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Efficiency of Energy Conversion Cycles in Finite Time Thermodynamics: The Case of Nuclear-Driven Carnot Cycle

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Abstract

This paper aims to clarify the conditions required to derive, within the framework of Finite Time Thermodynamics, the efficiency of "endoreversible cycles" (Eq. (1) of this paper). It is shown that the steady-state enthalpy and entropy balances of the working fluid at maximum power, the finite time thermodynamics optimum, lead to this efficiency. This result is obtained without introducing the "finite time" of the cycle, the most usual approach. To compute this efficiency for real systems, the heat source and sink temperatures must be clearly defined. In complex systems such as nuclear reactors, several choices of the upper temperature are possible. For a rational choice – using the steam generator as the heat source – these temperatures are relatively low, and the finite time thermodynamics optimum corresponds to unrealistic operating conditions, slightly underestimating actual efficiencies. Conversely, higher upper temperature values yield more realistic conditions but overestimate efficiency. The finite time thermodynamics limits are revealed by computing all thermodynamic cycle parameters and introducing irreversibilities in the working fluid. The cycle specific net work (MJ/kg) appears to be a more relevant optimisation criterion for real systems. This supports the idea that acceptable predictions from finite time thermodynamics efficiency result from compensating errors: low maximum temperatures balanced by the ideal cycle assumption. A general efficiency equation better illustrates why actual efficiency remains below Carnot's, clarifying optimisation issues in energy conversion systems and the trade-off between reducing irreversibilities and maximising average heat source temperature.

Keywords: Carnot cycle; Finite time thermodynamics; Power output optimum; Efficiency optimum

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1. Introduction

The origin of this work lies in recognition of extensive literature on "Finite Time Thermodynamics" (FTT) and "Endoreversibility", with the proposal of a straightforward equation, Eq. (1) hereafter, aimed to provide a more realistic estimation of the efficiency of energy conversion systems, particularly in comparison with the Carnot efficiency. This equation is:

$$\eta_t = 1 - \sqrt{\frac{T_d}{T_u}}, \tag{1}$$

with T_u being said to be the heat source temperature and T_d the heat sink temperature (both in Kelvin).

The main objective of this paper is to better understand the conditions and assumptions necessary to obtain this elegant equation.

Nomenclature

h – specific enthalpy, J/kg
$\dot{h}_d$ – heat sink heat transfer coefficient, W/K
$\dot{h}_u$ – heat source heat transfer coefficient, W/K
$\dot{m}$ – cycle mass flowrate, kg/s
$\dot{P}$ – cycle power output, W
P_i – pressures of the Carnot cycle, i = 1 to 4, Pa
$\dot{Q}_d$ – heat transfer with the surroundings for the heat sink, W
$\dot{Q}_u$ – heat transfer with the surroundings for the heat source, W
r – specific ideal gas constant, J/(kg·K)
s – specific entropy, J/(kg·K)
T_d – heat sink temperature, K
T_i – temperatures of the Carnot cycle, i = 1 to 4, K
T_u – heat source temperature, K
$\dot{W}_{ij}$ – work exchanged with the surroundings in the (i, j) process, W

Greek symbols

γ – ratio of specific heats at constant pressure and at constant volume
η – thermodynamic cycle efficiency
η_t – finite time thermodynamic optimum efficiency

Subscripts and Superscripts

d – heat sink parameters
u – heat source parameters

Abbreviations and Acronyms

FTT – finite time thermodynamics
PWR – pressurised water-cooled reactor

This equation is by some authors referred to as the Chambadal-Novikov-Curzon-Ahlborn efficiency to pay tribute to some of the pioneering authors. These pioneering authors, [1−3] and even [4], in establishing this equation, were mainly interested in energy conversion of nuclear power plants. Thus T_u is said to be the nuclear fuel cladding temperature by Yvon [1] or just the nuclear fuel temperature, Novikov [3]. Chambadal [2] mentions the nuclear primary circuit coolant temperatures. Curzon and Ahlborn [4] apply Eq. (1) to real systems by considering the mean temperature of the primary coolant. This approach appears to be the most rational within the framework of FTT with a well-defined heat exchanger for the cycle heat source (thus, the steam generator in nuclear reactors having two separate circuits for power conversion and reactor core cooling). This approach will therefore be adopted as the reference case in the present study.

In most papers, T_u is simply the heat source temperature. For a real and complex system, like nuclear power plants, it is not a definition precise enough, which could be a problem for a proper use of Eq. (1), see Section 2.2.

T_d poses much fewer definition issues. For a common steam cycle, it must be the condenser secondary side temperature, a temperature that usually varies very little. Here, we are close to having an infinite thermal capacity at constant temperature.

In some of these pioneering works, the derivation of Eq. (1) is obtained with the introduction of a cycle geometric mean temperature:

$$T_m = \sqrt{T_d\, T_u}\,.$$

Feidt [5], in his history of efficiency at maximum power, explains that such derivation was already achieved in a 19th century thermodynamic book. We will explain in Section 2.1 how to obtain this mean temperature.

Since these early publications, numerous FTT studies have been published, see for example [6−10]. FTT has evolved into a well-established branch of thermodynamics, with its inclusion in the French "Technique de l'ingénieur" [11], dedicated textbooks [12,13], and even its selection as a topic for the entrance examination of French elite engineering schools in 2023 [14].

Some of these works propose new discussions of the FTT approach. Others attempt to extend the use of FTT, for example, with the consideration of other energy conversion phenomena, like thermoelectricity, Ouerdane et al. [6], or with the introduction of irreversibilities, Wu and Kiang [7] or Feidt and Costea [8]. Le et al. [9] or Kaushik et al. [12] have shown that Eq. (1) could be useful for other cycles than the Carnot cycle originally considered. In this paper, only Carnot-like cycles will be examined.

It is worth noting from this abundant scientific literature that many of these studies share a common limitation: the absence of a clear definition of the thermodynamic system under consideration, which could explain difficulties encountered in properly defining T_u.

Our purpose is to revisit these analyses, to better understand the real interest of Eq. (1) and not at all to propose new optimisations of real system energy conversion cycles. Thus, in Section 2.1, we present a derivation of this equation without introducing a cycle time. In the following Section 2.2, some Eq. (1) numerical results will be compared to real cycle efficiencies.

In Section 2.3, we will examine further all the implications of the maximum power output by determining all the cycle variables in very simple cases (use of an ideal gas as the cycle working fluid).

In section 3, we will see that for a “Carnot cycle”, with internal irreversibilities, here non-isentropic compression and expansion (of course with irreversibilities, strictly speaking, it is no longer a Carnot cycle), we obtain a different result in terms of compromise between the maximum efficiency and maximum power output of the cycle.

Finally in Section 4, we will propose an “academic extension” of the Carnot efficiency to real cycles. The equation obtained is not of interest to calculate the efficiency of real systems, but it reveals the underlying causes, in fact well known, to be well below the Carnot efficiency.

It is important to note that this work is essentially academic in nature as it involves discussing a theory, the FTT, that is itself also essentially academic. The ambition of this paper is also pedagogical: to better understand the FTT theoretical background.

2. “Endoreversible” Carnot cycle calculation

Our objective is to revisit the derivation of Eq. (1) by a thermodynamic approach based on well-defined thermodynamic systems and without the introduction of a cycle “finite time” (for example, cycle time introduced in [4]). Being in steady-state, we thought there is no real need to introduce such a time.

The cycle calculated is a Carnot cycle (Fig. 1) with two heat exchangers for the heat source at T_u and the heat sink at T_d.

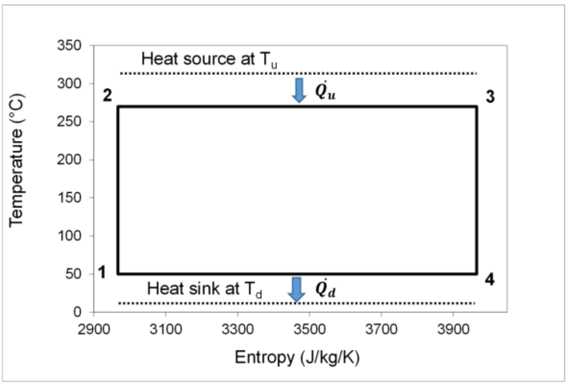

Fig. 1. The FTT Carnot cycle with its two heat exchangers.

The thermodynamic system considered is the working fluid having a mass flow rate $\dot{m}$ and describing the cycle formed by the four successive states 1 to 4 (Fig. 1), so it does not include the "materials" constituting the two heat exchangers. The cycle is characterised by four enthalpies (h_1 to h_4) and four entropies (s_1 to s_4).

As it is a Carnot cycle, we have:

- $s_2 = s_1$ and $s_4 = s_3$ (isentropic process), and
- $T_3 = T_2$ and $T_1 = T_4$ (isothermal heat addition and isothermal heat rejection).

To introduce the two heat exchangers, two heat transfer coefficients, $\dot{h}_u$ and $\dot{h}_d$, must be introduced to calculate the heat transfers, so the following simple equations are used:

– for the heat source

$$\dot{Q}_u = \dot{h}_u(T_u - T_2),$$

– for the heat sink

$$\dot{Q}_d = \dot{h}_d(T_d - T_1).$$

These heat transfer coefficients are assumed constant in the FTT approach, and in fact, they are the product of the actual heat transfer coefficient (in W/(m^2 K)) by the heat exchanger surface.

As a consequence, T_1 is the real low temperature seen by the working fluid of the cycle, and T_2 is the real high temperature of the cycle. To achieve such constant temperatures, with heat addition (or heat rejection), is only possible during a phase transition of the working fluid or by combining heat addition (or rejection) with the expansion (or compression) of a gas as the working fluid. For this simple thermodynamic cycle, the efficiency is, of course, given by the Carnot equation:

$$\eta_c = 1 - \frac{T_1}{T_2}.$$

The FTT purpose is to go further by expressing the efficiency with T_u and T_d, thus two temperatures not directly experienced by the thermodynamic system, but which may be of interest for the process evaluation. Usually, a thermodynamic analysis must use system temperatures, for example, the "exchanged entropy" $\delta Q/T$ must be calculated with *"the temperature of the part of the system receiving it"* (the Gibbs formulation). Therefore, in principle, using T_u and T_d would imply that the two heat exchangers (i.e., the materials responsible for heat conduction) are included in the thermodynamic system. Such a system will be considered in Section 4.

T_u and T_d are also assumed constant; it is often said that the two heat reservoirs are of infinite thermal capacity.

For a nuclear reactor like a PWR (pressurized water-cooled reactor), the very simple equation giving $\dot{Q}_u$ raises some questions, see Section 2.2. A rational assumption seems to take T_u as the primary coolant temperature, with several dozens of tons per second of water flow in the primary circuit, almost constant water temperatures in this circuit are obtained. In other real systems, such as coal power plants, much larger variations in the heat source primary side temperature are observed, several hundred degrees of Celsius, making the use of this hypothesis even more questionable.

2.1. Efficiency at the maximum power output

With the mass flow rate $\dot{m}$ of the working fluid used in this cycle, the system of ten equations to solve is made of the four energy balances:

$$\dot{m}(h_2 - h_1) = \dot{W}_{12},$$

$$\dot{m}(h_3 - h_2) = \dot{W}_{23} + \dot{h}_u(T_u - T_2),$$

$$\dot{m}(h_4 - h_3) = \dot{W}_{34},$$

$$\dot{m}(h_1 - h_4) = \dot{W}_{41} + \dot{h}_d(T_d - T_1),$$

$\dot{W}_{ij}$ being the work exchanged with the surroundings by the (i–j) process of the cycle.

In addition, we have the four entropy balances:

$$s_2 = s_1,$$

$$\dot{m}(s_3 - s_2) = \frac{\dot{h}_u(T_u - T_2)}{T_2},$$

$$s_4 = s_3,$$

$$\dot{m}(s_1 - s_4) = \frac{\dot{h}_d(T_d - T_1)}{T_1},$$

and finally:

$$T_3 = T_2,$$

$$T_4 = T_1.$$

For a working fluid with $h = h(P, T)$ and $s = s(P, T)$, and having chosen T_u, T_d, $\dot{h}_u$ and $\dot{h}_d$, we have these 10 equations to determine 13 unknowns:

– the four pressures and the four temperatures,
– the four works exchanged with the surroundings, $\dot{W}_{ij}$,
– and the cycle mass flowrate.

It is worth recalling that we are in steady-state, so all these unknowns are constant.

In the FTT approach, $T_u - T_2$ will be varied in order to find the maximum cycle power output. We will see just after that it is enough to fix $T_u - T_2$ to calculate the cycle efficiency and the cycle power output (specificity of the Carnot cycle). But to determine the 13 unknowns, two additional cycle parameters have to be fixed; we will see that there is an interest to determine all the cycle parameters, see Section 2.3.

The cycle power output $\dot{P}$ (a positive value is considered here) is given by the cycle energy balance (by summing up the 4 enthalpy equations):

$$\dot{P} = \left|\dot{W}_{12} + \dot{W}_{23} + \dot{W}_{34} + \dot{W}_{41}\right| = \dot{Q}_u + \dot{Q}_d\,,$$

with $x = T_u - T_2$ and $y = T_d - T_1$.

We have:

$$\dot{P} = \dot{h}_u x + \dot{h}_d y.$$

And the cycle entropy balance (by summing up the 4 entropy equations) gives:

$$\frac{\dot{h}_u(T_u - T_2)}{T_2} + \frac{\dot{h}_d(T_d - T_1)}{T_1} = \frac{\dot{h}_u x}{T_u - x} + \frac{\dot{h}_d y}{T_d - y} = 0.$$

It could be noted that these last two equations represent the global cycle energy and entropy balance. Therefore, it could be argued that the initial system of 13 equations was indeed not useful. In fact, this system is necessary for the complete determination of the cycle variables, see Section 2.3.

One can calculate:

$$y = \frac{T_d \dot{h}_u x}{x(\dot{h}_u + \dot{h}_d) - T_u \dot{h}_d}.$$

It is now possible to calculate the power output and the cycle efficiency as a function of x only:

$$\dot{P} = \dot{h}_u x\left(1 - \frac{T_d}{T_u - x(\dot{h}_u + \dot{h}_d)/\dot{h}_d}\right), \qquad (2)$$

$$\eta = \frac{\dot{P}}{\dot{h}_u x} = 1 - \frac{T_d}{T_u - x(\dot{h}_u + \dot{h}_d)/\dot{h}_d}. \qquad (3)$$

This Eq. (3) is a continuously decreasing hyperbola as a function of x. For non-negative values of x, the maximum efficiency is obtained for $x = 0$, this is the Carnot efficiency of a cycle producing no work, no heat being transferred to the cycle (because $x = 0$). We determine x_{max}, giving the maximum power, so the FTT optimum is found by solving $\frac{d\dot{P}}{dx} = 0$:

$$x_{max} = \frac{T_u \dot{h}_d}{\dot{h}_u + \dot{h}_d}\left(1 - \sqrt{\frac{T_d}{T_u}}\right).$$

And finally,

$$\eta(x_{max}) = 1 - \sqrt{\frac{T_d}{T_u}} = \eta_t.$$

The attractiveness of this equation lies in a very simple expression depending only on T_u and T_d and not on $\dot{h}_u$ and $\dot{h}_d$, making it independent of the two heat exchanger characteristics. It is worth noting that in the previous calculations, no assumptions have been made regarding the Carnot cycle's working fluid.

At this stage of our demonstration, one can already observe that if $\dot{h}_u$ and $\dot{h}_d$ are of the same order of magnitude, an apparently reasonable assumption for nuclear PWR but which could be discussed (see Section 2.2), and for typical T_u and T_d PWR temperatures, roughly 600 K and 300 K, one obtains x_{max} of about 90 K and y_{max} of about 60 K. These values are quite high compared to those of PWR systems, which usually operate with lower temperature differences, about 30 K for x. It is worth recalling that our choice of T_u means that we have considered the PWR steam generator as the cycle heat source heat exchanger, thus T_u is the primary coolant mean temperature. As already mentioned, a well identified heat exchanger seems the most suitable choice in the frame of FTT.

Returning to the works of the FTT pioneering authors, one of the main difference of some of these works, compared to the previous derivation, is the assumption of a low temperature of the Carnot cycle (T_1) equal to the heat sink temperature (T_d). This implies very large $\dot{h}_d$ values, so a value much larger than the $\dot{h}_u$ value. This could be named the "Chambadal-Novikov hypothesis" [2,3]. This assumption is almost opposite to our reference PWR case, where $\dot{h}_u$ and $\dot{h}_d$ are similar, however it is a justifiable assumption (see Section 2.2 below), which gives:

$$x_{max} = T_u - T_2 = T_u\left(1 - \sqrt{\frac{T_d}{T_u}}\right).$$

So, T_2 at maximum power output is:

$$T_2 = \sqrt{T_d T_u}\,.$$

And finally,

$$\eta = 1 - \frac{T_d}{\sqrt{T_d T_u}} = 1 - \sqrt{\frac{T_d}{T_u}} = \eta_t.$$

Another way to determine the maximum power output is to calculate $\dot{P}$ as a function of η instead of x. From Eq. (3),

$$x = \frac{\dot{h}_d}{\dot{h}_u + \dot{h}_d}\left(T_u - \frac{T_d}{1 - \eta}\right).$$

Introducing this value into Eq. (2) gives:

$$\dot{P} = \frac{\dot{h}_u \dot{h}_d}{\dot{h}_u + \dot{h}_d}\left(T_u - \frac{T_d}{1 - \eta}\right)\eta. \qquad (4)$$

We see that $\dot{P}$ is proportional to $\dot{h}_u \dot{h}_d/(\dot{h}_u + \dot{h}_d)$, which is the overall heat transfer coefficient of the heat source and heat sink "heat exchangers" in series. This overall heat transfer coefficient has a clear meaning for efficiencies close to zero, where we have:

$$\dot{P} = \frac{\dot{h}_u \dot{h}_d}{\dot{h}_u + \dot{h}_d}(T_u - T_d)\,\eta.$$

We confirm that $\dot{P}$ (Eq. (4)) is equal to zero for the maximum efficiency (which is a "Carnot efficiency"):

$$\eta_c = 1 - \frac{T_d}{T_u}.$$

The determination of the maximum power output by $\frac{d\dot{P}}{d\eta} = 0$ gives:

$$(1-\eta)^2 = \frac{T_d}{T_u},$$

leading to η_t of Eq. (1).

We can easily calculate the maximum power output:

$$\dot{P}_{max} = \frac{\dot{h}_u \dot{h}_d}{\dot{h}_u + \dot{h}_d}\left(\sqrt{T_u} - \sqrt{T_d}\right)^2.$$

The typical variation of $\dot{P}$ (and $\dot{Q}_u$) as a function of η (Eq. (4)) is shown in Fig. 2.

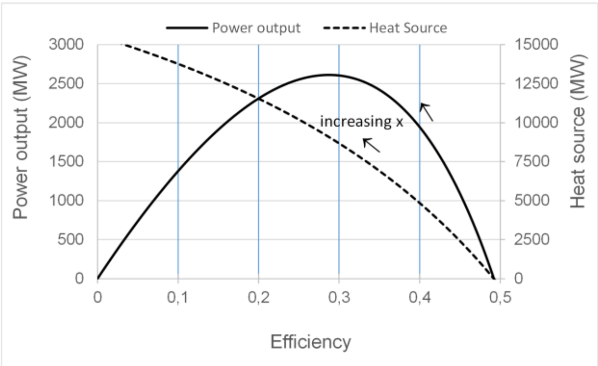

Fig. 2. Power output and cycle heat source as a function of efficiency for $x = T_u - T_2$ variation.

We will see (in Section 3) that such a variation is not observed in a real system. In fact, the previous equations cannot well explain the "trade-off" to be made between efficiency and power output.

2.2. Discussion of the applicability of Eq. (1)

Like Curzon and Ahlborn in [4], we try to see whether Eq. (1) offers good approximations of real system efficiency. For nuclear power plants, assuming $\dot{Q}_u = \dot{h}_u(T_u - T_2)$, the issue is not only whether T_u remains constant, but also the system's ability to handle the large variation of $\dot{Q}_u$. This large variation of $\dot{Q}_u$ behind the FTT approach (see Fig. 2) seems difficult to achieve by the same system. In a PWR reactor, the core's thermal design imposes specific core material temperatures, and therefore, the core power is fixed (by heat conduction within the core materials). The following simple equations can roughly describe the series of heat transfers between the core materials and the thermodynamic cycle:

$$\dot{Q}_u = \dot{h}'(T_{fuel} - T') = \dot{h}''(T' - T_2).$$

So,

$$\dot{Q}_u = \frac{\dot{h}'\dot{h}''}{\dot{h}' + \dot{h}''}\left(T_{fuel} - T_2\right),$$

T_{fuel} being representative of the fuel temperature (it could be the mean temperature of the UO_2 pellets), T' is the mean primary circuit temperature, and $\dot{h}'$, $\dot{h}''$ are the equivalent heat transfer coefficients of the core and steam generator, respectively. They are roughly constant for a given core and steam generator size. Both are a combination of conduction and convection heat transfers, but conduction is the dominant phenomenon.

When we assume $T_u = T'$ (said in the introduction to be a rational assumption), it can be seen that the variation in $\dot{Q}_u$ cannot be obtained by varying T_{fuel}; this variation requires changing the number of nuclear fuel elements (each fuel element producing a constant power). In this case, the FTT approach is justified by the constant value of $\dot{h}'$ (which corresponds to $\dot{h}_u$ here).

On the other hand, if we choose the fuel temperature for T_u, we have:

$$\dot{h}_u = \frac{\dot{h}'\dot{h}''}{\dot{h}' + \dot{h}''}.$$

And to be consistent with the FTT approach (constant $\dot{h}_u$ value), we cannot vary the heat source power by changing the number of nuclear fuel elements. Thus, it is the T' variation that will cause the variation in $\dot{Q}_u$. The large variation of the latter will lead to unusual values of the primary coolant temperature, with:

$$T' = \frac{\dot{h}' T_{fuel} + \dot{h}'' T_2}{\dot{h}'\dot{h}''}.$$

In this case, we can consider that the variations of $\dot{Q}_u$ occur without change in the "heat source system".

Our reference utilisation of Eq. (1) will consider the reactor primary coolant temperatures as the heat source temperature T_u. We have already noted that this temperature has small variations (lower than 40°C for a PWR). For the condenser secondary side (to determine T_d), temperature variations are even smaller (about 10°C). In both cases, the arithmetic mean of these temperatures will be used in Eq. (1).

2.2.1. Case of a 900 MWe PWR

The primary coolant mean temperature is about 304°C [15]. For the condenser secondary side (considering a sea site with direct cooling), the mean temperature is about 20°C. The "gross efficiency", so the ratio of the gross electrical generation over the heat transferred by the steam generators, is about 34% (see [15] or the IAEA Database on Nuclear Power Reactors). Taking into account the generator efficiency, the real thermodynamic cycle efficiency is about 35%.

By using Eq. (1), we obtain approximately 29%, which slightly underestimates the actual value. Of course, with these temperatures (T_u and T_d), the Carnot efficiency is equal to 49%, so a far too optimistic value. But the correct use of the Carnot efficiency requires the use of working fluid temperatures (so T_1 and T_2), thus the steam generator secondary side temperature (typically about 273°C) and the condensing temperature on the condenser's primary side (about 33°C) result in a Carnot efficiency of 44 %. With this latest result, it could be argued that the FTT efficiency provides a better approximation of the real efficiency.

Using these PWR data, we have also estimated the two FTT "heat transfer coefficients". We have: $\dot{h}_u$ = 90 MW/K (with a heat source of 2800 MW for a temperature difference of 304°C–273°C), and $\dot{h}_d$ = 140 MW/K (with a heat sink of 1820 MW for a temperature difference of 33°C–20°C). We recall that these latter values are calculated for a primary coolant heat source temperature, this heat source being about 2800 MW.

These data are used in the following numerical applications, presented in Section 2.2.3.

2.2.2. Case of a French sodium cooled reactor

For this type of reactor, the mean sodium temperature of the steam generator primary side is about 420°C. For a heat sink with direct condenser cooling, the temperature to be considered is again 20°C. The thermodynamic efficiency of this cycle is about 42%, while 35% is obtained by Eq. (1).

As in the previous PWR case, the FTT efficiency tends to give a lower value (around 20%) than the actual one. However, in both cases, we can agree that it provides a conservative estimate of the plant's efficiency.

2.2.3. Numerical applications for various heat source choices

We propose numerical applications of Eq. (2) and Eq. (3) with $\dot{h}_u$ and $\dot{h}_d$ values estimated in Section 2.2.1. As expected with this heat transfer coefficients choice, for typical reactor power outputs (about 1000 MW) the T_u–T_2 value is about 30 K (black dot in Fig. 3) with a much higher efficiency and a much lower power output compared to the FTT optimum, which is obtained for a T_u–T_2 value of about 100 K (Fig. 3). For this low T_2 temperature (about 200°C) an acceptable efficiency (0.29) is obtained because this low temperature is used in an ideal thermodynamic cycle: the Carnot cycle.

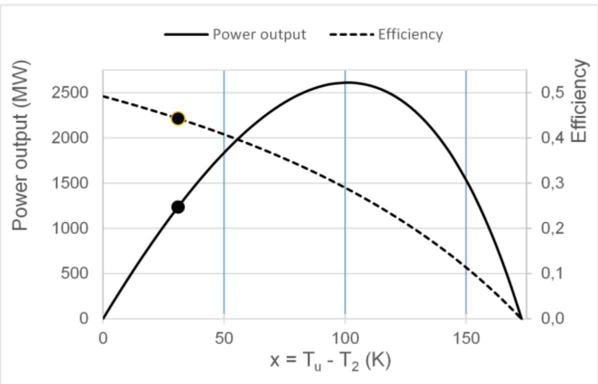

Fig. 3. Cycle power output and efficiency as a function of $x = T_u - T_2$ (here: T_u = 577 K, $\dot{h}_u$ = 90 MW/K, T_d = 293 K and $\dot{h}_d$ = 140 MW/K).

The FTT optimum power is about 2600 MW. For realistic use of this optimum, it would be necessary to reduce $\dot{h}_u$ and $\dot{h}_d$ by a factor of three to align with the typical core power. This would mean reducing the heat exchange surfaces of the steam generator and condenser. We are not aware of any commercial nuclear reactor operating under these conditions (i.e. smaller heat exchangers with larger temperature differences).

To be truly comprehensive on the FTT approach, we examine other justifiable choices of the nuclear reactor heat source temperature T_u, in fact to be closer to the nuclear fuel temperatures, see the beginning of Section 2.2. Two T_u choices are proposed: the fuel cladding temperature (in contact with the primary coolant with a temperature close to the saturation temperature of the primary circuit) and the mean fuel pellet temperature (temperature values from reference [15]). The new $\dot{h}_u$ values are calculated like in Section 2.2.1, of course, the T_u increase leads to a $\dot{h}_u$ decrease.

In both cases, the variation of T_2 (or x), thus of $\dot{Q}_u$, results in a change in the primary coolant temperature (see comments at the beginning of Section 2.2). The FTT optimum of these different assumptions are presented in Table 1.

Table 1. FTT optimal values under different heat source hypotheses, defined by (T_u, $\dot{h}_u$).

Heat source	PWR reference	Fuel cladding	Mean fuel pellet	Coal plant
T_u (°C)	304	345	715	1000
$\dot{h}_u$ (MW/K)	90	39	6.3	4
x (°C)	101	151	431	644
$\dot{P}$ (MW)	2611	1828	1235	1340
Efficiency	0.29	0.31	0.455	0.52
T_2 (°C)	203	194	284	356
$\sqrt{T_d T_u}$ (°C)	138	153	265	338

It can be observed that, as T_u increases, x and the power outputs move towards more realistic values. On the contrary, as expected with this increase, Eq. (1) now overestimates the actual efficiency. With the $\dot{h}_u$ decrease, the use of the "Chambadal-Novikov hypothesis" is more and more justified, thus the "Chambadal-Novikov temperature" ($\sqrt{T_d T_u}$) becomes an interesting approximation.

To discuss further this issue, a coal-fired power plant is examined. Here, we have a "simpler" heat source at much higher temperatures. We observe a temperature decrease from 1700°C to 300°C for the coal combustion products to produce subcritical steam at approximately 500°C and 100 to 150 bar [16]. Due to the poor heat transfer with the combustion products, the $\dot{h}_u$ value is low. To be consistent with previous calculations, we will continue to determine T_u with a mean temperature and $\dot{h}_u$ like in Section 2.2.1 (for a boiler secondary side temperature of 310°C). The FTT optimum values (Table 1) are perfectly consistent with the previous PWR "mean fuel pellet" case. We have a reasonable FTT value for T_2 (steam boiler being at saturation temperatures between 310°C and 340°C), a good "Chambadal-Novikov temperature" but an efficiency that is much too high (the actual cycle efficiency being about 35%). This latter result is in disagreement with [4]. We believe that Curzon and Ahlborn did not use the correct heat source temperature T_u for their coal power plant case, as it appears that Eq. (1) was applied using the steam temperature at the boiler outlet.

To conclude this section, it is worth noting again the relevance of the temperatures obtained when the "Chambadal-Novikov" hypothesis can be made (thus when $\dot{h}_d >> \dot{h}_u$). This also concerns the maximum power output equation:

$$\dot{P}_{max} = \dot{h}_u\left(\sqrt{T_u} - \sqrt{T_d}\right)^2.$$

2.3. Complete determination of the cycle variables

To go further and gain a better understanding of the practical implications of the previous calculations, we propose to calculate all the cycle variables, in particular the cycle mass flowrate. To achieve this, it is necessary to choose two additional cycle parameters (see the description of the system of 10 equations and 13 unknowns in Section 2.1).

To simplify significantly the computations, the working fluid is assumed now to be an ideal gas with an entropy variation for an (*a*-*b*) process given by:

$$s_b - s_a = r\left(\ln\left(\frac{T_b}{T_a}\right)^{\frac{\gamma}{\gamma-1}} - \ln\left(\frac{P_b}{P_a}\right)\right),$$

where r being the specific ideal gas constant (in J/kg/K), and γ is the ratio of specific heats at constant pressure and at constant volume.

Of course, such a working fluid is far from the steam/water used in PWR or conventional power plants cycles, but our objective here is only to explore all the possible implications of the FTT analysis, not to get new insights in the optimisation of these energy conversion systems.

For the two additional cycle parameters required, we propose to set the maximum and minimum pressures of the cycle, so P_2 and P_4, the two remaining cycle pressures are then:

$$P_1 = P_2\left(\frac{T_1}{T_2}\right)^{\frac{\gamma}{\gamma-1}},$$

$$P_3 = P_4\left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}}.$$

From Section 2.1 computations, we know that the cycle entropy balance can be used to determine T_1 as a function of T_2. With:

$$\dot{m}(s_3 - s_2) = \frac{\dot{h}_u(T_u - T_2)}{T_2},$$

we have:

$$\dot{m} = \frac{\dot{h}_u\, x}{rT_2 \ln\left(\frac{P_2}{P_3}\right)}.$$

So,

$$\dot{m} = \frac{\dot{h}_u x}{r(T_u - x)\left(\ln\left(\frac{P_2}{P_4}\right) + \frac{\gamma}{\gamma-1}\ln\left(\frac{\dot{h}_d T_d}{\dot{h}_d T_u - x(\dot{h}_u + \dot{h}_d)}\right)\right)}.$$

As seen in Fig. 4, for our "reference PWR case" (with the parameters from Fig. 3), the cycle mass flowrate is almost proportional to x. Therefore, a significantly higher mass flow rate would be required at the point of maximum power output (16 200 kg/s in this case) compared to the high-efficiency range.

Even with a reduction by a factor of three in the heat transfer coefficients (see Section 2.2.3), a very high flow rate is still obtained. These high flowrates are, of course, also a consequence of the choice of a simple air Carnot cycle. Additionally, such variations in flow rate seem difficult to achieve without changes in the heat transfer coefficients (assumed constant in the FTT approach) and in the pressure losses within the heat exchangers, which are also neglected in the FTT approach. For higher T_u assumptions, lower mass flow rates are observed at the FTT optimum, decreasing by a factor of three for the 'mean fuel pellet' case.

Now a very important cycle parameter can be computed, the cycle specific net work $\dot{P}/\dot{m}$ (in J/kg), which is equal to:

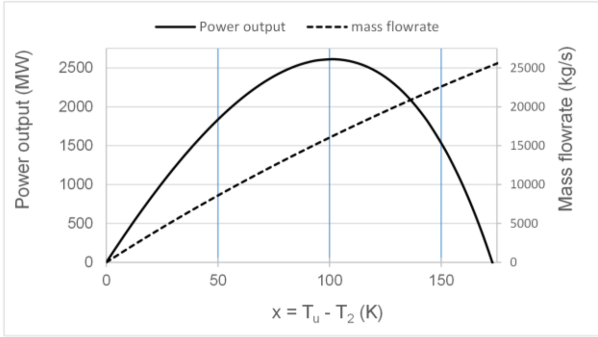

Fig. 4. Variations of the power output and of the mass flowrate (reference PWR case, air working fluid, P_2 = 200 bar and P_4 = 1 bar).

$$\frac{\dot{P}}{\dot{m}} = r(T_u - x)\left(1 - \frac{T_d \dot{h}_d}{T_u \dot{h}_d - x(\dot{h}_u + \dot{h}_d)}\right) \times$$
$$\times \left(\ln\left(\frac{P_2}{P_4}\right) + \frac{\gamma}{\gamma-1}\ln\left(\frac{T_d \dot{h}_d}{T_u \dot{h}_d - x(\dot{h}_u + \dot{h}_d)}\right)\right).$$

In fact, high specific net works mean lower cycle mass flow rates for the same power output. For our "reference PWR case", the specific net work is maximum when x is close to zero, thus when the efficiency is maximum, see Fig. 5. This could justify that PWR systems operate well below the FTT optimum of this reference case.

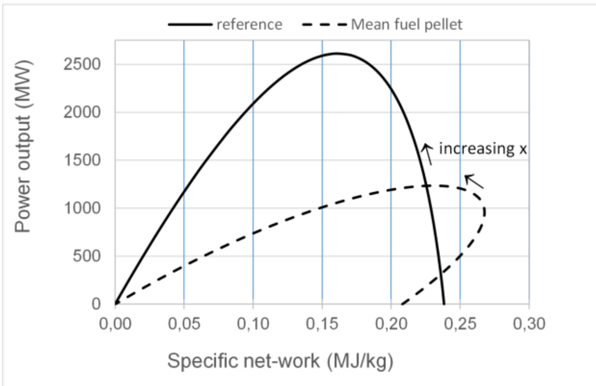

Fig. 5. Power output as a function of the specific net work for $x = T_u - T_2$ variation.

For our "reference PWR case", the specific net work is maximised when x is close to zero (both the power output and the cycle mass flowrate tend to zero), which corresponds to the maximum efficiency, see Fig. 5. For higher T_u assumptions, we have a different behaviour with a well identified maximum value of the specific net work, this value appearing much closer to the FTT optimum power.

To better understand these differences, we attempt to compute the maximum value of the specific net work. This can be achieved by making use of "the Chambadal-Novikov hypothesis" ($\dot{h}_d >> \dot{h}_u$), which gives:

$$\frac{\dot{P}}{\dot{m}} = r(T_u - x)\left(1 - \frac{T_d}{T_u - x}\right)\left(\ln\left(\frac{P_2}{P_4}\right) + \frac{\gamma}{\gamma-1}\ln\left(\frac{T_d}{T_u - x}\right)\right).$$

The zero of the derivative of this function leads to:

$$\frac{T_d}{T_u-x} + \ln\left(\frac{T_d}{T_u-x}\right) = 1 - \frac{\gamma-1}{\gamma}\ln\left(\frac{P_2}{P_4}\right).$$

Introducing the Lambert W function gives:

$$\frac{T_d}{T_u-x} = W\left(e^{1-\frac{\gamma-1}{\gamma}\ln\left(\frac{P_2}{P_4}\right)}\right).$$

So,

$$x = \frac{T_u W - T_d}{W}.$$

Using the previously mentioned air characteristics, we have:

$$x = T_u - \frac{T_d}{0.4}.$$

We can see that for high T_u cases, positive x values are obtained at the specific net work maximum, which explains the dashed curve in Fig. 5.

Finally, regardless of the assumption made for T_u, these results confirm that real systems operate close to the maximum of the specific net work, which is an effective way to limit the size of components for a given power output.

To conclude this section, in which we have seen that different values of T_u can be assumed for a complex system like a PWR, we can state the following:

- for "low T_u and high $\dot{h}_u$" values, our reference case (the most rational choice for a PWR with a well-identified heat source exchanger), the operating conditions at the FTT optimum are not realistic, while the FTT efficiency provides a conservative estimation of the real efficiency,
- for "high T_u and low $\dot{h}_u$" values, the operating conditions at the FTT optimum are more realistic, but the FTT efficiency overestimates the real efficiency like the efficiency given by the Carnot equation.

It is worth noting that this conservative FTT efficiency for the "low T_u" case is obtained with low maximum cycle temperatures (much lower than the actual ones) balanced by an ideal thermodynamic cycle, the Carnot cycle, without any losses. All these results clearly question and challenge the universal validity of Eq. (1).

3. More realistic relationship between power output and efficiency

We believe that one of the drawbacks of the previous approach is that it does not adequately illustrate the trade-off between maximum power output and maximum efficiency. To address this better, cycle irreversibilities must be introduced. Therefore, we now assume that the compression process (1–2) and the expansion process (3–4) are not isentropic. The cycle is no more an endoreversible cycle.

This was the objective of reference [7] or [8] with the introduction of a constant ratio:

$$R_s = \frac{s_3-s_2}{s_4-s_1} < 1.$$

Here $(s_4 - s_1) - (s_3 - s_2)$ is the increase of entropy (strictly positive) resulting from the irreversible compression (1−2) and from the irreversible expansion (3−4).

The update of the calculations from Section 2.1 is straightforward, involving an update of the global entropy balance:

$$\frac{\dot{h}_u(T_u-T_2)}{T_2} = R_s\,\frac{\dot{h}_d(T_d-T_1)}{T_1}.$$

So:

$$y = \frac{T_d\dot{h}_u x}{x(\dot{h}_u+R_s\dot{h}_d)-R_s\dot{h}_d T_u}.$$

Thus:

$$\dot{P} = \dot{h}_u x\left(1-\frac{T_d\dot{h}_d}{T_u R_s\dot{h}_d - x(\dot{h}_u+R_s\dot{h}_d)}\right),$$

and

$$\eta = 1-\frac{T_d\dot{h}_d}{T_u R_s\dot{h}_d - x(\dot{h}_u+R_s\dot{h}_d)}.$$

This latter equation gives:

$$x = \frac{\dot{h}_d}{\dot{h}_u+R_s\dot{h}_d}\left(R_s T_u - \frac{T_d}{1-\eta}\right),$$

and

$$\dot{P} = \frac{\dot{h}_u\dot{h}_d}{\dot{h}_u+R_s\dot{h}_d}\,\eta\left(R_s T_u - \frac{T_d}{1-\eta}\right).$$

The derivation of the previous equation gives the efficiency at the maximum power output:

$$\eta = 1-\sqrt{\frac{T_d}{R_s T_u}}. \tag{5}$$

This efficiency is lower than the efficiency given by Eq. (1) and remains independent of $\dot{h}_u$ and $\dot{h}_d$. Increasing irreversibilities, thus decreasing R_s, reduces efficiency, which becomes zero for:

$$R_s = \frac{T_d}{T_u}.$$

The relationship between power output and efficiency obtained is shown in Fig. 6.

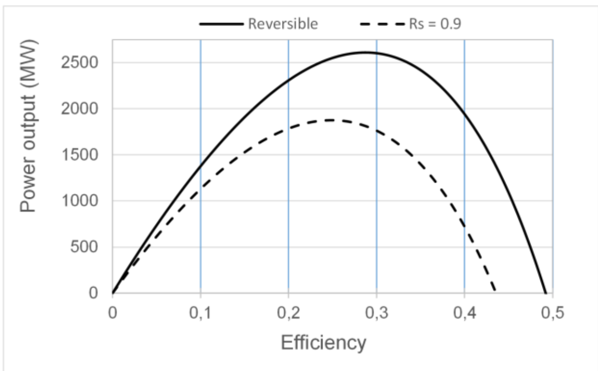

Fig. 6. Power output as a function of cycle efficiency, with and without irreversibilities (reference PWR case).

The curve is very similar to the reversible case, with maximum efficiency occurring at zero power output. We believe it still does not properly represent the power – efficiency relationship.

Our viewpoint is that R_s being constant, the entropy production by irreversibilities does not depend of the pressure ratio of the (1−2) compression and the (3−4) expansion, making this analysis of the "irreversible Carnot cycle" irrelevant. So, in accordance with Feidt and Costea [8], we propose assuming that these irreversibilities will generate an entropy increase equal to:

$$\dot{m}(s_2 - s_1) = \alpha\,(T_2 - T_1)$$

and

$$\dot{m}(s_4 - s_3) = \alpha\,(T_3 - T_4).$$

To simplify the calculations, α is assumed to be a constant positive value, identical for the compression and expansion. The cycle entropy balance is now:

$$\frac{\dot{h}_u x}{T_u - x} + \frac{\dot{h}_d y}{T_d - y} + 2\alpha(T_u - x - T_d + y) = 0.$$

This is a second-order equation with respect to y, so a more complicated equation than the cycle entropy equation of Section 2.1. Solving it gives:

$$y = T_d + \frac{1}{2}\left(B - \sqrt{B^2 + \frac{2T_d\dot{h}_d}{\alpha}}\right),$$

with,

$$B = \frac{\dot{h}_u x + (T_u - x)[2\alpha(T_u - x) - \dot{h}_d]}{2\alpha(T_u - x)}.$$

Finally, we have:

$$\dot{P} = \dot{h}_u x + \dot{h}_d\left(T_d + \frac{1}{2}\left(B - \sqrt{B^2 + \frac{2T_d\dot{h}_d}{\alpha}}\right)\right), \qquad (6a)$$

and

$$\eta = 1 + \frac{\dot{h}_d}{\dot{h}_u x}\left(T_d + \frac{1}{2}\left(B - \sqrt{B^2 + \frac{2T_d\dot{h}_d}{\alpha}}\right)\right). \qquad (6b)$$

The variations of power output and efficiency as a function of x (Fig. 7) show that this time, we have two distinct maximum values of interest: one for efficiency and one for power output.

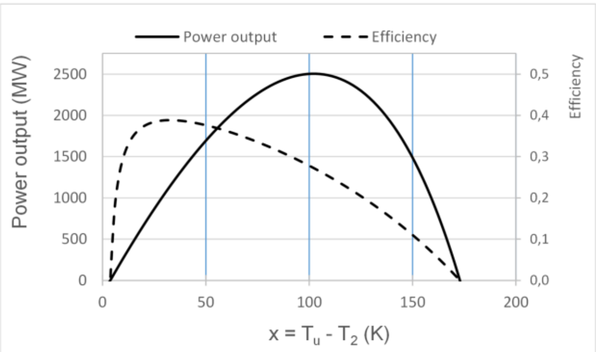

Fig. 7: Power output and efficiency as a function of $x = T_u - T_2$ (reference PWR case and α = 1000 W/(K m²)).

The value of α used (1000 W/(K m²)) is somewhat arbitrary, although it could give entropic losses of the correct order of magnitude (about 0.5 MW/K for PWR conditions).

For very low x, the thermodynamic cycle is not feasible, at higher x values, very low power outputs are achieved, and the heat generated by irreversible compression and expansion is removed by the heat sink. The maximum efficiency is now obtained for non-zero values of x. For our reference case, these x values (about 30 K, Fig. 7) are close to those of real systems (thanks to the choice of α). The x value at the FTT optimum is still very high (~100 K) and is accompanied by a significant reduction in efficiency, compared to the maximum value.

For higher T_u cases, the optimum efficiency and power output are obtained at large x values (310 K and 450 K).

The power output as a function of η (Fig. 8) now differs a lot from the variations obtained without irreversibilities (Fig. 2) or with a questionable introduction of irreversibilities (Fig. 6).

The marker (black dot) in Fig. 8 indicates the point corresponding to the actual x values of a PWR reactor, 31 K for our reference case, 442 K for the "mean fuel pellet" case.

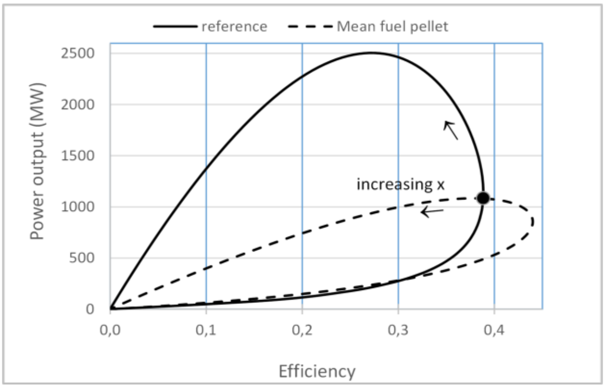

Fig. 8. Power output as a function of efficiency (α = 1000 W/(K m²)).

For the former, this point corresponds to the maximum efficiency, for the latter to the maximum power output. Thus, as without irreversibilities, the "high T_u" case leads to a more realistic FTT optimum. From the variation of the cycle specific net work with respect to efficiency (Fig. 9), it is observed that, in both cases, the actual PWR temperatures correspond to a point (black dot) very close to the maximum net work.

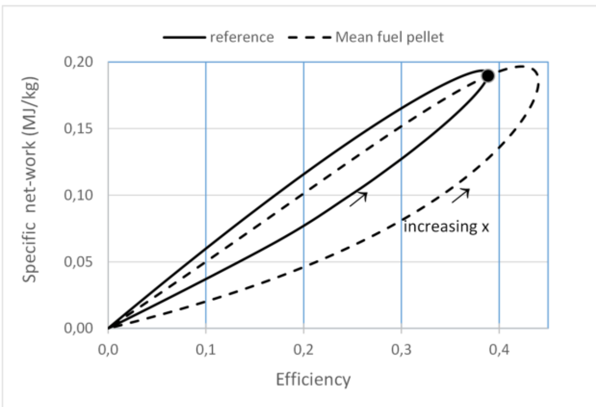

Fig. 9. Cycle net work as a function of efficiency (α = 1000 W/(K m²)).

For the “high T_u” case, higher efficiency can be achieved at smaller x values, but this would result in unreasonable primary circuit temperatures at the optimum efficiency (the PWR primary pressure being 155 bar). It would be interesting to explore this temperature range with a more realistic thermodynamic cycle.

Finally, as expected, we can observe that with this more realistic approach to accounting for compression and expansion irreversibilities, the efficiency at the maximum power output is no longer independent of $\dot{h}_u$ and $\dot{h}_d$, which is a key characteristic of the FTT optimum, still obtained with Eq. (5).

4. An equation for the efficiency of energy conversion cycles

To further support our view on the too restrictive vision of cycle optimisation issues using the FTT approach, we propose in this section a more general efficiency equation. This equation will better illustrate why the actual efficiency is much lower than the Carnot efficiency and provide another perspective on the system optimisation issue.

A real cycle in steady state is a thermodynamic system, made of working fluids (and eventually of solid media in particular if heat exchangers are part of the system), exchanging heat and work with the surroundings, and having a set of internal irreversible processes (friction, diffusion, conduction, mixing …), producing entropy.

We assume that the exchange of heat with the surroundings can be described by a series of elementary terms:

- $\delta\dot{Q}_{u.i}$ with the heat sources, $T_{u.i}$ being here the temperature of the thermodynamic system receiving $\delta\dot{Q}_{u.i}$, and
- $\delta\dot{Q}_{d.j}$ with the heat sinks, $T_{d.j}$ being the temperature of the thermodynamic system losing $\delta\dot{Q}_{d.j}$ (cycle heat losses to the environment can be accounted for in this term).

These lead to the following variation of the system entropy by heat exchanges:

$$\sum_i \frac{\delta\dot{Q}_{u.i}}{T_{u.i}} + \sum_j \frac{\delta\dot{Q}_{d.j}}{T_{d.j}}.$$

The entropy production by irreversibilities inside the cycle components (turbines, pumps, heat exchangers, …) will be described by k elementary terms:

$$\sum_k \dot{\sigma}_k.$$

With $\dot{W}_l$ denoting the set of exchange of work with the surroundings, the global energy and entropy balances of the cycle are:

$$\dot{P} = \left|\sum_l \dot{W}_l\right| = \sum_i \delta\dot{Q}_{u.i} + \sum_j \delta\dot{Q}_{d.j},$$

$$\sum_i \frac{\delta\dot{Q}_{u.i}}{T_{u.i}} + \sum_j \frac{\delta\dot{Q}_{d.j}}{T_{d.j}} + \sum_k \dot{\sigma}_k = 0.$$

We introduce the mean entropic temperature $\overline{T_u}$ for the heat sources and the mean entropic temperature $\overline{T_d}$ for the heat sinks by:

$$\frac{\sum_i \delta\dot{Q}_{u.i}}{\overline{T_u}} = \sum_i \frac{\delta\dot{Q}_{u.i}}{T_{u.i}},$$

$$\frac{\sum_j \delta\dot{Q}_{d.j}}{\overline{T_d}} = \sum_j \frac{\delta\dot{Q}_{d.j}}{T_{d.j}}.$$

It is worth noting that cycle heat losses to the environment will induce an increase of $\overline{T_d}$ signifying a decrease in efficiency, see Eq. (7) just hereafter.

Thus, the cycle efficiency will be given by:

$$\eta = \frac{\dot{P}}{\sum_i \delta\dot{Q}_{u.i}} = 1 + \frac{\sum_j \delta\dot{Q}_{d.j}}{\sum_i \delta\dot{Q}_{u.i}} = 1 - \frac{\overline{T_d}}{\overline{T_u}} - \frac{\overline{T_d}}{\sum_i \delta\dot{Q}_{u.i}} \sum_k \dot{\sigma}_k. \quad (7)$$

In this equation, we have the two well-known reasons which explain that a real system efficiency is always lower than the Carnot efficiency:

– The difficulty to maintain constant temperatures for both the heat source side and the heat sink side, and as close as possible to the available heat source temperatures and heat sink temperatures;
– The use of real machines and components with internal irreversibilities.

We must acknowledge that a similar equation has already been proposed in [7], but for a Carnot cycle only, with irreversibilities in both the compression and expansion processes, so with constant heat source and heat sink temperatures.

We propose to apply Eq. (7) to the "FTT Carnot cycle" calculated in section 2.1. To do that, the thermodynamic system considered now includes the two theoretical heat exchangers (in addition of the cycle working fluid). With this new definition of the thermodynamic system, T_u and T_d are now temperatures seen by the thermodynamic system. With the variables of Section 2.1, Eq. (7) gives:

$$\eta = 1 - \frac{T_d}{T_u} - \frac{T_d}{\dot{Q}_u}(s_{i.u} + s_{i.d}).$$

$s_{i.u}$ and $s_{i.d}$ being the entropy production by the heat transfer irreversible processes within the two theoretical heat exchangers. We have:

$$s_{i.u} = \dot{Q}_u\left(\frac{1}{T_2} - \frac{1}{T_u}\right) = \frac{\dot{h}_u x^2}{T_u(T_u - x)},$$

$$s_{i.d} = -\dot{Q}_d\left(\frac{1}{T_d} - \frac{1}{T_1}\right) = \frac{\dot{h}_d y^2}{T_d(T_d - y)}.$$

Thus, we have an efficiency depending on x and y (T_u, T_d, $\dot{h}_u$ and $\dot{h}_d$ being fixed variables):

$$\eta = 1 - \frac{T_d}{T_u} - \frac{T_d}{\dot{h}_u x}\left(\frac{\dot{h}_u x^2}{T_u(T_u - x)} + \frac{\dot{h}_d y^2}{T_d(T_d - y)}\right).$$

To eliminate y, we can use again the entropy balance:

$$\frac{\dot{h}_u x}{T_u} + \frac{\dot{h}_d y}{T_d} + \frac{\dot{h}_u x^2}{T_u(T_u - x)} + \frac{\dot{h}_d y^2}{T_d(T_d - y)} = 0.$$

which can be rewritten to obtain the entropy balance from Section 2.1:

$$\frac{\dot{h}_u x}{T_u - x} + \frac{\dot{h}_d y}{T_d - y} = 0,$$

thus:

$$y = \frac{T_d \dot{h}_u x}{x(\dot{h}_u + \dot{h}_d) - \dot{h}_d T_u}.$$

So:

$$s_{i.d} = \frac{\dot{h}_d y^2}{T_d(T_d - y)} = -\frac{\dot{h}_u x}{(T_u - x)} \frac{\dot{h}_u x}{x(\dot{h}_u + \dot{h}_d) - \dot{h}_d T_u}.$$

The efficiency is now:

$$\eta = 1 - \frac{T_d}{T_u} - \frac{T_d}{\dot{h}_u x}\left(\frac{\dot{h}_u x^2}{T_u(T_u - x)} - \frac{\dot{h}_u x}{(T_u - x)} \frac{\dot{h}_u x}{x(\dot{h}_u + \dot{h}_d) - \dot{h}_d T_u}\right).$$

This equation can easily be rewritten as:

$$\eta = 1 - \frac{T_d \dot{h}_d}{T_u \dot{h}_d - x(\dot{h}_u + \dot{h}_d)}.$$

Thus, Eq. (3) from Section 2.1 is obtained.

Equation (7) can also be used to introduce the irreversibilities in the compression process (1–2) and the expansion process (3–4), so:

$$\eta = 1 - \frac{T_d}{T_u} - \frac{T_d(s_{i.u} + s_{i.d})}{\dot{Q}_u} - \frac{2\alpha(T_2 - T_1)}{\dot{Q}_u}.$$

Using previous calculations, one has:

$$\eta = 1 - \frac{T_d}{T_u} - \frac{T_d}{\dot{h}_u x}\left(\frac{\dot{h}_u x^2}{T_u(T_u - x)} + \frac{\dot{h}_d y^2}{T_d(T_d - y)}\right) - \frac{2\alpha(T_u - x - T_d + y)}{\dot{h}_u x}.$$

With the entropy balance to compute y:

$$\frac{\dot{h}_u x}{T_u - x} + \frac{\dot{h}_d y}{T_d - y} + 2\alpha(T_u - x - T_d + y) = 0,$$

we will obtain Eq. (6).

It must be acknowledged that the usefulness of Eq. (7) for computing the system efficiency (as in the previous examples) is very limited. In fact, its interest is essentially academic, serving to clarify energy conversion optimisation issues through:

– temperatures as close as possible to the maximum and minimum available values,
– limiting the irreversibilities of the system's components and machines.

Of course, optimisation means result in an increase of the system cost. For certain cycles, a compromise has to be made between these two optimisation paths. For example, in a steam Rankine cycle, water reheating, which involves mixing of cold water and steam, allows an increase of the temperature $\overline{T_u}$, but mixing generates irreversibilities. These irreversibilities reduce the cycle efficiency, but the increase in $\overline{T_u}$ has finally a more significant contribution.

5. Final comments and conclusions

The finite-time thermodynamics (FTT) approach to endoreversible cycles enables the formulation of an elegant relationship known as the FTT efficiency. Derivation of this expression requires consideration of a Carnot cycle. However, given the well-known limitations of this idealised cycle, the Carnot efficiency provides only limited insight into the performance of real systems. Consequently, the same limitation may apply to the FTT efficiency, which is, in essence, a reformulation of the Carnot efficiency expressed in different variables.

It is worth recalling that the derivation of this equation presented in this paper is a quite classical and simple approach in thermal sciences (enthalpy and entropy balances in steady-state conditions), in particular without introducing a "finite time" for the cycle. Indeed, being in steady state, the use of a "finite time" may appear as a paradox.

The use of the FTT efficiency to estimate the efficiency of real systems leads to the problem of an unambiguous choice of the heat source temperature T_u. For a nuclear reactor like a PWR, it seems logical to choose the primary coolant temperature, thus the temperature of the steam generator primary side in order to have a well identified heat exchanger. In this case, we have relatively moderate T_u values, and it is shown that the FTT optimum (the maximum power output which gives the FTT efficiency) gives an acceptable estimation of efficiency (with a slight underestimation, thus a conservative value). But this optimum corresponds to unrealistic operating conditions (much too large differences of temperature within the two FTT heat exchangers). It is these unrealistic conditions which in fact allow the acceptable estimation of efficiency by FTT.

On the contrary, when much larger values of T_u can be assumed, for example by choosing a heat source extending to the nuclear fuel, the FTT optimum was achieved under more realistic operating conditions but with an overestimated efficiency. The determination of all the cycle parameters, obtained with an ideal gas, confirms these conclusions. These results clearly challenge the universality of the FTT efficiency equation.

This discussion about the heat source temperature has highlighted the relevance of the results obtained (cycle temperatures and power output) when the "Chambadal-Novikov" hypothesis can be made, thus when the heat transfer coefficient at the heat sink side is much larger than the heat transfer coefficient at the heat source side. Finally, we recommend to have a well identified thermodynamic system to minimise ambiguity in the selection of temperatures.

Since the FTT approach is driven mainly by maximising cycle work output, we have discussed the "trade-off' between the maximum of efficiency, the maximum of power output and the maximum of specific net work. We have a better illustration of this issue with the introduction of irreversibilities within the cycle and not only on the heat source and heat sink sides. Regardless of the assumption made for T_u, our calculations tend to show that PWR reactors are designed in the high specific net work range.

To further illustrate these cycle optimisation issues, an academic "generalisation" of the Carnot efficiency is presented. This generalisation highlights another important compromise: the trade-off between maximising the heat source temperature (and minimising the heat sink temperature) and limiting the irreversibilities of the system's components and machines.

Having considered in this paper only Carnot cycles with constant heat source temperatures, complementary analyses and computations on the Rankine cycles and on the Brayton cycles will be proposed in a further paper.

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