



SDI FINAL EVALUATION FORM 1.1

PART 1:

Journal Name:	British Journal of Applied Science & Technology
Manuscript Number:	2014_BJAST_12111
Title of the Manuscript:	Power plant driven by residual heat rejected by the bottoming low pressure steam turbines

PART 2:

FINAL EVALUATOR'S comments on revised paper (if any)	Authors’ response to final evaluator’s comments
I would like to thank you for your response to this review. In your response you still write that it is not a Carnot engine. I agree with this statement but in my comments you may read that they don’t say that your cycle has to be a Carnot engine in order to be bound by the Carnot statement. In fact, the standard textbooks refers to all thermodynamic cycles and not just the Carnot cycle. This thus includes your cycle as well. Your cycle has striking resemblance to the Brayton cycle used for gas turbines. This is why you need to give an exhaustive proof of your statement about the cycle efficiency that you present.	Thank you. Your comments and questions about the Carnot engine are reasonable and consistent according to the conventional fundamentals of the known thermodynamics. However in what follows I will try to explain your reasonable doubts by using as an example a reciprocating piston-cylinder. Consequences of the Kelvin Plank Statement <i>It is impossible to construct such a Heat Engine that operates on cyclic process and converts all the heat supplied to it into an equivalent amount of work.</i> The following conclusions can be made from the statement 1. No cyclic engine can converts whole of heat into equivalent work. 2. There is degradation of energy in a cyclic heat engine as some heat has to be degraded or rejected. - a) 2d principle doesn’t consider at all the meaning of the so called Carnot statement. According to the Kelvin Plank statement the thermal efficiency of any thermal engine must be lower than 100%. Nothing more has been said. Nothing has been stated with regard to the Carnot limitations. -b) The meaning of the Carnot efficiency or Carnot Factor (CF) defined as $CF = 1 - \frac{T_L}{T_H}$ Doesn’t express nothing more than the thermal efficiency of a Carnot based engine (composed by two isothermal processes and two isentropic processes). However the statement has been interpreted to be applied to all types of thermal engines, which is a flagrant violation and extrapolation of the meaning of the formula defining the Carnot Factor. Inexplicably, the scientific community has admitted this misunderstanding or error over centuries, for more than 180 years. According to the arguments given in a) and b), follows that other possible types of thermal engines may not obey to the Carnot statement, which is partially correctly expressed when refers to <i>all thermal engines</i> instead of <i>Carnot based engines</i>. Comparison of Brayton and proposed cycle The proposed cycle, - is implemented undergoing a single or double reciprocation cylinder - is active (performs mechanical work) while absorbing heat, so that performs mechanical work while is absorbing heat energy. -rejects heat of high grade so that this energy is useful to use in the regeneration.



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The Brayton cycle

- develops mechanical work by a rotating machine which undergoes energy degradation due to its adiabatic expansion.
- Rejects heat along the transformation (4)-(1) at lower grade due to the degradation of the heat energy during its quasi-adiabatic expansion along the transformation (3)-(4).
- develops mechanical work while degrading the heat energy due to the quasi-isentropic expansion and compression.

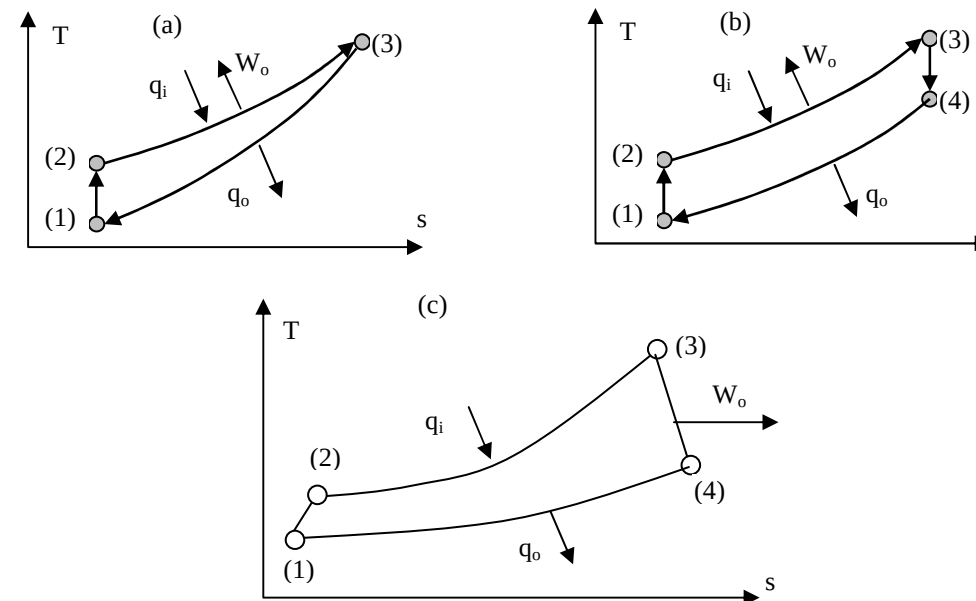


Fig. 1. Comparison of non regenerative proposed and Brayton cycles. (a) Trilateral proposed cycle. (b), Quadrilateral proposed cycle. (c) Brayton cycle.

In terms of its physical structure, the configuration that most resembles the proposed cycle corresponds to a reciprocating steam Rankine cycle in which the admission of steam to the piston-cylinder is performed throughout his career, which means along the full cylinder stroke.

Closed processes based cycle to demonstrate the difference between Carnot based engines and the proposed cycle

1. Analysis of the heating process following load-dependent path functions

In what follows, the studied trilateral thermal cycle undergoes three closed transformations instead of three open processes. Let's assume that heat is added to a unit mass of working fluid following an isobaric path function from the state point (2) to the state point (3) as shown in Fig. 2, undergoing a structure similar to that of a reciprocating piston-cylinder, and suppose the resulting expansion causes the state variables to change from state (2) to state (3).



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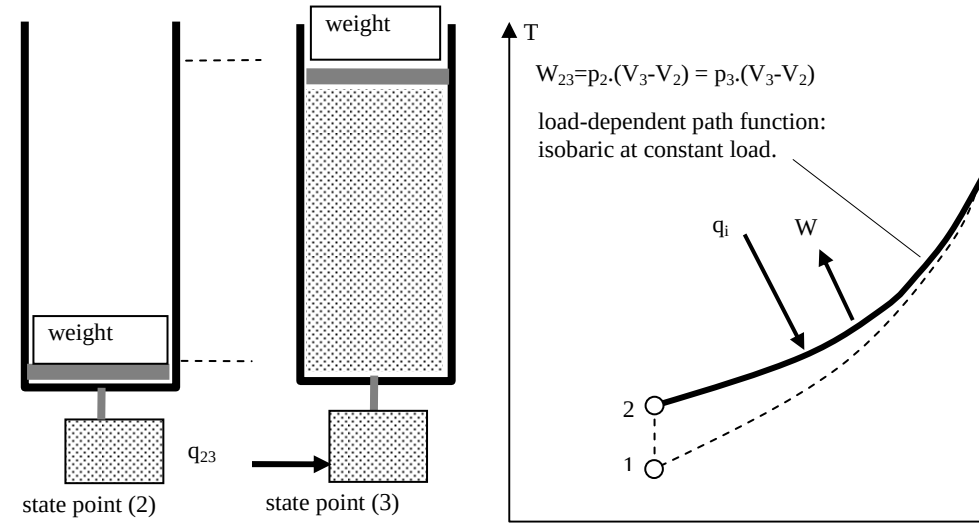


Fig. 2. The process of heating a working fluid at constant pressure (closed process). The fluid is confined into the heater and the cylinder. At constant load, the pressure is constant so that the heating process undergoes an isobaric path function

The first principle for closed transformations where irreversibilities, potential and kinetic energy are neglected yields

$$\sum q - \sum W - \Delta u = 0 \rightarrow q_{23} - W_{23} = \Delta u = u_3 - u_2 \quad (1)$$

where the work W_{23} done by a unit mass of the working fluid is defined as

$$W_{23} = \int_{V_2}^{V_3} p dV = p_2(V_3 - V_2) = p_3(V_3 - V_2) \quad (2)$$

Equation (2) can be expressed according to the definition given in (1) yielding

$$W_{23} = q_{23} - \Delta u_{23} \quad (3)$$

The C_R (conversion ratio) or efficiency of this closed reversible transformation (2)-(3) is

$$C_R = \eta_{23} = \frac{W_{23}}{q_{23}} = \frac{q_{23} - \Delta u_{23}}{q_{23}} = 1 - \frac{\Delta u_{23}}{q_{23}} = 1 - \frac{C_v \cdot (T_3 - T_2)}{C_p \cdot (T_3 - T_2)} = 1 - \frac{1}{\gamma} \quad (4)$$

Surprisingly, the expression defining the C_R or transformation efficiency exhibits very low dependence on the temperature, that is, practically is not temperature dependent.

This suggests to us that for low top temperatures the Carnot efficiency is lower than the value of this ideal thermal efficiency, since when the top temperature approaches the bottom temperature CF tends towards zero.

Considering an approach to ideal gases such as helium or argon, the corresponding conversion ratio definition yields



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$$\eta_{23} = \frac{W_{23}}{q_{23}} = 1 - \frac{1}{\gamma_{He}} = 1 - \frac{1}{1.665} = 0.4 \quad \text{for helium as working fluid,} \quad (5)$$

$$\eta_{23} = 1 - \frac{1}{\gamma_{Ar}} = 1 - \frac{1}{1.67} = 0.4012 \quad \text{for argon, and,} \quad (6)$$

$$\eta_{23} = 1 - \frac{1}{\gamma_{H2}} = 1 - \frac{1}{1.41} = 0.29 \quad \text{for hydrogen} \quad (7)$$

for any temperature $T_3 > T_1$.

From expressions (4) to (7) follows that the conversion ratio C_R is only a function of the adiabatic expansion coefficient, so that independent of the temperatures ratio.

Nevertheless, in order to convert heat to mechanical work under useful modes, a thermal cycle must be applied, and this requirement undergoes the implementation of a thermal engine structure. In such case, the cycle efficiency η will not match the C_R for which the maximum efficiency could not be equal to the efficiency achieved in (4), so that $\eta < C_R$ to fulfil the 2^d law. Thus, next sub-section deals with such subject when a complete cycle is considered.

2. Analysis of a Trilateral Thermal Cycle Composed by Closed System Based Transformations

Considering now a thermal cycle composed by closed system based transformations, with reference to the Fig. 3 for the structure and Fig. 4 for the T-s and the p-V diagrams, follows that the energy balance of the TTC has been considered assuming that the process (2)-(3) is a closed transformation, since there is not exchange of matter, so that the amount of specific energy added from an external heat source is defined as

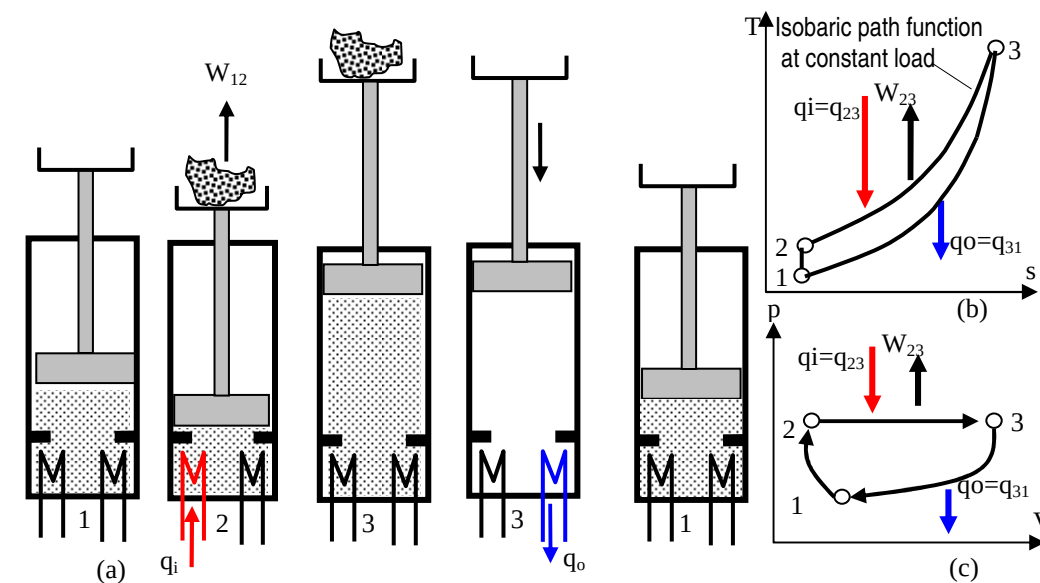


Fig. 3. A thermal cycle composed by closed systems transformations. (a), a trilateral cycle consisting of a cylinder that absorbs heat q_i , delivering work simultaneously along the transformation (2)-(3) and rejecting heat during the transformation (3)-(1). (b), the T-s diagram of the trilateral cycle. (c), the corresponding p-V diagram. At constant load, the pressure is constant so that the heating process undergoes an isobaric path function



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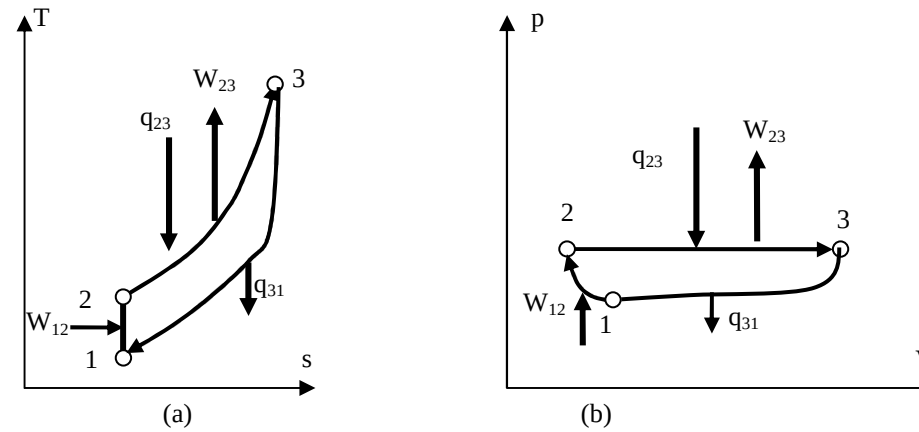


Fig. 4. Diagrams of the cycle. (a), T-s diagram. (b), p-V diagram

$$q_i = q_{23} = W_{23} + \Delta u_{23} = p_2 \cdot (V_3 - V_2) + u_3 - u_2 \quad (8)$$

and for the process (3)-(1) the amount of heat rejected to the low temperature heat sink considered also as a closed transformation yields

$$q_o = q_{31} = W_{31} + \Delta u_{31} = 0 + \Delta u_{31} = u_3 - u_1 \quad (9)$$

Consequently the net mechanical work is

$$W_N = q_i - q_o = q_{23} - q_{31} = W_{23} + \Delta u_{23} - \Delta u_{31} = W_{23} - u_2 + u_1 \quad (10)$$

Thus the thermal efficiency is given as

$$\eta = \frac{W_N}{q_i} = \frac{q_i - q_o}{q_i} = \frac{W_{23} + u_1 - u_2}{W_{23} + u_3 - u_2} \quad (11)$$

From (11) follows that the thermal efficiency can also be expressed in terms of the state point temperatures, so that

$$\begin{aligned} \eta &= \frac{W_N}{q_{23}} = \frac{q_{23} - q_{31}}{q_{23}} = \frac{Cp \cdot (T_3 - T_2) - Cv \cdot (T_3 - T_1)}{Cp \cdot (T_3 - T_2)} = \\ &= 1 - \frac{Cv \cdot (T_3 - T_1)}{Cp \cdot (T_3 - T_2)} = 1 - \frac{1}{\gamma} \frac{(T_3 - T_1)}{(T_3 - T_2)} \end{aligned} \quad (12)$$

From expression (12) follows that for a very low pressure ratio (p_2/p_1) the T_2 tends to T_1 . Consequently, the maximum attainable ideal efficiency will be achieved as temperature T_2 approaches T_1 , as shown below, yielding

$$\lim_{T_2 \rightarrow T_1} \eta = \lim_{T_2 \rightarrow T_1} 1 - \frac{1}{\gamma} \frac{(T_3 - T_1)}{(T_3 - T_2)} \rightarrow 1 - \frac{1}{\gamma} \quad (13)$$

The result given in (13) is similar to the result achieved by (4) which defines the maximum achievable efficiency. As can be observed, the maximum theoretical thermal efficiency is not dependent on any cycle temperature, but on the adiabatic expansion coefficient, so that this efficiency cannot be related although



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	compared with the Carnot factor. Furthermore, as shown below, there are a range and ratio of operating temperatures for which a flagrant violation of Carnot statement without violating the Clausius or Kelvin-Plank statements (second law) exist. Furthermore, for low temperatures, the value corresponding to the expression (13) is greater than its corresponding CF. Consequently, Carnot statement is not applicable to load driven trilateral thermal cycles composed by closed systems based transformations. Thus, since the ideal (maximum obtainable efficiency) is $\eta = 1 - \frac{1}{\gamma}$ and $CF = 1 - \frac{1}{T_R}$, follows that for $\eta = CF, \quad \gamma = T_R \tag{14}$ which implies that $CF < \eta \quad \text{for } T_R < \gamma \tag{15}$ Based on equation (14), the following statement is proposed: “In the thermal cycles composed by closed system based transformations in which $T_3 > T_1$, there is a ratio of temperatures $T_R = T_3/T_1$ such that $\eta > CF$”, since for $T_R = 1$, (that is $T_3 = T_1$), $CF = 0$. Thus, the maximum temperature T_3 for which $CF < \eta$ is achieved from (14) as $\gamma = T_R = \frac{T_3}{T_1},$ resulting in $T_3 = T_1 \cdot \gamma \tag{16}$ Assuming helium, argon and hydrogen as working fluids, and applying (16) to obtain the maximum temperatures for which the Carnot factor is lower than the thermal efficiency, the performed analysis yields the results shown in Table 1. Table 1. The maximum temperature T_3 for which CF is lower than γ (adiabatic expansion coefficient) for helium, argon and hydrogen as working fluids. <table> <tr> <th></th> <th>γ</th> <th>$T_1(K)$</th> <th>$T_3(K)$</th> </tr> <tr> <td>He</td> <td>1.665</td> <td>295</td> <td>491.2</td> </tr> <tr> <td>Ar</td> <td>1.67</td> <td>295</td> <td>492.6</td> </tr> <tr> <td>H₂</td> <td>1.41</td> <td>295</td> <td>416</td> </tr> </table> Thus, table 1 summarises the range of temperatures for which the thermal efficiency is greater than the CF for helium, argon and hydrogen as working fluids.		γ	$T_1(K)$	$T_3(K)$	He	1.665	295	491.2	Ar	1.67	295	492.6	H₂	1.41	295	416
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