_{\text{H}} T_{\text{H}}}{\varepsilon_{\text{L}}} \tag{40}$$

$$a_{5,D} = \frac{1}{2} Rk \dot{C}_p \varepsilon_H^2 T_H^2 - R \dot{C}_p \varepsilon_H T_H T_L + \frac{1}{2} \dot{C}_p \varepsilon_H^2 T_H - \frac{Rk \dot{C}_p \varepsilon_H^2 T_H^2}{\varepsilon_L} \tag{41}$$

Solving eq. (33), it can obtain an equation for temperature, T_2, as a function of net power output of the Diesel cycle:

$$T_{2,\text{D}} = \frac{-A_{2,\text{D}} \pm \sqrt{A_{2,\text{D}}^2 - 4A_{1,\text{D}} A_{3,\text{D}}}}{2A_{1,\text{D}}} \tag{42}$$

Ecological function

Suggested by Angulo-Brown [1, 6] and later modified by Yan [18], the ecological function provides high power ratings with low entropy generation rates, so this is known for the good ratio between net power and availability loss. This function is given by the difference between the net power and availability loss as shown by the equation below:

$$\dot{E} = \dot{W} - T_0 \dot{S}_g \tag{43}$$

Using this assumption and using the eqs. (1)-(9) and eq. (19), the following equation is obtained:

$$\dot{E} = C_1\dot{W} + C_2T_2 + C_3 \tag{44}$$

where the coefficients C_1, C_2, and C_3 are expressed:

$$C_1 = 1 + \frac{T_0}{T_\mathrm{L}} \tag{45}$$

$$C_2 = T_0\dot{C}_\mathrm{W}\varepsilon_\mathrm{H}\left(\frac{1}{T_\mathrm{L}} - \frac{1}{T_\mathrm{H}}\right) \tag{46}$$

$$C_3 = T_0\dot{C}_\mathrm{W}\varepsilon_\mathrm{H}\left(\frac{1}{T_\mathrm{L}} - \frac{1}{T_\mathrm{H}}\right) - T_0\dot{C}_I\left(\sqrt{\frac{T_\mathrm{L}}{T_\mathrm{H}}} - \sqrt{\frac{T_\mathrm{H}}{T_\mathrm{L}}}\right)^2 \tag{47}$$

Results and discussion

The parameters for construction of curves presented in this work are: R = 1.2, $NTU_\mathrm{H} = NTU_\mathrm{L}$ = 1, k = 1.4, $T_\mathrm{H}/T_\mathrm{L}$ = 4, $\dot{C}/\dot{C}_v$ = 0.2, and $T_0 = T_\mathrm{L}$. Figure 3 shows the behavior of the dimensionless ecological function $\dot{E}/\dot{C}T_\mathrm{H}$ as a function of dimensionless power $\dot{W}/\dot{C}T_\mathrm{H}$ for Otto and Diesel cycles for different values of the temperature ratio $T_\mathrm{H}/T_\mathrm{L}$. The graph has a wedge shape, where it can observe the values of maximum net power and ecological power for the two cycles. The region defined by these two points is an optimum region of operation, which there would be the highest values of power and ecological function. It is noted that the curve has negative values, due to the high entropy generation rate for low values of power, and as the power increases, ecological power values increase to the maximum values for both cycles. The increase of the temperature ratio $T_\mathrm{H}/T_\mathrm{L}$ causes the increase in both maximum power as the ecological power. It is evident that the Diesel cycle has a higher net power than the Otto cycle besides greater ecological power.

The curves in fig. 4 show a loop where it can be observed the points of maximum thermal efficiency and ecological power for both cycles. There are points of maximum thermal efficiency and maximum ecological function for the Otto and Diesel cycles, where the Diesel

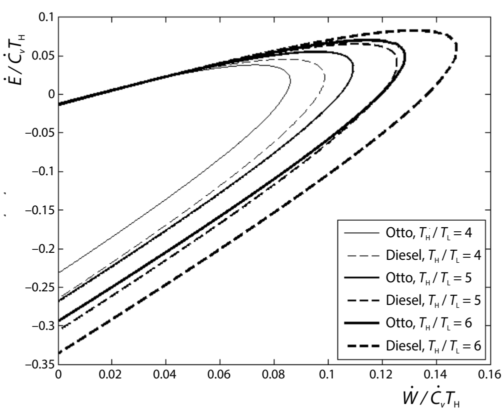

Figure 3. Dimensionless ecological function $\dot{E}/\dot{C}_v T_\mathrm{H}$ as a function of dimensionless power $\dot{W}/\dot{C}T_\mathrm{H}$ for Otto and Diesel cycles with different values of the ratio $T_\mathrm{H}/T_\mathrm{L}$

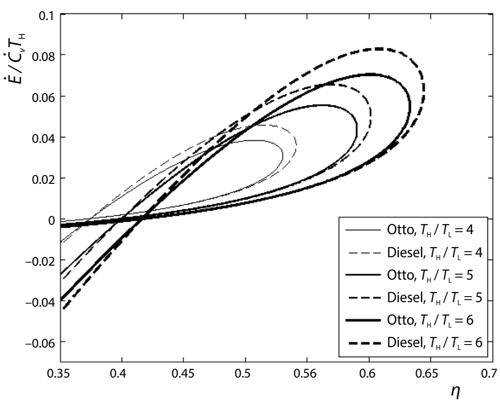

Figure 4. Dimensionless ecological function $\dot{E}/\dot{C}_v T_\mathrm{H}$ as function of thermal efficiency for Otto and Diesel cycles with different values of the ratio $T_\mathrm{H}/T_\mathrm{L}$

cycle has higher values of thermal efficiency and ecological power when compared to the Otto cycle. Thus, the increase of the parameter causes an increase in both thermal efficiency and ecological power. It is observed that for the same values of thermal efficiency in Otto and Diesel cycles, there is a significant difference in the values of the ecological function.

Analyzing fig. 5 it is observed that there is a point where the ecological function is maximum, while the entropy generation rate has a minimum point, the values of the entropy generation rate increasing to the point of ecological power. From this point, the ecological function is decreased due to the significant increase in the entropy generation rate. Thus, the operation points of minimum entropy generation rate imply very low values of net power output, significantly reducing the ecological function. The increased ratio T_H/T_L causes an increase of the ecological power for both cycles, while optimal entropy generation rate is shifted to the right, in the other words, their values increased. It is found that the Diesel cycle has high ecological power values when compared to Otto cycle, and also the values of the optimum entropy generation rate Diesel cycle increases in a discrete manner in relation to the Otto cycle for the same values of the ratio T_H/T_L.

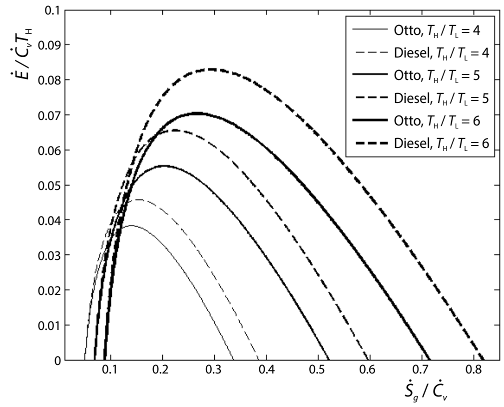

Figure 5. Dimensionless ecological function $\dot{E}/\dot{C}T_H$ as a function of dimensionless entropy generation rate for Otto and Diesel cycles with different values of the ratio T_H/T_L

It can be observed that the points of maximum ecological function have optimal power (fig. 3), efficiency (fig. 4), and entropy generation rate (fig. 5) points. In figs. 3 and 4, points of maximum power and efficiency are observed, respectively, whereas, in fig. 5, points of minimum entropy generation rate are verified. Thus, the behavior of the net power under conditions of maximum ecological function is 85% lower than the power at maximum conditions. Analyzing the thermal efficiency in conditions of maximum ecological function it is verified that it is about 20% greater than the thermal efficiency in conditions of maximum power. The entropy generation rate at conditions of maximum ecological function is about 50% less than the entropy generation rate at maximum power conditions.

Conclusions

This paper presents a mathematical modeling of the irreversible Otto and Diesel cycles. The cycles were modeled between two reservoirs of thermal energy, a reservoir of high temperature and another reservoir of low temperature. The irreversibility of these cycles is derived from the temperature difference between the thermal reservoir and the working fluid, the heat leakage rate, which is transferred directly from the high temperature to low temperature without any power output, and irreversibilities resulting from the non-isoentropic compression and expansion process, being expressed by the irreversibility parameter.

Through mathematical modeling of the cycles, two optimizations using different thermodynamic criteria were used: net power output and ecological function. The ecological optimization criterion was analyzed and compared on the criterion of net power allowing a better understanding of ecological function. When comparing the Otto and Diesel cycles in ecological optimizations, results of net power output and thermal efficiency, Diesel cycle has obtained better results than the Otto cycle for the same parameters of the cycles. Regarding the entropy generation rate, the Otto cycle showed slightly lower results than the Diesel cycle, but these values are overshadowed by the significant difference of maximum values of ecological function.

The ecological function had lower values of net power ecologically optimized for both the Otto cycle and Diesel cycle for (about 85% of maximum power), whereas the thermal efficiency under the condition optimized power greatly increased, reaching 20% increase in thermal efficiency at maximum power for Otto and Diesel cycle and finally the generation rate of entropy cycles fell to levels 50% of the rate generation entropy maximum power condition. In general optimization causes an ecological small drop in net power production, but there is a large decrease in the rate of generation of entropy and hence the gain comes in the form of an increase in the thermal efficiency of the cycle.

Acknowledgment

This work is supported by CAPES (Coordination of Improvement of Higher Education Personnel) and UNESP (Sao Paulo State University *Julio de Mesquita Filho*) and has the support of IFPR (Federal Institute of Parana).

Nomenclature

A – heat transfer area, [m²]
$\dot{C}_I$ – internal conductance rate of the heat engine, [WK^{-1}]
$\dot{C}_p$ – heat capacity rate at constant pressure of the working fluid, [WK^{-1}]
$\dot{C}_v$ – heat capacity rate at constant volume of the working fluid, [WK^{-1}]
$\dot{E}$ – ecological function, [W]
k – specific heat ratio, [–]
NTU – number of transfer units, [–]
$\dot{Q}$ – heat transfer rate, [W]
R – irreversibilities parameter
$\dot{S}_g$ – entropy generation rate, [WK^{-1}]
T – temperature, [K]
U – overall heat transfer coefficient, [Wm^{-2}K^{-1}]
$\dot{W}$ – power output, [W]

Greek symbols

ε – heat exchanger effectiveness, [–]
η – thermal efficiency, isentropic efficiency, [–]

Subscripts

0 – environment
Av – average
C – cycle
D – Diesel
H – hot side
I – heat leakage
L – cold side
O – Otto
S – isentropic
T – total
W – corresponding to power optimization

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Paper submitted: June 13, 2017
Paper revised: August 9, 2017
Paper accepted: August 24, 2017