Thermodynamic Assessment and Multi-Objective Optimization of Performance of Irreversible Dual-Miller Cycle

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Abstract

Although different assessments and evaluations of Dual-Miller cycle performed, specified output power and thermal performance associated with engine determined. Besides, multi objective optimization of thermal efficiency, Ecological Coefficient of performance (ECOP) and Ecological function (Eₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑᵉ(112,704),(872,942)

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Based on the results, performances of dual-Miller cycles and their optimization are improved.

**Keywords:** Dual-Miller cycle; thermodynamic analysis; power; ecological coefficient of performance; thermal efficiency; entropy generation; multi-objective optimization

**Nomenclature**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>DMC</td>
<td>Dual-Miller cycle</td>
</tr>
<tr>
<td>( m )</td>
<td>Mass flow rate</td>
</tr>
<tr>
<td>( n )</td>
<td>Polytropic exponent</td>
</tr>
<tr>
<td>( k )</td>
<td>The specific heat ratio (adiabatic exponent)</td>
</tr>
<tr>
<td>( P )</td>
<td>Power</td>
</tr>
<tr>
<td>( Q )</td>
<td>Heat</td>
</tr>
<tr>
<td>( T )</td>
<td>Temperature</td>
</tr>
<tr>
<td>( ECOP )</td>
<td>Ecological Coefficient of Performance</td>
</tr>
<tr>
<td>( E_{\text{un}} )</td>
<td>Ecological function</td>
</tr>
<tr>
<td>( V )</td>
<td>Volume</td>
</tr>
<tr>
<td>( C_v )</td>
<td>The specific heat at constant volume</td>
</tr>
<tr>
<td>( C_p )</td>
<td>The specific heat at constant pressure</td>
</tr>
<tr>
<td>( \sigma_{\text{un}} )</td>
<td>Total entropy generation</td>
</tr>
<tr>
<td>( \eta )</td>
<td>Efficiency</td>
</tr>
<tr>
<td>( r_M )</td>
<td>The Miller cycle ratio of a Dual-Miller cycle</td>
</tr>
<tr>
<td>( \rho )</td>
<td>The cut-off ratio</td>
</tr>
<tr>
<td>( \lambda )</td>
<td>The pressure ratio</td>
</tr>
<tr>
<td>( \varepsilon )</td>
<td>The compression ratio</td>
</tr>
</tbody>
</table>

1. **Introduction**

FTT is one of the most reliable optimization tools to assess the performance of internal combustion engine cycles (ICEC) [1–13]. Recent studies of thermodynamic systems [14–23], comprehensive investigations have been carried out [24]. Entropy optimization [25–29], \( E_{\text{un}} \) criterion [30–34] and \( ECOP \) criterion [35–40] are some of the recent various optimization objectives in ICEC analysis. Generally, entropy reduction is not equal to enhancing thermal efficiency or maximum power generation. Under certain circumstances, minimizing entropy generation leads to the highest power generation [41]. Blank et al. [42] investigated the efficiency of an endoreversible air standard dual cycle considering the system heat loss. Chen and colleagues [43] investigated an air standard dual cycle taking the friction and heat loss into
Multi-objective optimization is a valuable method to overcome various engineering difficulties [69-71]. Answering a multi-objective optimization question needs simultaneous substantiation of various objectives. Consequently, evolutionary algorithms presented and advanced to answer multi-objective problems applying various methods [72]. A proper approach to find a solution to for a multi-objective problem is to examine a group of routes, each satisfies the objectives at an acceptable level and do not interfere with other routes [73]. Multi-objective optimization problems generally represent a practicably numerous collection of routes named frontier of Pareto, which examined vectors show the possible primary connections in the whole area of the objective function. New studies indicate that multi-objective optimizations for different thermodynamic cycles applied in various engineering problems [74-101].

In this study, investigated the performance of Irreversible Dual-Miller Cycle. Also, the presented effect of critical parameters on the performance of Dual-miller cycle. Key parameters that presented include $\varepsilon$, $\rho$ and the $n$. The effects of these parameters on the power, efficiency, $ECOP$ and $E_{un}$ of the system evaluated. Then done multi-objective optimization to obtain the best point of performance of Dual-Miller Cycle.

2. Dual-Miller cycle through a polytropic stage
An air standard Dual-Miller cycle is presented in Fig.1. To increase the accuracy of performance assessment, the polytropic process replaces by the reversible adiabatic stage, which is impractical to attain in the improved Dual-Miller cycle [64]. As it is depicted in Figure 1, cycle 1–2–3–4–5–6–1 represents the condition in which $n=k$. 
Fig. 1 T-S diagram of DMC: \( n \) less than \( k \) (a) and \( n \) higher than \( k \) (b) [102].

2.1 Ideal air standard Dual-Miller cycle

In ideal gas systems \((n = k)\), the state elements of each stage can be practically obtained, by the ideal gas state equation. The first law of thermodynamics clear that the heat transfer rates and power generation of the cycle can be determined. Equation (1) presents \( \varepsilon \), \( \lambda \), \( \rho \) and \( r_\text{M} \), respectively:

\[
\varepsilon = \frac{V_1}{V_2}, \quad \lambda = \frac{P_3}{P_2}, \quad \rho = \frac{V_4}{V_3}, \quad r_\text{M} = \frac{V_6}{V_1}
\]  

(1)

The primary thermodynamic equations of each stage defined as:

\[
T_2 = T_1 \varepsilon^{k-1}
\]  

(2)

\[
T_3 = T_2 \lambda
\]  

(3)
\[ T_4 = T_3 \rho \] (4)
\[ T_5 = T_4 \left( \frac{\rho}{\varepsilon r_M} \right)^{k-l} \] (5)
\[ T_6 = T_5 \left( \frac{r^k_M}{\lambda r^k} \right) \] (6)
\[ T_i = \frac{T_6}{r_M} \] (7)

The heat transfer ratios of the system fluid are as follows:
\[ Q_{in} = Q_{23} + Q_{34} = \dot{m} \left( C_v (T_3 - T_2) + C_p (T_4 - T_3) \right) \] (8)
\[ Q_{out} = Q_{56} + Q_{61} = \dot{m} \left( C_v (T_5 - T_6) + C_p (T_6 - T_4) \right) \] (9)

In the ideal reversible air standard Dual-Miller cycle, the heat transfer impact is not considered, while it considered for an actual DMC. This waste is considered relevant to the temperature difference of working fluid and the cylinder wall as follows [102,103]
\[ Q_1 = \frac{B}{2} (T_2 + T_4 - 2T_0) \] (10)

The power production and performance are calculated as:
\[ P = Q_{in} - Q_{out} \] (11)
\[ \eta = \frac{P}{Q_{in} + Q_1} \] (12)

### 2.2 Air standard Dual-Miller cycle

As it is depicted in Fig.1, T1 in a Dual-Miller cycle (\( n \) less than \( k \)), is higher than the T1 in air standard Dual-Miller cycle. In order to keep T2 fixed through the compression stage, heat must be extracted through the polytropic stage 1′–2. Considering heat waste through the heat input stage, T4′ is lower than T4. In order to keep T5 fixed, heat should be increased through the polytropic stage 4′–5.

The equations of each stage are defined as:
\[ T_2 = T_1 \varepsilon^{\left( \frac{k(n-1)}{n} \right)} \] (13)
The heat transfer rate of polytropic stage 1’–2, is defined as follows:

\[ Q_{1'2} = mC_v \left( \frac{k - n}{n - 1} \right) (T_2 - T_1') \]  \hspace{1cm} (15)

The heat transfer rate of stages 2–3 and 3–4’, are defined as follows:

\[ Q_{23} = mC_v (T_3 - T_2) \]  \hspace{1cm} (16)

\[ Q_{34'} = mC_p (T_4' - T_3) \]  \hspace{1cm} (17)

The heat transfer rate of stage 4’–5, is defined as follows:

\[ Q_{4'5} = mC_v \left( \frac{k - n}{n - 1} \right) (T_4' - T_5) \]  \hspace{1cm} (18)

The heat transfer rate of stages 5–6 and 6–1’, are defined as follows:

\[ Q_{56} = mC_v (T_5 - T_6) \]  \hspace{1cm} (19)

\[ Q_{61'} = mC_p (T_6' - T_1') \]  \hspace{1cm} (20)

The heat input of the cycle is

\[ Q_{in1} = Q_{23} + Q_{34'} + Q_{4'5} \]  \hspace{1cm} (21)

The heat output of the cycle is

\[ Q_{out1} = Q_{1'2} + Q_{56} + Q_{61'} \]  \hspace{1cm} (22)

For an ideal air standard Dual-Miller cycle, the ratio of the highest temperature to the lowest temperature is defined as follows:

\[ \frac{T_{4'}}{T_i} = \rho \lambda e^{k-l} \]  \hspace{1cm} (23)

As stated by the ref. [102,103], the heat waste ratio is defined as follows:

\[ Q_{lw} = \frac{B}{2} (T_2 + T_4' - 2T_0) \]  \hspace{1cm} (24)
As a result, the generated power and the first law efficiency of the system are defined as follows:

$$P_1 = Q_{in1} - Q_{out1}$$  \hspace{1cm} (25)

$$\eta = \frac{P_1}{Q_{in}} = \frac{P_1}{Q_{23} + Q_{34'} + Q_{45} + Q_{11}}$$  \hspace{1cm} (26)

Considering the assumption in ref. [104], the exhaust gas recirculation due to the heat transfer loss is determined as follows:

$$\sigma_{q1} = \frac{B(T_2 + T_{4'} - 2T_0)}{2T_0}$$  \hspace{1cm} (27)

The exhaust gas recirculation due to the working fluid heat rejection is defined as [105]:

$$\sigma_{pq1} = m \left( \int_{T_{1}}^{T_{2}} C_p \left( \frac{l}{T_0} - \frac{l}{T} \right) dT + \int_{T_{2}}^{T_{3}} C_v \left( \frac{l}{T_0} - \frac{l}{T} \right) dT + \int_{T_{3}}^{T_{4}} C_v \left( \frac{k-n}{n-1} \right) \left( \frac{l}{T_0} - \frac{l}{T} \right) dT \right)$$  \hspace{1cm} (28)

As a result, the total entropy generation ($\sigma_{un1}$) of the system is defined as follows:

$$\sigma_{un1} = \sigma_{q1} + \sigma_{pq1}$$  \hspace{1cm} (29)

According to refs. [30–34], ECOP of the cycle is defined as follows:

$$ECOP = \frac{P_1}{T_0 \sigma_{un1}}$$  \hspace{1cm} (30)

According to references. [30–34], $E_{un}$ is defined as follows:

$$E_{un1} = P_1 - T_0 \sigma_{un1}$$  \hspace{1cm} (31)

### 2.3 Air standard Dual-Miller cycle

As it is depicted in Fig.2, T2 the highest temperature of the adiabatic stage 1–2 is less than that of the polytropic stage, due to the heat waste through the isochoric stage 2–3 ($n$ higher than $k$). Thus, more heat should be applied through the polytropic stage 1–2'. On the other hand, heat is extracted through the stage 4–5' as T5 the minimum temperature of the adiabatic stage 4–5.
is greater than that of the polytropic stage 4–5' taking the heat waste through the isobaric stage 3–4, into account.

The equations of polytropic stages are defined as follows:

\[ T_{2'} = T_1 e^{n-1} \]  
\[ T_{5'} = T_4 \left( \frac{P}{e \cdot r_m} \right)^{n-1} \]

For polytropic stage 1–2', the heat transfer ratio is defined as:

\[ Q_{12'} = \dot{m} C_v \left( \frac{n-k}{n-1} \right) (T_{2'} - T_1) \]  

For stages 2'–3 and 3–4, heat transfer rates are defined as:

\[ Q_{23} = \dot{m} C_v (T_3 - T_{2'}) \]  
\[ Q_{34} = \dot{m} C_p (T_4 - T_3) \]

For stage 4–5', the heat transfer rate is defined as:

\[ Q_{45'} = \dot{m} C_v \left( \frac{n-k}{n-1} \right) (T_4 - T_{5'}) \]

For stages 5'–6 and 6–1', heat transfer rates are defined as:

\[ Q_{56} = \dot{m} C_v (T_5' - T_0) \]  
\[ Q_{61} = \dot{m} C_p (T_6 - T_1) \]

The net heat input ratio is calculated as:

\[ Q_{\text{in}2} = Q_{12'} + Q_{23} + Q_{34} \]

The net heat output ratio is calculated as:

\[ Q_{\text{out}2} = Q_{45'} + Q_{56} + Q_{61} \]

The heat waste ratio is calculated as [102,103]:

\[ Q_{12} = \frac{B}{2} (T_{2'} + T_4 - 2T_0) \]

The power generation and the first law efficiency of the system are defined as follows:

\[ P_2 = Q_{\text{in}2} - Q_{\text{out}2} \]  
\[ \eta_2 = \frac{P_2}{Q_{\text{in}}} = \frac{P_2}{Q_{12'} + Q_{23} + Q_{34} + Q_{12}} \]

The exhaust gas recirculation of the heat transfer loss is calculated as follows [104]:
\[ \sigma_{q2} = \frac{B(T_2 + T_4 - 2T_0)}{2T_0} \]  

(45)

The exhaust gas recirculation due to the working fluid heat rejection is as follows [105]:

\[ \sigma_{pq2} = m\left( \int_{r_1}^{r_0} C_p \left( \frac{1}{T_0} - \frac{1}{T} \right) dT + \int_{r_0}^{r_1} C_v \left( \frac{1}{T_0} - \frac{1}{T} \right) dT + \int_{r_1}^{r_2} C_v \left( \frac{n-k}{n-1} \right) \left( \frac{1}{T_0} - \frac{1}{T} \right) dT \right) \]  

(46)

The total exhaust gas recirculation of the system is defined as follows:

\[ \sigma_{un2} = \sigma_{q2} + \sigma_{pq2} \]  

(47)

According to refs. [30–34], ECOP \( E \) of the cycle is defined as follows:

\[ ECOP = \frac{P_2}{T_0 \sigma_{un2}} \]  

(48)

The \( E \) is defined as follows:

\[ E_{un2} = P_2 - T_0 \sigma_{un2} \]  

(49)

3. Optimization Development: Evolutionary algorithm

3.1. Genetic algorithm

Genetic algorithms provide the best suitable answer of the studding system employing a repetitious and random exploration approach and duplicate it by simple basics of biological evolution [72]. The individual that is a possible solution to the optimization case [73] presents the values of the decision elements. More explanations about Genetic algorithms and its function is available in References [72, 73].

Results and Discussions

4.1 Performance analyses for the condition \( n \) less than \( k \)

Fig.2 depicts the impact of \( n \) on the performance relations among power, efficiency and compression ratio. It is evident that as \( \varepsilon \) increases, \( P_1 \) and \( \eta_1 \) initially increase and finally decrease. It should be noted that that \( P_1,\max \) and \( \eta_1,\max \) do not take place at the same time. On the other hand, \( P_1,\max \) and \( \eta_1,\max \) increase by the enhancement of \( n \). Furthermore, the efficiency at maximum power rises by the enhancement of \( n \).
Fig. 2 Impact of $n$ ($n < k$) on $P_1 - \varepsilon$ (a), $\eta_1 - \varepsilon$ (b) and $P_1 - \eta_1$ (c) relations.

Fig. 3 illustrates the impact of $\rho$ on the relationship between $P_1$ and $\varepsilon$ at $n = 1.2$.

Fig. 4 depicts the $E_{un}$ changes against $P_1$ and $\eta_1$ relations at various $n$. It is obvious that $n$ has a direct relationship with $P_1$ and $\eta_1$. The maximum $E_{un}$ point is adjacent to $P_1,\text{max}$ and $\eta_1,\text{max}$. In other words, optimum values of $P_1$ and $\eta_1$ could be achieved when $E_{un}$ is optimized.
As shown in Fig. 5a. As the compression ratio increases, with a very steep gradient, ECOP first increases to its maximum point and then begins to decrease. Also, in a constant compression, the ECOP increases with the increase of the $n (n<k)$. Figures 5b and 5c show that the maximum value of the coefficient of performance for various $n (n<k)$ will occur at almost the maximum power and maximum thermal efficiency.
4.2 Performance evaluation at the condition of $n$ higher than $k$

Fig.6 depicts the impact of $n$ on the performance relations among power, efficiency and compression ratio. It is obvious that when $\varepsilon$ increases, $P_1$ and $\eta_1$ initially increase and finally
decrease. Enhancing $n$ leads to slowly reduction of $P_2$ and $\eta_2$. It should be noted that $P_2$, max and $\eta_2$, max, do not take place at the same value of epsilon. Hence, the $\eta_2$ at $P_2$, max reduces by enhancement of $n$. 

(a) 

(b)
Fig. 6 Impact of $n$ ($n>k$) on $P_2 - \varepsilon$ (a), $\eta_2 - \varepsilon$ (b) and $P_2 - \eta_2$ (c).

Fig. 7 illustrates the impact of $\rho$ on the relationship between $P_2$ and $\varepsilon$ at $n = 1.6$.

Fig. 7 Effect of $\rho$ on $P_2$ against $\varepsilon$ ($n=1.6$).

Figure 8 presents the $E_{un}$ impact on $P_2$ and $\eta_2$ at various $n$. It is obvious that increasing $n$ leads to $E_{un}$, and $E_{un}$ reduction. The $E_{un}$, max is adjacent to $P_2$,max and $\eta_2$,max. In other words, optimum values of $P_2$ and $\eta_2$ could be achieved when $E_{un}$ is optimized.
As shown in Fig. 9a. As the compression ratio increases, with a very steep gradient, the \textit{ECOP} first increases to its maximum point and then begins to decrease. In addition, in a constant compression, the \textit{ECOP} decreases with the increase of the \( n > k \). Figures 9b and 9c show that the maximum value of the coefficient of performance for various \( n > k \) will occur at almost the maximum power and maximum thermal efficiency.
Fig. 9 Affect of \( n (n>k) \) on \( ECOP - \epsilon \) (a), \( ECOP - P_z \) (b) and \( ECOP - \eta_z \) (c).

4.3. Optimization results for \( n \) less than \( k \)
Three objective functions are utilized in this optimization: $\eta_1$, $ECOP$ and $E_{unl}$, described by Eqs. (26), (30) and (31), respectively. Also, three decision variables are considered: $\varepsilon$, $\rho$ and $n$.

Although the decision variables might be various in the optimizing plan, but they typically need to be fitted in a sensible range. Thus, the objective functions are determined by relative limits:

1. $18 \leq \varepsilon \leq 25$  \hspace{1cm} (50)
2. $1.5 \leq \rho \leq 1.8$  \hspace{1cm} (51)
3. $1.2 \leq n \leq 1.4$  \hspace{1cm} (52)

In this study $\eta_1$, $ECOP$ and $E_{unl}$ of the dual Miller cycle are maximized concurrently employing multi-objective optimization by the mean of the NSGA-II approach. The objective functions are illustrated by Eqs. (26), (30) and (31) and the limitations by Eqs. (50) - (52).

The decision parameters of optimization are as follow: $\varepsilon$, $\rho$ and $n$. The Pareto optimal frontier of objective functions (the thermal efficiency, $ECOP$ and $E_{unl}$) is depicted in Fig. 10. Selected points with different decision-making methods are presented, as well.
Fig. 10. The distribution of the Pareto optimal frontier

Table 1 outlines and compares the optimal results associated with decision elements and objective functions utilizing LINMAP, TOPSIS, and Bellman-Zadeh decision-making methods.

Table 1. Decision making results of this study ($n < k$).

<table>
<thead>
<tr>
<th>Decision Making Method</th>
<th>Decision variables</th>
<th>Objectives</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$\varepsilon$</td>
<td>$\rho$</td>
</tr>
<tr>
<td>TOPSIS</td>
<td>19.4528063</td>
<td>1.685157</td>
</tr>
<tr>
<td>LINMAP</td>
<td>19.865730</td>
<td>1.640773</td>
</tr>
<tr>
<td>Fuzzy</td>
<td>20.070760</td>
<td>1.542415</td>
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</table>
4.4. Optimization results for $n$ higher than $k$

Three objective functions are utilized in this optimization: $\eta$, $ECOP$ and $E_{un2}$, described by Eqs. (44), (48) and (49), respectively. Also, tree decision variables are considered: $\varepsilon$, $\rho$ and $n$.

Although the decision variables might be different in the optimizing plan, but they typically need to be fitted in a sensible range. Thus, the objective functions are determined by relative limits:

$$18 \leq \varepsilon \leq 25$$  
$$1.5 \leq \rho \leq 1.8$$  
$$1.4 < n \leq 1.6$$

In this study, $\eta$, $ECOP$ and $E_{un2}$ of the dual Miller cycle are maximized concurrently utilizing multi-objective optimization based on the NSGA-II approach. The objective functions are illustrated by Eqs. (48), (49) and (50) and limitations by Eqs. (53) - (55).

The decision parameters of optimization are as follow: $\varepsilon$, $\rho$ and $n$. The Pareto optimal frontier of objective functions (the thermal efficiency, $ECOP$ and $E_{un2}$) is depicted in Fig. 11. Selected points with different decision-making methods are presented, as well.
Fig. 11. The distribution of the Pareto optimal frontier

Table 2 outlines and compares the optimal results associated with decision elements and objective functions utilizing LINMAP, TOPSIS, and Bellman-Zadeh decision-making methods.

Table 2. Decision making results of this study \((n>k)\).

<table>
<thead>
<tr>
<th>Decision Making Method</th>
<th>Decision variables</th>
<th>Objectives</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>(\zeta )</td>
<td>(\rho )</td>
</tr>
<tr>
<td>TOPSIS</td>
<td>20.218658</td>
<td>1.687013</td>
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<tr>
<td>LINMAP</td>
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</tr>
<tr>
<td>Fuzzy</td>
<td>20.382900</td>
<td>1.524323</td>
</tr>
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5. Conclusions
A thermodynamic optimization has been carried to obtain the thermal efficiency, $ECOP$ and $E_{un}$ of the Dual-Miller Cycle. The compression ratio, the cut-off ratio, and the polytropic index are examined by the NSGA-II approach. Employing various decision-making methods (LINMAP, TOPSIS and fuzzy), the best optimum answer selected from the Pareto frontier. The study achieved a promising and satisfactory state of operation for Dual-Miller systems. The three methods give closed results (with a relative difference less than 3% on compression ratio, 5% on cut-off ratio, 2% on the objective function.

References


