OPTIMIZATION OF AN AGRAWAL HEAT ENGINE BY MEANS OF THE SAVING FUNCTIONS

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ABSTRACT

Finite Time Thermodynamics (TTF) (Andresen B., Salamon P. and Berry R. S., 1997) is a branch of Thermodynamics that is dedicated to obtaining theoretical models of the operation of power plants, operating limits on thermodynamic variables such as: efficiency, power and entropy production among other thermodynamic variables. One method to improve the design of such power plants through TTF is to propose objective functions applied to different thermal engines, such as the Curzon-Ahlborn (CA) heat engine (1975). Velasco et al (2000) presented a new objective function in terms of a cost-effective type process applied to a CA heat engine. The objective function defined as a savings function, which allows a reduction of unwanted secondary effects in the operation of the power plant. Agrawal (2009) proposed a simplified version of the CA heat engine, in which he assigns the same thermal resistance and the same temperature difference to the upper and lower isotherms of the Carnot cycle. In the engine model proposed by Agrawal, the efficiency at maximum power coincides with efficiencies values reported for real engines, but with a slightly lower power production. In 2021, Barranco-Jiménez et al (2021) made use of the so-called saving functions to analyze the Novikov (1958) heat engine model by varying the degree of participation of the processes proposed by Velasco et al (2000), in addition to considering two different heat transfer laws.

In this work we propose the use of savings functions following the approach used by Velasco but applied to the Agrawal engine, through a Newton-type heat transfer law between the energy reservoirs. In addition, an analysis is presented on the variation of the weight coefficients such as that carried out by Barranco-Jiménez et al (2021) to finally present a numerical comparison with the reported efficiency of some real power plants.

1 INTRODUCTION

Some of the objectives of the TTF is to determine theoretical models of operating modes of power plants in order to optimize them and obtain some thermodynamic variables such as efficiency, output power, entropy production, among others. Various authors have proposed objective functions such as the maximum power criterion (Hoffmann et al, 1997), the efficient power criterion (Yilmaz, 2006), the so-called ecological function (Angulo., 1992), the maximum power density (Sahim et al, 1996) among others. A very studied heat engine model is the CA engine, we will describe it below, the schematic representation of the Curzon-Ahlborn (1975) heat engine is shown in Figure 1, where T_1 is the temperature of the hot reservoir, T_{1w} represents the temperature of the working substance when it is in contact with the reservoir T_1 and α the thermal conductivity between the reservoir hot thermal and the working substance, T_2 represents the temperature of the cold reservoir, T_{2w} the temperature of the working substance when it is in contact with the reservoir T_2 and β the thermal conductance between

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the working substance T_{2w} and the thermal reservoir T_2, also, $T_1 > T_{1w} > T_{2w} > T_2$ and a reversible Carnot cycle between the intermediate temperatures T_{1w} and T_{2w}.

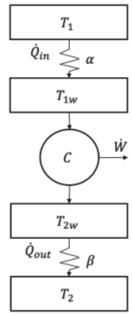

Figure 1: Curzon-Ahlborn heat engine

This heat engine is defined as a cycle in which heat transfer from thermal reservoirs is irreversible as it is considered time-dependent, while the remaining processes that are not related to heat transfer are reversible. Curzon and Ahlborn (1975) determined that the efficiency for this heat engine is given by

$$\eta_{CA} = 1 - \sqrt{\frac{T_2}{T_1}} = 1 - \sqrt{\tau}, \quad (1)$$

where $\tau = T_2/T_1$. The work carried out by Curzon and Ahlborn is considered the work of Finite Time Thermodynamics (FTT), since the internal irreversibilities of the heat engine are considered. This branch of physics (FTT) is of great importance for the design of real thermal machines since (), based on the different proposed thermal machines and their modes of operation, it has been possible to establish predictions in the efficiencies that a real thermal engine can deliver for thus generating energy. Velasco et al (2000) presented a new optimization criterion for the Curzon-Ahlborn (1975) heat engine model, considering that power plants operate following a thermal cycle in which a working substance absorbs a flow of heat $\dot{Q}_{in}$ from a heat source, rejects a heat flow $\dot{Q}_{out}$ to a heat sink at a lower temperature, to generate a power output $\dot{W}$, which can be written due to the conservation of energy as

$$\dot{Q}_{in} = \dot{W} + \dot{Q}_{out}, \quad (2)$$

These authors considered three interdependent processes are taken into account, a profitable type process that is power production and the other two processes of unwanted type, which are the input and output heat flows. This criterion is known as savings functions. These authors considered that an unwanted process can be written mathematically by a function $F(\{x\};\{\lambda\})$ where $\{x\}$ represents the set of independent variables and $\{\lambda\}$ represents the set of controllable parameters, so they defined the savings function associated with $F(\{x\};\{\lambda\})$ by:

$$f(\{x\};\{\lambda\}) = 1 - \frac{F(\{x\};\{\lambda\})}{F_{max}(\{\lambda\})}, \quad (3)$$

where $F_{max}(\{\lambda\})$ is the maximum value of $F(\{x\};\{\lambda\})$ for the allowed range of values of $\{x\}$, said maximum value corresponds to the most inefficient mode of operation of the system, so the savings function is less than as the system's operating regime becomes more inefficient. With this in mind, Velasco (2000) defended two savings functions; one associated with fuel consumption and therefore proportional to the input heat flow $\dot{Q}_{in}$ defined by

$$q_{in} = 1 - \frac{\dot{Q}_{in}}{(\dot{Q}_{in})_{max}}, \quad (4)$$

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where $(\dot{Q}_{in})_{max}$ is the maximum heat flow that can be extracted from the thermal reservoir at temperature T_1 without supplying power. The other associated savings function is associated with thermal pollution pollution and is therefore proportional to the output heat flux, defined by

$$q_{out} = 1 - \frac{\dot{Q}_{out}}{(\dot{Q}_{out})_{max}}, \quad (5)$$

Where $\dot{Q}_{out}$ is the heat output of the heat engine, $(\dot{Q}_{out})_{max} = (\dot{Q}_{in})_{max}$, is the maximum heat flow considered as a measure of the size of the power plant according to how it was proposed by De Vos (1992). In their study Velasco et al (2000) considered the following three processes: (1) the production of useful energy defined by $\omega = \dot{W}/\dot{W}_{mp} = (\dot{Q}_{in} - \dot{Q}_{out})/\dot{W}_{mp}$, where $\dot{W}$ represents the output power and $\dot{W}_{mp}$ represents the maximum output power of the engine; (2) savings in fuel consumption, characterized by the q_{in} function; and (3) the reduction in thermal pollution, characterized by the q_{out} function. With the simultaneous optimization of these three processes, the optimal operating mode of the heat engine is sought, that is, a function of the form $\Phi = (\omega, q_{in}, q_{out})$ is optimized. With the above considerations, Velasco (2000) proposed two objective functions, one of them defined in the linear formalism supported by the first law of thermodynamics, and the other objective function based on a power law formalism, in which the Energy production, fuel consumption and pollution reduction are optimized simultaneously. The objective functions are defined by

$$\Phi_A = a_1\omega + a_2 q_{in} + a_3 q_{out}, \quad (6)$$

$$\ln\Phi_B = b_1 \ln\omega + b_2 \ln q_{in} + b_3 \ln q_{out}, \quad (7)$$

where a_i and b_i (for $i = 1, 2,3$) are weight coefficients that measure the degree of participation of each process in the optimization criterion. Velasco (2000) considered the case in which the degree of participation in each process is done without discrimination, which is when $a_1 = a_2 = a_3$ and $b_1 = b_2 = b_3$. In 2021 Barranco-Jimenez (2021) studied the Novikov heat engine model using the savings functions with a Dulong-Petit heat transfer law (Sullivan 1990) through different heat transfer laws and considering different degrees of participation in the process, i.e. with different values in the weight coefficients $a_1 \neq a_2 \neq a_3$ and $b_1 \neq b_2 \neq b_3$. Furthermore, these authors analyzed the ratio of heat flows in the different operating modes of the Novikov heat engine to make a comparison in the reduction of thermal pollution. En este trabajo realizamos el análisis de la máquina térmica de Agrawal mediante las funciones de ahorro y considerando diferente grado de participación en los coeficientes de peso de manera similar a lo reprtado por Barranco-Jimenez (2021).

2 SIMPLIFIED VERSION OF A CURZON-AHLBORN HEAT ENGINE

For the representation of the CA heat engine showed in Figure 1, can be considered the following temperature differences $X = T_1 - T_{1w}$ and $Y = T_{2w} - T_2$. Applying the first law of thermodynamics and using a Newton-type heat transfer law for the CA heat engine, the heat fluxes $\dot{Q}_{in}$ and $\dot{Q}_{out}$ can be written as follows:

$$\dot{Q}_{in} = \alpha(T_1 - T_{1w}) = \alpha x, \quad (8)$$

$$\dot{Q}_{out} = \beta(T_{2w} - T_2) = \beta y. \quad (9)$$

In 2009, Agrawal (2009) proposed a simplified version of the CA heat engine, which is obtained by considering two conditions, one is to consider equal temperature differences, that is, $X = Y$ and the second condition is that the thermal conductances between the thermal reservoirs and the working substance are also the same $\alpha = \beta$. The so-called Agrawal heat engine is shown in Figure 2.

$$\dot{Q}_{in} = \alpha(T_1 - T_{1w}) = \alpha X, \quad (10)$$

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$$\dot{Q}_{out} = \beta(T_{2w} - T_2) = \alpha X. \tag{11}$$

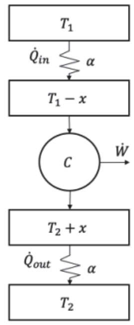

Figure 2: Agrawal heat engine

Applying the first law of thermodynamics, using a Newton-type heat transfer law for the CA engine and taking into account the previous conditions, the heat fluxes $\dot{Q}_{in}$ and $\dot{Q}_{out}$ for the Agrawal heat engine can be written as

$$\dot{Q}_{in} = \alpha(T_1 - T_{1w}) = \alpha X, \tag{10}$$

$$\dot{Q}_{out} = \beta(T_{2w} - T_2) = \alpha X. \tag{11}$$

Agrawal (2009) used an algebraic method known as the componendo and dividendo rule to obtain the efficiency of the CA engine, with the aim of making it easier for undergraduate students to understand. This author showed that the efficiency of the simplified CA heat engine is given by

$$\eta_A = 1 - \frac{T_1+3T_2}{3T_1+T_2} = 1 - \frac{1+3\tau}{3+\tau}. \tag{12}$$

3 HEAT TRANSFER LAWS

3.1 Newton heat transfer law

Heat transfer laws are used to model heat flows between thermal reservoirs and working substances, one of these laws is the Newton-type heat transfer law given by:

$$\frac{dQ}{dt} = \alpha(T_1 - T_2), \tag{13}$$

where T_1 and T_2 are the extreme temperatures during the heat exchange process, and Q the heat transferred, lost or gained by the system, at any instant of said heat exchange process, and alpha the thermal conductance that is assumed constant. However, this law is an idealization of the way in which heat transfer occurs, which depends on the materials and the medium between the system and the surroundings. According to our model we denote the temperatures of the hot and cold reservoirs, respectively, as T_1 and T_2, and the high and low temperatures of the working substance as T_{1w} and T_{2w}, respectively.

4 OPTIMIZATION OF THE SAVING FUNCTION WITH NEWTON HEAT TRANSFER LAW FOR THE AGRAWAL HEAT ENGINE

4.1 Linear Formalism considering the three process with the same degree of contribution

Following the approach by Velasco (2000), we consider the efficiency $\eta(\equiv \{x\})$ as the independent variable and $\tau(\equiv \{\lambda\})$ as the controllable parameter of the system. Applying the first law of thermodynamics and considering a Newton-type heat transfer law between the thermal reservoirs and

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the working substance for the Agrawal heat engine (Figure 2), we can write $\dot{Q}_{in} = \alpha X$ and considering that the power output, $\dot{W} = \eta \dot{Q}_{in}$, we obtain

$$\dot{Q}_{in} = A\left(\frac{1-\tau-\eta}{2-\eta}\right), \tag{14}$$

where $\tau = T_2/T_1$ and η the efficiency of the heat engine and A is an η-independent constant. It is possible to verify that for this engine $\dot{W}_{mp} = A\left(3+\tau-2\sqrt{2(1+\tau)}\right)$, and, $\left(\dot{Q}_{in}\right)_{max} = A\left(\frac{1-\tau}{2}\right)$ that is the maximum value of input heat flow, and considering that $\dot{Q}_{out} = (1-\eta)\dot{Q}_{in}$. The objective function based in the linear formalism (6) with $a_1 = a_2 = a_3 = a$ that is the arithmetic mean of the three performances, is

$$\Phi_A = \frac{a\eta\left(13-8\sqrt{2(1+\tau)}+\tau(2+\tau)+\eta\left(-7-\tau+4\sqrt{2(1+\tau)}\right)\right)}{(-2+\eta)(-1+\tau)\left(3+\tau-2\sqrt{2(1+\tau)}\right)}. \tag{15}$$

In Figure 3a we show the behavior of the objective function (15) against the internal efficiency of the heat engine for with $a = 1/3$ and different values of $\tau = 0.25, \tau = 0.5$ and $\tau = 0.75$, as can be seen in the plot for all values of τ, there is a maximum for η that is given by

$$\eta_{\phi_A} = \frac{-4\left(-7-\tau+4\sqrt{2(1+\tau)}\right)-2\sqrt{2}\sqrt{7+\tau-7\tau^2-\tau^3-4\sqrt{2}\sqrt{1+\tau}+4\sqrt{2}\tau^2\sqrt{1+\tau}}}{2\left(7+\tau-4\sqrt{2}\sqrt{1+\tau}\right)}. \tag{16}$$

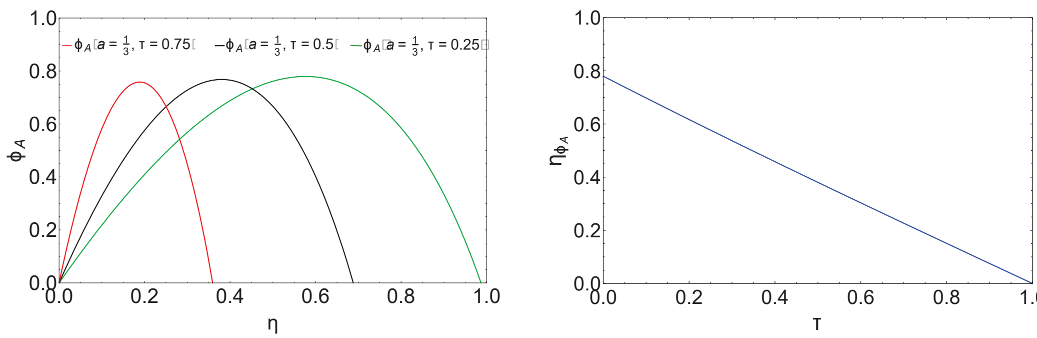

Figure 3a: Savings functions by linear formalism vs internal efficiency

Figure 3b: Optimal efficiency vs temperature ratio

4.2 Linear Formalism considering the three process with different degree of participation

Now we can consider that each of the three processes has a different degree of participation, that is, that the weight coefficients have different values, then we can write the objective function (6) when the weight coefficient a_1 is constant in the form

$$\phi_{A1} = \frac{\eta\left((-1+\tau)(-1+\eta+\tau)-\beta_1(1+\tau)\left(-3-\tau+2\sqrt{2(1+\tau)}\right)+\gamma_1(-3+2\eta+\tau)\left(-3-\tau+2\sqrt{2(1+\tau)}\right)\right)}{(-2+\eta)(-1+\eta)\left(3+\tau-2\sqrt{2(1+\tau)}\right)}, \tag{17}$$

where $\beta_1 = a_2/a_1$, $\gamma_1 = a_3/a_1$ and $\tau = T_2/T_1$. The plot of equation (16) for different values of β_1 and γ_1 is shown in Figure 5a.

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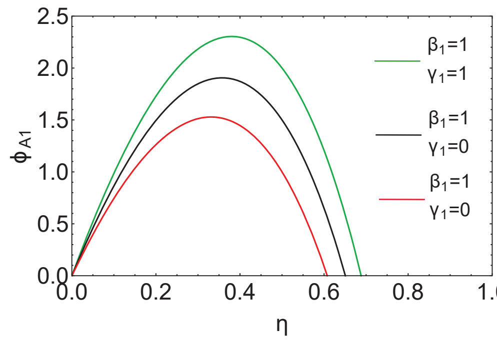

Figure 4a: Savings functions by linear formalism vs internal efficiency

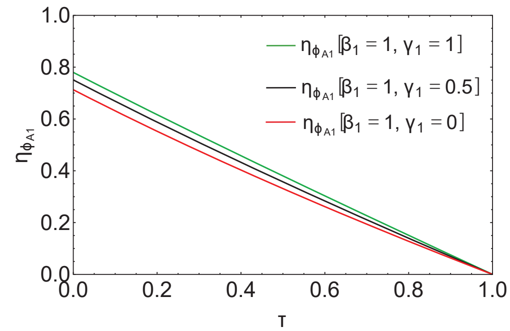

Figure 4b: Optimal efficiency vs temperature ratio

From Figure 4a it is possible to observe that the function has a maximum for η that we can determine by $(d\phi_{A1}/d\eta) = 0$ at,

$$\overline{\eta}_{\phi_{A1}} = \frac{\left(-2-12\gamma_1+2\tau-4\gamma_1+8\sqrt{2}\gamma_1\sqrt{1+\tau}+\sqrt{2}\sqrt{(1+\tau)\left(-1+\tau+2\sqrt{2}\gamma_1\sqrt{1+\tau}-\gamma_1(3+\tau)+\beta_1\left(3+\tau-2\sqrt{2(1+\tau)}\right)\right)\left(-1+\tau+\gamma_1\left(-6-2\tau+4\sqrt{2(1+\tau)}\right)\right)}\right)}{\left(-1+\tau+\gamma_1\left(-6-2\tau+4\sqrt{2(1+\tau)}\right)\right)},$$
(18)

In Figure 4b we show the behavior of the optimal efficiency (equation (18)) for the case in which the weight coefficient a_1 remains constant and taking some values for parameters β_1 and γ_1.
In an analogous way we can carry out the study when the weight coefficients a_2 and a_3 are constant. For the case where a_2 remains constant to the following expressions

$$\phi_{A2} = \frac{\eta\left(\alpha_1(-1+\tau)(-1+\eta+\tau)+\left(-3-\tau+2\sqrt{2(1+\tau)}\right)\left(-1-\tau+\gamma_1(-3+2\eta+\tau)\right)\right)}{(-2+\eta)(-1+\tau)\left(3+\tau-2\sqrt{2(1+\tau)}\right)}, \tag{19}$$

$$\overline{\eta}_{\phi_{A2}} = \frac{\left(2\alpha_1(-1+\tau)+8\sqrt{2}\gamma_1\sqrt{1+\tau}-4\gamma_1(3+\tau)+\sqrt{2}\sqrt{\left((1+\tau)\left(\alpha_1^2(-1+\tau)^2+\alpha_1(-1+3\gamma_1)(-1+\tau)\left(-3-\tau\sqrt{2(1+\tau)}\right)-2(-1+\gamma_1)\gamma_1\left(-17+12\sqrt{2(1+\tau)}\right)+\tau\left(-14-\tau+4\sqrt{2(1+\tau)}\right)\right)\right)}\right)}{\left(\alpha_1(-1+\tau)+4\sqrt{2}\gamma_1\sqrt{1+\tau}-2\gamma_1(3+\tau)\right)}. \tag{20}$$

And for the case that a_3 is constant, we have

$$\phi_{A3} = \frac{\eta\left(\alpha_1(-1+\tau)(-1+\eta+\tau)-(3+\beta_1-2\eta+(-1+\beta_1)\tau)\left(-3-\tau+2\sqrt{2(1+\tau)}\right)\right)}{(-2+\eta)(-1+\tau)\left(3+\tau-2\sqrt{2(1+\tau)}\right)}, \tag{21}$$

$$\overline{\eta}_{\phi_{A3}} = \frac{\left(-2(6+\alpha_1)+2(-2+\alpha_1)\tau+8\sqrt{2(1+\tau)}+\sqrt{2}\sqrt{(1+\tau)\left(\alpha_1^2(-1+\tau)^2-\alpha_1(-3+\beta_1)(-3-\tau)+2\sqrt{2(1+\tau)}+2(-1+\beta_1)\left(-17+12\sqrt{2(1+\tau)}+\tau\left(-14-\tau+4\sqrt{2(1+\tau)}\right)\right)\right)}\right)}{\left(-6+\alpha_1(-1+\tau)-2\tau+4\sqrt{2(1+\tau)}\right)}. \tag{22}$$

The behaviors of equations (19- 22) are similar to the behavior of equation (17) and equation (18), and such behavior can be seen in Figure 4a and Figure 4b. Therefore, we consider that it is not necessary to present the plots again.

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5 NUMERICAL WORK

In this section we present the efficiencies of some old real power plants and compare the values with those we obtain by using equation (16) and equation (18) as can be seen in Table 1.

Table 1: Efficiencies of old real power plants using $\eta_C, \eta_{CA}, \eta_A, \eta_{\phi_A}, \eta_{\phi_{A1}}$ and η_{OBS} (in %)

Power source	T_1/T_2	η_C	η_{CA}	η_A	η_{ϕ_A}	$\eta_{\phi_{A1}}$	η_{OBS}
1936-1940: Central vapor power stations in the UK	298/698	57.3	34.7	33.4	43.7	31.1	28
1944: Closed-cycle gas turbine in Switzerland	298/963	69.1	44.4	41.7	52.9	38.2	32
1950: Closed-cycle gas turbine in France	298/953	68.7	44.1	41.5	52.7	38	34
1956: Calder Hall Nuclear Reactor in the UK	298/583	48.9	28.5	27.8	37.1	26.1	19

We also present the efficiencies of modern real power plants to compare them with the efficiencies we obtain when using equation (16) and equation (18) as can be seen in Table 2.

Table 2: Efficiencies of modern real power plants using $\eta_C, \eta_{CA}, \eta_A, \eta_{\phi_A}, \eta_{\phi_{A1}}$ and η_{OBS} (in %)

Power source	T_1/T_2	η_C	η_{CA}	η_A	η_{ϕ_A}	$\eta_{\phi_{A1}}$	η_{OBS}
1983: Almaraz II (nuclear reactor with pressurized water) in Spain	290/600	51.7	30.5	29.7	39.3	34.3	34.5
1984: Cofrentes (nuclear reactor with boiling water) in Spain	289/562	48.6	28.3	27.6	36.9	32.1	34
1985: Doel 4 (nuclear reactor with with pressurized water) in Belgium	283/566	50.0	29.3	28.6	38	33.1	35
1988: Heysham (advanced nuclear reactor cooled with gas) in the UK	288/727	60.4	37.1	35.6	46.1	40.5	40
1955: Sizewell B (nuclear reactor with pressurized water) in the UK	288/581	50.4	29.6	28.9	38.4	33.4	36.3

Finally in Figure 5 we show the behavior of equation (16) and equation (18) and plot them together with the Carnot efficiencies defined as:

$$\eta_C = 1 - \frac{T_2}{T_2} = 1 - \tau \qquad (23)$$

The Curzon-Ahlborn efficiency defined by equation (1) and the Agrawal efficiency represented by the equation (12).

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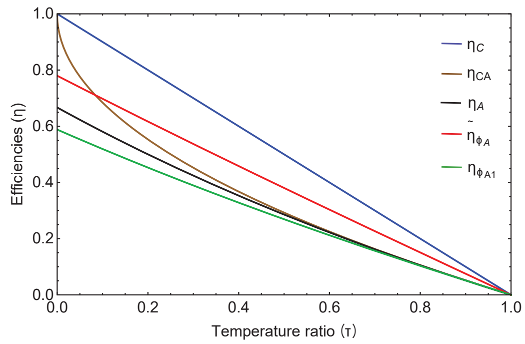

Figure 5: Optimal efficiencies (η) vs temperature ratio (τ)

6 CONCLUSIONS

In this article we carry out the analysis of a simplified Curzon-Ahlborn engine which we call the Agrawal engine, using the criterion of the savings functions proposed by Velasco using a transfer law Newton type heat. In our analysis, the optimal efficiency of the thermal engine was determined through a linear approach. We also considered two cases within our analysis: (1) in which the weight coefficients that represent the degree of participation of the processes of: energy production useful, fuel consumption and thermal pollution, each of them has the same degree of participation and (2) in which the weight coefficients of said processes have different values, that is, considering that the degrees of participation of the processes are different.

Our numerical results show that for case (2) where the value of the weight coefficients has a different degree of participation, it is true that: $\eta_C > \eta_{CA} > \eta_{\phi_{A1}} > \eta_{OBS}$ for the case of old real power plants and $\eta_C > \eta_{OBS} > \eta_{\phi_{A1}} > \eta_{CA} > \eta_A$ for the case modern real power plants. Furthermore, when different values are considered in the degrees of participation, we were able to observe that efficiencies decrease, which agrees with the work reported by Barranco-Jiménez (2021). An analysis can be done for different heat transfer laws as well as variation in the different degrees of participation of the processes involved, however we only consider the linear heat transfer law since the expressions obtained with laws do not linear lines are too extensive. In Figure 4b we show the behavior of the efficiencies obtained in this work and compare them with those reported in the literature.

NOMENCLATURE

a_i, b_i	weight coefficients
A	η-independent constant
F	value for the allowed range of values
q	saving function
$\dot{Q}$	rate of heat Flow
T	temperature of the heat reservoir, k
$\dot{W}$	output power
x	independent variables
X, Y	temperature differences

Greek symbols

$\boldsymbol{\alpha}, \boldsymbol{\beta}$	thermal conductances
β_i, γ_i	weight coefficient quotient
λ	controllable parameters
η	efficiency

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$\overline{\eta}$ optimal efficiency
τ temperature ratio
ω production of useful energy
Φ objective function

Subscript
A Agrawal
A Linear Formalism
$A1$ Linear Formalism considering the three process with different degree of participation
C Carnot
CA Curzon-Ahlborn
iw working substance
max maximum
mp maximum output power
OBS observed

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