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Efficient disruptive power plant-based heat engines doing work by means of strictly isothermal closed processes

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Abstract

This research discusses a methodology to integrate strictly isothermal closed processes within thermal cycles characterized by working through contraction processes by extracting heat at free cost. An analysis of a preliminary design study of an engine and cycle doing useful work by expansion and contraction is carried out, whereby the energy balance equations are adjusted when considering contraction work as the core of the problem-solving strategy. The results of the preliminary design study will be applied to the implementation of the disruptive power unit prototype operating with real gasses as working fluids, which allows a precise and clear understanding of the issue of generating useful work through the expansion, contraction, and regeneration of heat by applying advanced heat recovery techniques to convert heat into useful work, thus achieving efficient power units that exhibit the ability to exceed 100% of added thermal energy due to the contribution of the contraction-based work performed at free cost..

Keywords: contraction work, cooling work, forced convection transfer, isothermal contraction work, vacuum work.

Nomenclature

Symbols/units	description
p(bar)	pressure
q_i(kJ/kg)	specific heat in
q_o(kJ/kg)	specific heat out
s(kJ/kg-K)	specific entropy
T(K)	temperature
T_H(K)	top temperature
T_L(K)	bottoming temperature
u(kJ/kg)	specific internal energy
v(m^3/kg)	specific volume
V(m^3)	volume
w(kJ/kg)	specific work
w_i(kJ/kg)	specific work in
w_o(kJ/kg)	specific work out
q_{i12} (kJ/kg)	Input heat to cycle process 1-2
q_{i23} (kJ/kg)	Input heat to cycle process 2-3
q_{i12+23} (kJ/kg)	Input heat to cycles processes 1-2 and 2-3
$q_{iext} = q_i - q_{reg34}$	External heat added to the cycle
q_{o34} (kJ/kg)	Output heat to cycle process 3-4
q_{o41} (kJ/kg)	Output heat to cycles process 4-1
q_{o34+41} (kJ/kg)	Output heat to cycles processes 3-4 and 4-1
q_{reg34} (kJ/kg)	Regenerated heat extracted from cycle process 3-4
w_{oexp} (kJ/kg)	Output expansion work w_{o23} due to added heat
w_{ocont} (kJ/kg)	Output contraction work w_{o41} due to extracted heat
w_n (kJ/kg)	Net useful work ($w_{oexp} + w_{ocont}$) = ($w_{o23} + w_{o41}$)

RF (%)	Cycle regeneration factor = q_{reg34}/q_{o34}
LF (%)	Losses factor (thermal and mechanical irreversibilities)
η_{th} (%)	Cycle thermal efficiency
η_{th_reg} (%)	Regenerative cycle thermal efficiency
η_{th_exp} (%)	Thermal efficiency of expansion process
η_{th_cont} (%)	Thermal efficiency of contraction process
Min_pres (bar)	Minimum cycle pressure
Ref_pres = p_{ref} (bar)	Reference cycle pressure (initial cycle pressure)
Max_pres (bar)	Maximum cycle pressure

1 Introduction

This research combines two concepts that, despite being well known, have never been considered for the design and development of technically disruptive and highly efficient power plants. Such concepts consist of the well-known vacuum used in both reciprocating and rotary steam engines, which includes Rankine and ORC and thermal cycles strictly composed by isothermal closed thermodynamic processes of both expansion and contraction or a vacuum.

The idea that vacuums or contraction pressure (any pressure lower than atmospheric pressure or lower than the initial or reference pressure of a contraction-based thermal cycle) can be used to perform useful mechanical work in heat engines is an ancient concept. Practical vacuum systems are available for carrying out useful mechanical work using vacuums obtained by cooling a thermal working fluid in several ways. For instance, some vacuum systems undergo a change of state via the condensation of the thermal working fluid (from steam to liquid water), which can be carried out in both open and closed processes.

Open processes correspond to Thomas Savery, Thomas Newcomen, and, later, James Watt engines, in addition to actual Rankine and organic Rankine cycles equipped with ultimate state-of-the-art improvements, which condense steam, contributing to the generation of a vacuum. When steam is cooled inside a Rankine cycle condenser, entropy decreases even at a constant temperature. Due to the generated vacuum, a significant amount of work is produced in addition to the work obtained by steam expansion. This makes it possible to obtain useful mechanical work when entropy decreases.

The fact that useful work is obtained when entropy decreases, which is evident by observation, seems contradictory to the second law of thermodynamics. However, it is the key to achieving a thermal machine that uses a vacuum to carry out useful mechanical work via the thermal contraction of the thermal working fluid. If strictly isothermal expansion and contraction are added to this technique, highly disruptive and efficient thermal machines are achieved.

No relevant advances were made to the use of vacuums to carry out mechanical work until the 16th century. Elizabeth Peterson [1–9] reviewed the evolution of original steam engines operating with a vacuum. This review shows that various inventors have contributed to technological progress leading to the current state of the art in this field. These inventors include Jerónimo de Ayanz y Beaumont (1553–1613), a Spanish inventor and engineer who was very active in various technical fields [2]; Thomas Savery (1650–1715) [3]; Thomas Newcomen (1664–1729) [4]; and James Watt (1736–1819) [5]. J.G. van der Kooij (2015) [6] made a valuable contribution to the diffusion of steam engines based on vacuums, highlighting Newcomen's atmospheric steam engine types, in which useful mechanical work is performed under vacuums achieved by condensing the expanded steam exhausted from a cylinder. Later, Watt improved the concept of the atmospheric pressure steam engine designed by Newcomen, implementing prototypes equipped with a steam condenser located independently of the actuator cylinder.

Moreover, Alessandro Nuvolari (2004) [7] reviewed steam power technology involving water vapor considering steam engines that were initially operated at atmospheric pressure through the thermal contraction of steam (vacuum) when it condensed through cooling by heat extraction. Later, Gerald Müller (2013) [8] presented an innovative concept concerning low-temperature-based atmospheric steam engines. The author extended the theory of the atmospheric steam engine operating under a vacuum achieved by heat extraction to show that operation is possible at temperatures between 60 °C and 100 °C, although efficiency is further reduced as the temperature increases.

Similarly, Gerald Müller and George Parker (2015) [9] conducted a series of experiments to assess this theory by including a forced expansion stroke. Recently, the atmospheric steam engine (which implies that useful work is due to the presence of a vacuum) was re-evaluated. According to the authors, the theoretical efficiency of the ideal engine can be increased from 6.5% to 20%.

Three disruptive technological challenges must be overcome to implement power units (PUs) capable of being operated by means of thermal contraction based on a vacuum under strictly isothermal closed processes. The first challenge is that a thermal machine must be able to operate with the aforementioned thermal cycle (i.e., it must be capable of operating through thermal contraction). The second challenge is that the thermal cycles of a thermal machine must be able to operate with strictly isothermal processes of both thermal expansion and contraction. The third technological challenge is that a thermal machine must be able to develop highly effective forced thermal convection heat transfer media at the transfer rate required by the nominal power of each PU, where every PU is composed of a pair of RDACs equipped with associated heat transfer equipment.

The use of vacuums in steam engines is the origin of performing useful mechanical work by thermal contraction. While using a vacuum to obtain useful mechanical work in steam engines has been successfully applied for centuries (Savery and Watt steam engines), it has not been possible to develop strictly isothermal processes for efficient commercial industrial applications. However, during the last three decades, some attempts to advance isothermal processes have been made [10–17].

Recently, Vítor Augusto Andreghetto Bortolin et al. (2021) [10] proposed a complete thermodynamic model for the adiabatic and isothermal atmospheric steam cycle that uses real gas data. The model was constructed to accommodate the forced expansion of low-pressure steam. The results show that the adiabatic cycle is more efficient than the isothermal cycle and that the amount of heat needed to keep the expanding steam at a constant temperature is prohibitive for practical applications. Knowlen C. et al. (1997) [11] developed an automotive propulsion concept of an open Rankine cycle that utilizes liquid nitrogen as the working fluid and discussed several means of achieving quasi-isothermal expansion. The authors claim that if sufficient heat input during the expansion process can be realized, then this cryogenic propulsive system will provide greater automotive ranges and lower operating costs than those of electric vehicles currently being considered for mass production.

Cicconardi S. et al. (1999) [12] studied a steam cycle and the effect of flow variation on cycle performance. The authors claim that isothermal expansion with increasing flow decreases cycle efficiency due to the greater condenser losses and the impossibility of fully recovering the available heat at the end of the expansion at high superheated temperatures. In other research, Cicconardi S. et al. (2001) [13] showed that isothermal expansion can achieve high efficiency (up to 70% of HHV) when the waste heat at the turbine outlet is recovered for pre-heating water, hydrogen, and oxygen. Park J.K. et al. (2012) [14] analyzed a quasi-isothermal thermal cycle for its application in an underwater energy storage system. The analysis of the heat transfer cycle confirmed the validity of the quasi-isothermal nature of the design based on water pistons.

Kim Y.M. et al. (2013) [15] reviewed current thermo-electric energy storage (TEES) systems and proposed a novel isothermal TEES system with transcritical CO_2 cycles. For the given efficiencies of the compressor and expander, the maximum round-trip efficiency decreased rapidly with an increase in the back work ratio. The authors showed that the round-trip efficiency of the isothermal TEES system can be increased because it has a lower back work ratio than in the isentropic case.

Meanwhile, Opubo N. et al. (2013) [16] proposed a variant of a thermal engine that uses isothermal expansion to achieve a theoretical efficiency close to the Carnot limit and in which steam generated inside a power cylinder eliminates the need for an external boiler. The device is suitable for slow-moving applications, and preliminary experiments have shown a cycle efficiency of 16% and a high work ratio of 0.997. Later, Opubo N. et al. (2014) [17] reviewed various low-temperature vapor power cycle heat engines with quasi-isothermal expansion using methods to realize the heat transfer. In this experiment, the heat engines took the form of either a Rankine cycle with continuous heat addition during the expansion process or a Stirling cycle with a condensable vapor as the working fluid.

R. Ferreiro et al. [18–21] presented state-of-the-art technologies for thermal cycles that allow operation with strictly isothermal closed processes of both thermal expansion and contraction. Meanwhile, some advances in the ability to achieve highly effective forced thermal convection heat transfer media at the required transfer rate appear in other research [22–24].

During the last three decades, a great effort has been made to implement highly effective heat recovery units. For this reason, there are many studies on advanced design techniques. Of particular interest is the work of Meeta Sharma and Onkar Singh (2013) [22], who considered the physical parameters of an heat recovery steam generator HRSG to study their implications on HRSG design by comparing an existing plant design with an optimized plant design. Another relevant work is that by Meeta Sharma and Onkar Singh (2014) [23], who presented the exergy analysis of an HRSG for calculating exergy losses, heat transfer, and pressure losses for different physical components. They found that various HRSG sub-sections have different physical parameters, such as fin density, fin thickness, fin height, tube diameter, and fin spacing, and have a noticeable effect on exergy loss minimization. Such information aids in designing new HRSG technologies and reducing thermal losses in existing HRSGs. Furthermore, such design strategies and advanced methodologies could be applied to the task of designing HREs to achieve prototypes with greater heat transfer effectiveness.

In Newcomen and James Watt steam engines [4–5] that operate at atmospheric pressure in the high-pressure zone and with a vacuum in the low-pressure zone, the volumetric zone of the cylinder in contact with the condenser was subjected to a pressure lower than the atmospheric pressure known as a "vacuum." The work obtained in these engines was due to the generation of a vacuum by cooling via heat extraction from the condenser in an open process. The open process of cooling carried out by heat extraction responsible for this vacuum decreases entropy while providing useful mechanical work. This common phenomenon occurs in reciprocating steam engines at constant pressure, constant temperature, and change of state in open process-based transformations.

Based on the above description, the concept of obtaining useful work accompanied by the decrease in entropy can be expressed by the following statement concerning isothermal vacuum-based work: "It is not possible to perform mechanical work by open process-based contraction due to a vacuum without a decrease in entropy" or "The performance of mechanical work by open-process-based contraction due to a vacuum leads to a decrease in entropy." According to this statement, the decrease in entropy is a consequence of cooling only (independent of temperature and pressure).

Experimental validation involves observing that no useful mechanical work is done if the vacuum is eliminated due to a lack of cooling in the condenser because there is no difference in pressure between atmospheric and vacuum, causing the absence of forces between both sides of the piston. This idea is considered in this research work under closed processes to achieve mechanical work by strictly isothermal contraction under closed processes.

2 Description of a disruptive thermal engine and its thermal cycle through a case study

This section deals with the structure of a heat converter-based PU and its associated thermal cycle capable of doing useful work with strictly isothermal closed processes through both the expansion and contraction of the thermal working fluid. Implementing a PU responsible for converting heat into electrical power via useful mechanical work requires the arduous task of preliminary design to reduce as much error and deviation as possible from the expected results. Thus, a prototype can be obtained that allows testing that requires adjusting the results to the desired goals and proposed objectives.

Based on a patent pending application (priority number: P202200035), the structure of such a PU must be specified. This PU must also be able to withstand a thermal cycle characterized by operating with closed adiabatic and strictly isothermal processes to obtain useful mechanical work for both the expansion and contraction of the thermal working fluid. Each PU consists of two actuators based on reciprocating double-acting cylinders (RDACs) that work intermittently; that is, when one of the actuator cylinders performs mechanical work due to the displacement of the piston, the other actuator cylinder remains at rest, and vice versa. This mode of operation is necessary because the isochoric processes of adding and removing heat do not allow the piston to move. Therefore, the time during which the piston of an actuating cylinder remains immobile is used to carry out isochoric processes. During that time, the second actuating cylinder also performs useful mechanical work.

2.1 PU structure

As depicted in Figure 1, a single reciprocating double-acting cylinder actuator should be equipped with at least the following basic accessories:

-- A heat supply circuit comprising power sources, recirculation pumps, and heat supply transfer fluid (thermal oil, molten salt fluids)

-- A countercurrent heater (liquid/gas or gas/gas) to transfer heat from the heat transfer fluid to the hot side of the thermal working fluid

-- A hot thermal working fluid heated by forced convection by means of forced convection fans

-- A cold thermal working fluid circuit cooled by forced convection by means of forced convection fans

-- A countercurrent cooler (liquid/gas or gas/gas)

-- A heat extraction circuit comprising a heat sink, recirculation pump (not shown in the figures), and heat extraction transfer fluid (cool water or air)

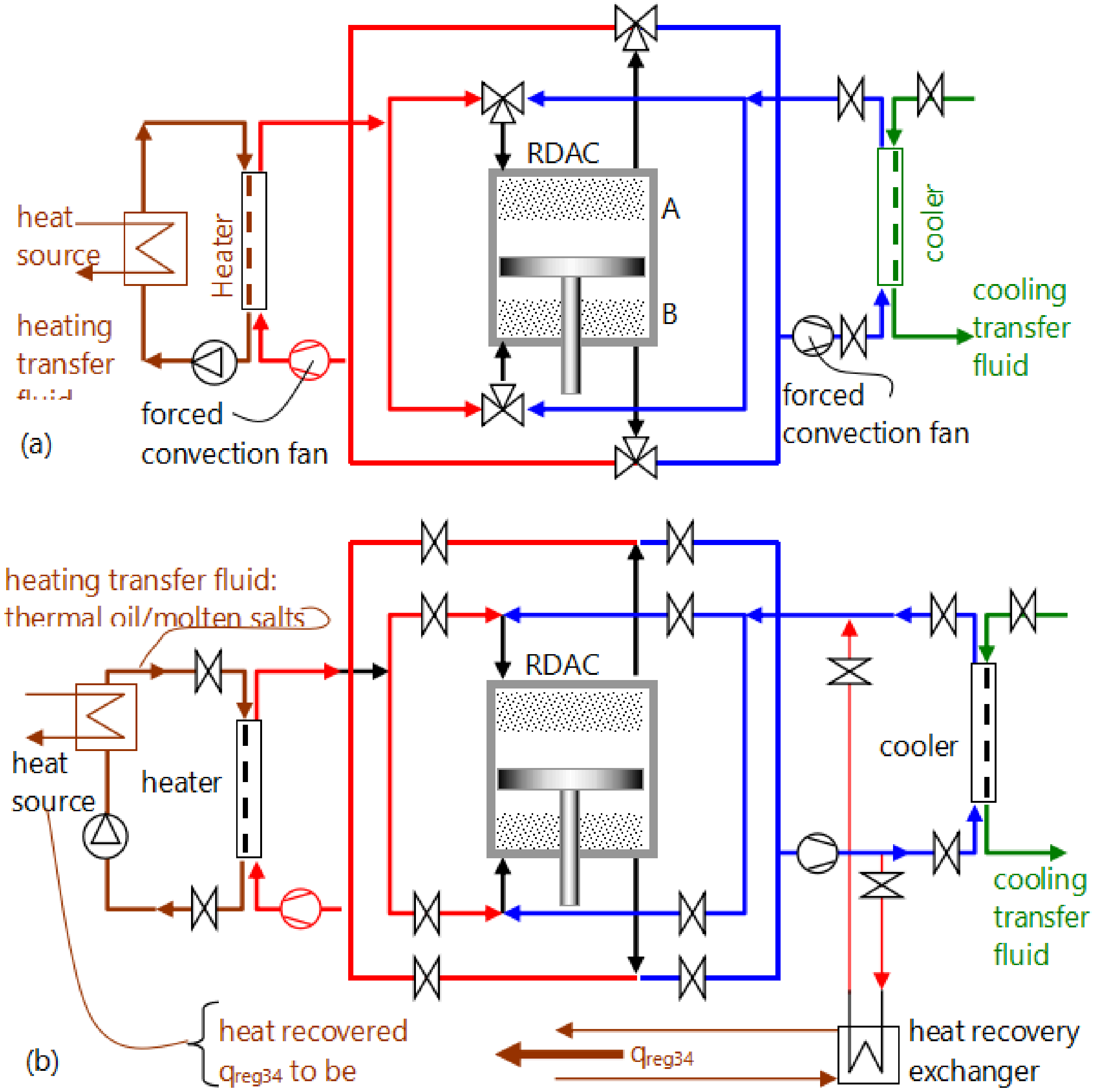

Figure 1. Simplified scheme of a discontinuous motion prototype for a PU (patent priority number: P202200035), characterized by requiring two RDACs to complete a continuous motion cycle: (a) a PU without heat recovery system at cycle level and (b) a regenerative PU equipped with a heat recovery system at cycle level. The cylinder actuator must be equipped with a volumetric clearance (per cylinder chamber A and B), assumed to be a dead volume inside each cylinder chamber.

2.2 Non-regenerative isochoric-isothermal-isochoric-isothermal (VTVT) thermal cycle

The transformations carried out in the cycle are sequentially done in the following order: isochoric heat addition (V) constant, isothermal expansion (T) constant, isochoric heat extraction (V) constant, and isothermal contraction (T) constant. This sequence of closed processes is denoted as “VTVT.” The VTVT thermal cycle proposed to operate the thermal machine shown in Figure 1 is represented in Figure 2 and Table 1. This cycle comprises a sequence of four closed thermodynamic processes carried out along the proposed thermal cycle with a given thermal working fluid.

The first process is an isochoric heat addition process 1-2, during which the pressure, temperature, and entropy increase while the piston remains stationary (motionless). The second process is a closed process of isothermal expansion 2-3, which is responsible for doing output useful mechanical work. This requires the addition of an amount of heat equal to the work done. Since this process is not isochoric, the piston moves the entire stroke, while the pressure decreases as volume increases. The expansion process associated with the increase in entropy ends when the reference pressure in point 1 is reached at the end of the stroke. The third process is an isochoric process of heat removal 3-4, during which the pressure, temperature, and entropy decrease while the piston remains stationary. Finally, the fourth process is a closed process of isothermal contraction 4-1, which causes the displacement of the piston throughout its stroke until the starting position at point 1. This process produces useful mechanical work, which leads to a decrease in entropy and which requires heat extraction equal to the work produced.

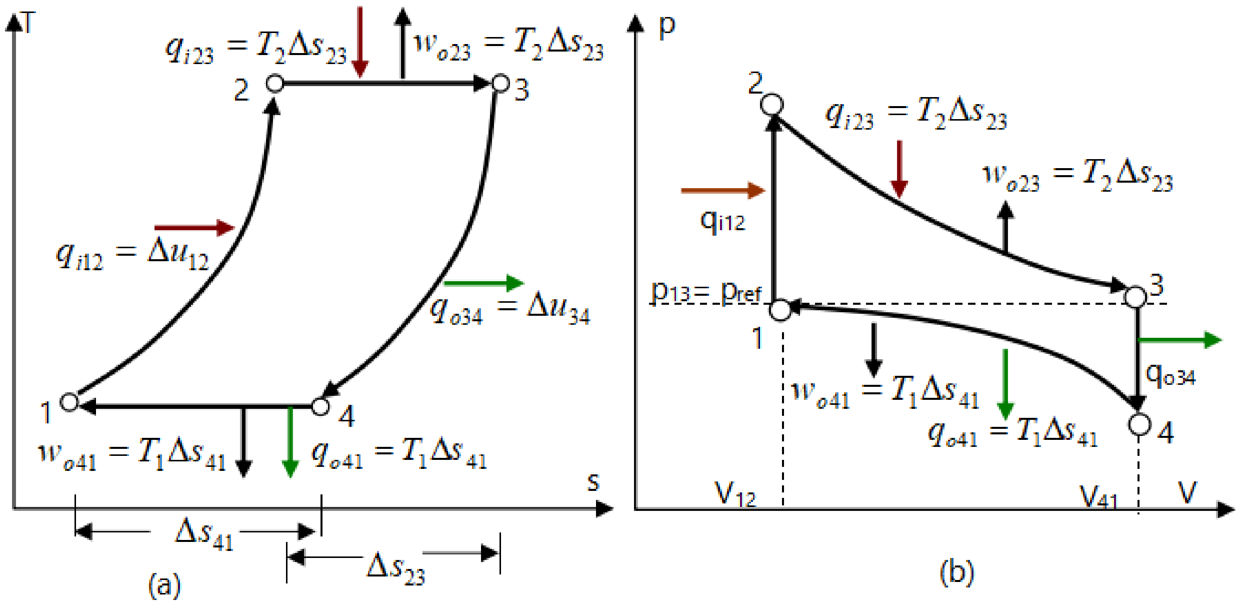

Figure 2. T-s and p-V diagrams of the VTVT cycle showing the math models of its processes.

Table 1. The sequence of heat-work interactions carried out simultaneously in both cylinder chambers A and B of the reciprocating double-acting cylinder actuator shown in Figure 1. The process equations are presented in Equations (8)–(12).

Cycle performed into cylinder chamber A				Cycle performed into cylinder chamber B			
point	useful work	constant	heat in/out	point	useful work	constant	heat in/out
1-2	motion blocked	V	$q_{i23} = T_2\Delta s_{23}$	3'-4'	motion blocked	V	$q_{o34} = \Delta u_{34}$
2-3	$q_{i23} = T_2\Delta s_{23}$	T	$q_{i23} = T_2\Delta s_{23}$	4'-1'	$q_{i12} = \Delta u_{12}$	T	$q_{o41} = T_1\Delta s_{41}$
3-4	motion blocked	V	$q_{o41} = T_1\Delta s_{41}$	1'-2'	motion blocked	V	$q_{i12} = \Delta u_{12}$
4-1	$q_{i12} = \Delta u_{12}$	T	$q_{o41} = T_1\Delta s_{41}$	2'-3'	$q_{i23} = T_2\Delta s_{23}$	T	$q_{i23} = T_2\Delta s_{23}$

As shown in Figure 3, the transformations in the cycle are carried out in the following order: isochoric heat addition (constant V), isothermal expansion (constant T), isochoric heat extraction (constant V), and isothermal contraction constant T).

2.3 Thermal cycle execution

Figure 3 details the operation mode for simultaneously executing the VTVT cycle in each cylinder chamber (A and B).

According to Figure 3, a thermal cycle is executed within chambers A and B of the cylinder. Both are identical and carried out simultaneously but half a cycle out of phase with one another. Therefore, the mode of operation is as described below.

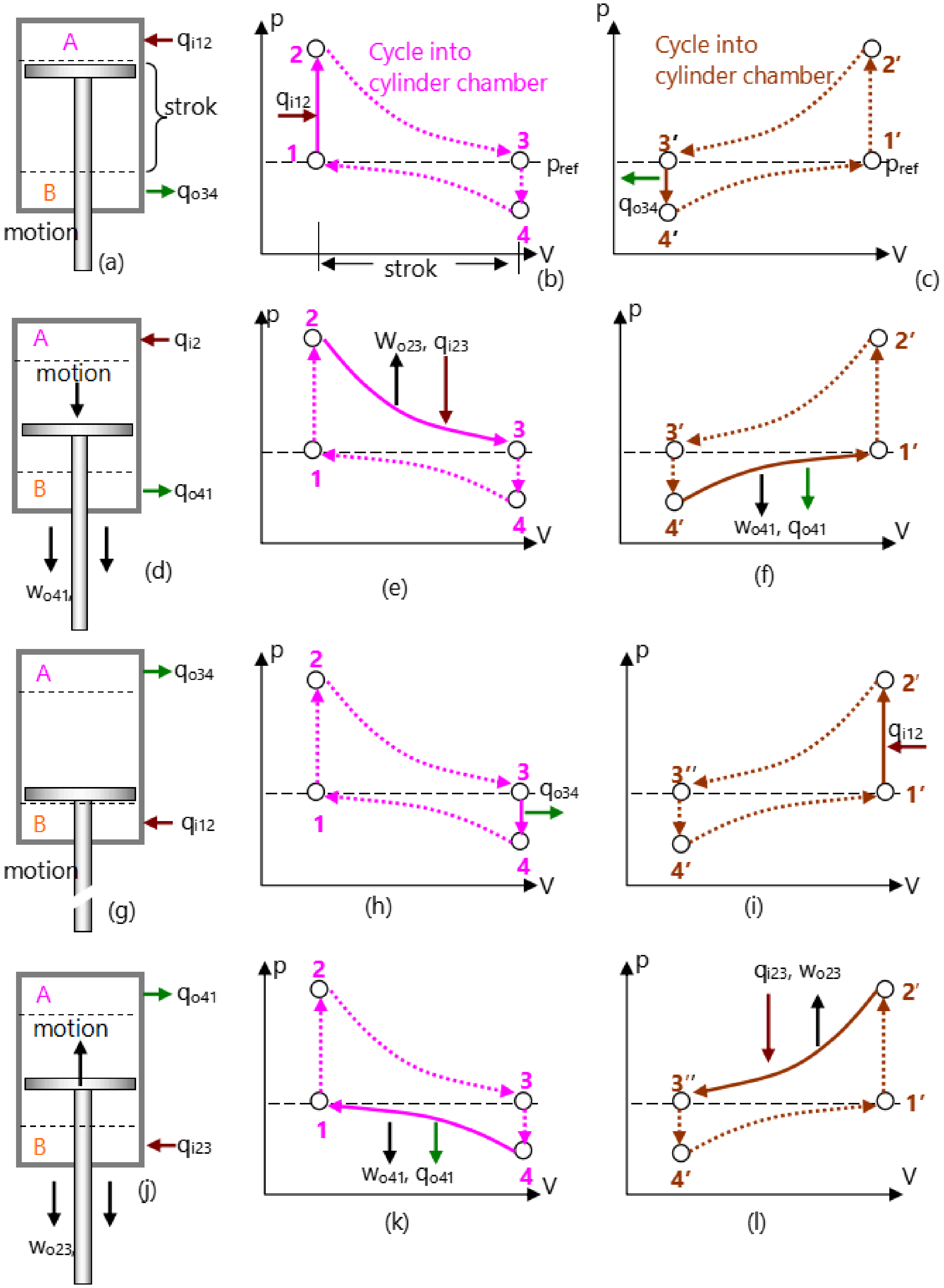

Figure 3. A reciprocating double-acting cylinder that can execute a thermal cycle comprising two closed isochoric processes and two closed isothermal processes in each cylinder chamber (A and B).

Figures 3(a), 3(b), and 3(c) depict the cycle phase of closed isochoric heat addition q_{i12} to cylinder chamber A and closed isochoric heat extraction q_{o34} from cylinder chamber B. Figures 3(d), 3(e), and 3(f) depict the cycle phase of closed isothermal heat addition q_{i23} to cylinder chamber A, which undergoes output useful work w_{o23}, and isothermal heat extraction q_{o41} from cylinder chamber B, which undergoes output useful work w_{o41}. Figures 3(g), 3(h), and 3(i) depict the cycle phase of closed isochoric heat extraction q_{o34} from cylinder chamber A and closed isochoric heat addition q_{i12} to cylinder chamber B. Figures 3(j), 3(k), and 3(l) depict the cycle phase of closed isothermal heat extraction q_{o41} from cylinder chamber A, which undergoes output useful work w_{o41}, and closed isothermal heat addition q_{i23} to cylinder chamber B, which undergoes output useful work w_{o23}.

2.4 Heat transfer strategy

According to Figure 3, isochoric heat addition and extraction contribute to changes in internal energy and pressure. Meanwhile, isothermal heat addition and extraction contribute to useful work by the expansion and contraction, respectively, of a thermal working fluid, which undergoes an increase and decrease of entropy associated with a decrease or increase in pressure, respectively.

The cylinder actuator depicted in Figure 3(a) shows the processes of closed isochoric heat addition to cylinder chamber A by countercurrent forced convection heat transfer driven by a recirculation fan and closed isochoric heat extraction from cylinder chamber B by countercurrent forced convection heat transfer generated by a recirculation fan, represented by p-V diagrams (b) and (c), respectively. Piston motion remains blocked during both simultaneous heat transfer processes so that closed isochoric processes can be achieved.

The cylinder actuator depicted in Figure 3(d) shows the processes of closed isothermal heat addition to cylinder chamber A by countercurrent forced convection heat transfer driven by a recirculation fan and closed isothermal heat extraction from cylinder chamber B by countercurrent forced convection heat transfer generated by a recirculation fan, represented by the p-V diagrams shown in Figures 3(e) and 3(f). During both simultaneous heat transfer processes, piston motion does useful work via closed isothermal expansion in (e) and by closed isothermal contraction in (f).

The cylinder actuator depicted in Figure 1(g) shows the processes of closed isochoric heat extraction from cylinder chamber A by countercurrent forced convection heat transfer driven by a recirculation fan and closed isochoric heat addition to cylinder chamber B by countercurrent forced convection heat transfer generated by a recirculation fan, represented by the p-V diagrams shown in Figures 3(e) and 3(f). Piston motion remains blocked during both simultaneous heat transfer processes so that closed isochoric processes can be achieved.

The cylinder actuator depicted in Figure 1(j) shows the processes of closed isothermal heat extraction from cylinder chamber A by countercurrent forced convection heat transfer driven by a recirculation fan and isothermal heat addition in cylinder chamber B by countercurrent forced convection heat transfer generated by a recirculation fan, represented by the p-V diagrams shown in Figures 3(e) and 3(f). During both simultaneous heat transfer processes, piston motion does useful work by contraction in (k) and expansion in (l).

Figure 4 summarizes the sequence of processes executed in the reciprocating double-acting cylinder shown in the p-V diagrams in Figure 3. These diagrams correspond to the sequence of processes depicted in Table 1, which shows the thermodynamic model of the closed heat-work interactions carried out simultaneously in cylinder chambers A and B.

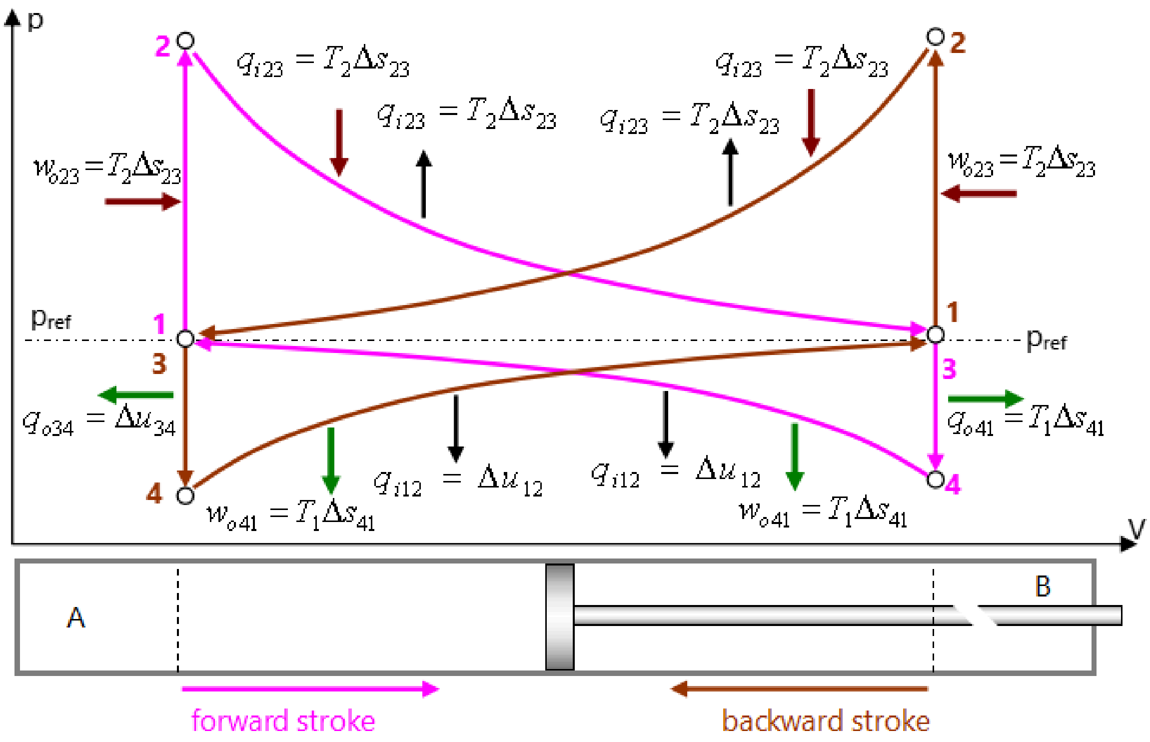

Figure 4. Summary of the operation modes of the reciprocating double-acting cylinder showing the p-V diagram, which includes the adiabatic and isothermal closed process models carried out during each cycle according to process Equations (8–12).

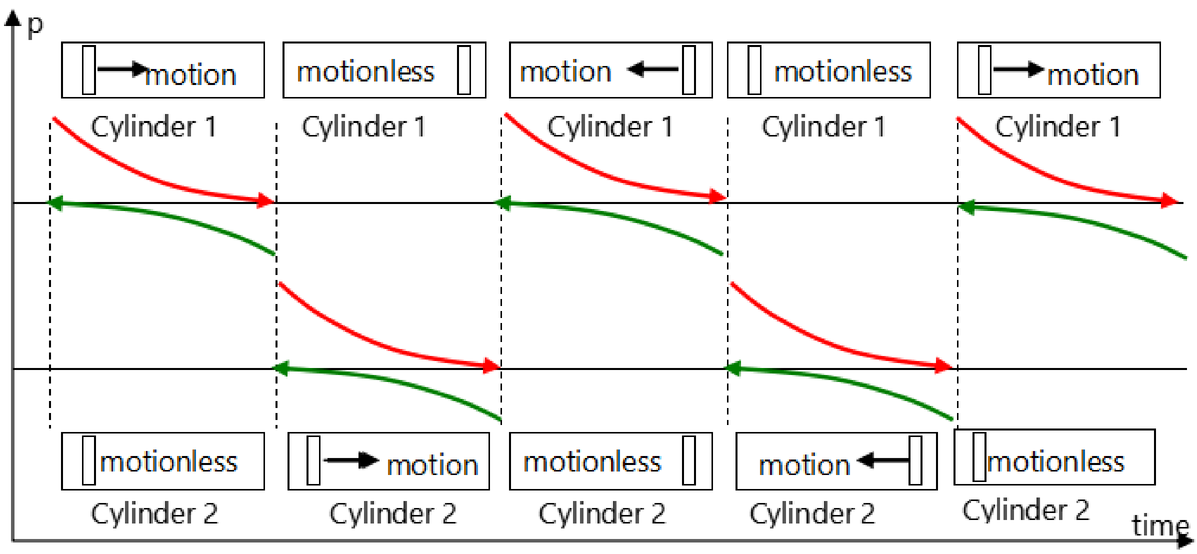

Figure 5. Timing chronogram depicting the action of each cylinder actuator (in terms of motion doing work by expansion and contraction processes) to complete a cycle.

In the schedule shown in Figure 5, the work performed by each cylinder is associated with the performance of mechanical work of each actuating cylinder is represented. In the schedule shown in Figure 5, the performance of two actuator cylinders operating alternatively to complete each cycle of the power unit is represented. The alternative mode of operation is that when one of the cylinders is active performing useful mechanical work, its complementary cylinder remains stationary. During the time in which one of the cylinders remains stationary, the heat addition task is carried out in one of the chambers, while the heat extraction task is simultaneously carried out in the complementary chamber.

2.5 Achieving continuous motion

Because the piston motion remains locked during the isochoric heating and cooling time, the movement of the piston must be intermittent in each cycle. The time during which each piston remains stationary is used to transfer heat at a constant volume. Therefore, two cylinders per PU need to operate alternatively to achieve continuous movement. That is, when the motion of one cylinder is blocked, the other cylinder is free to move and do useful work, and vice versa.

Figure 5 shows a timing chronogram depicting the action of each single-cylinder actuator. As shown in the figure, when one piston's motion is blocked, the other piston moves. The result of the composition of the individual motion of each cylinder is a continuous displacement or rotating movement. This forced convection heat transfer strategy (patented) inherently allows strictly isothermal processes in which almost all the heat added to the process is converted to useful mechanical work. This heat transfer technique is applied to both the addition and removal of heat, thus generating useful mechanical work due to thermal contraction. Therefore, the useful mechanical work due to the addition and removal of heat could be greater than the amount of heat added to the thermal cycle under restricted losses. The mechanism responsible for concatenating the individual movements of both cylinders to achieve a continuous movement is described in patent application number P202200035.

2.6 VTVT cycle transformations

The thermal cycle depicted in Figure 2 exhibits certain thermodynamic properties for closed processes in terms of symmetries between changes in entropy and temperature. Such properties obey both empirical and experimental observations.

Therefore, assuming that

$$T_2 = T_3;\ \ T_1 = T_4\ , \tag{1}$$

undergoes the equalities of

$$u_2 = u_3;\ \ u_1 = u_4\ . \tag{2}$$

Eq. (2) has the following consequences:

$$T_2 - T_1 = T_3 - T_4 \tag{3}$$

Equation (2) has the following consequences:

$$\Delta T_{12} = \Delta T_{34} = \Delta T;\ \ \Delta u_{12} = \Delta u_{34} = \Delta u \tag{4}$$

Concerning entropy changes, it follows that

$$\Delta s_{23} = \Delta s_{41} = \Delta s \tag{5}$$

The equality of Equation (5) implies that

$$w_{o23} = T_2 \cdot \Delta s_{23} = T_2 \cdot \Delta s = q_{i23} \tag{6}$$

and

$$w_{o41} = T_1 \cdot \Delta s_{41} = T_1 \cdot \Delta s = q_{o41} \tag{7}$$

The following thermodynamic transformations are necessary to carry out a performance analysis.

Total added heat:

$$q_i = q_{i12} + q_{i23} = Cv \cdot (T_2 - T_1) + T_2 \cdot (s_3 - s_2) \tag{8}$$

According to Equation (3), this can be written as

$$q_i = q_{i12} + q_{i23} = Cv \cdot \Delta u + T_2 \cdot \Delta s \tag{9}$$

Total extracted heat:

$$q_o = q_{o34} + q_{o41} = Cv \cdot (T_3 - T_4) + \cdot T_1 (s_4 - s_1) \tag{10}$$

According to equation (5), this can be written as

$$q_o = q_{o34} + q_{o41} = Cv \cdot \Delta u + T_1 \cdot \Delta s \tag{11}$$

The output work due to adding heat or heating responsible for expansion work (w_{oexp}) is

$$w_{o\exp} = w_{o23} = T_2 \cdot (s_3 - s_2) = T_2 \cdot \Delta s \tag{12}$$

The output work due to extracting heat or cooling responsible for contraction work (w_{cont}) is

$$w_{ocont} = | w_{o41} | = T_1 \cdot | (s_4 - s_1) | = T_1 \cdot | \Delta s | \tag{13}$$

The total output of useful work due to isothermal expansion and isothermal contraction is

$$w_o = w_n = w_{o23} + | w_{o41} | = T_2 \cdot (s_3 - s_2) + T_1 \cdot | (s_4 - s_1) | \tag{14}$$

which can be written as

$$w_o = w_n = T_1 \cdot | \Delta s | + T_2 \cdot \Delta s = (T_1 + T_2) \Delta s \tag{15}$$

since $| \Delta s | = \Delta s$

Thermal efficiency, which is the ratio of the output useful work to the total added heat, according to Equations (13) and (7), is

$$\eta_{th} = ratio\ (w_o / q_i) = \frac{(T_1 + T_2) \cdot \Delta s}{Cv \cdot \Delta u + T_2 \cdot \Delta s} \tag{16}$$

Equation (16) is assimilated as the thermal efficiency for a non-regenerative thermal cycle that does useful mechanical work through closed and strictly isothermal processes.

2.6.1 Controversial modeling of the VTVT thermal cycle

It is necessary to consider that part of the useful work of the cycle is due to the thermal contraction phase carried out by isothermal cooling, owing to the extraction of heat in a closed process, during which useful mechanical work is done while entropy decreases. This phenomenon is responsible for the controversy between energy balances. This thermal cycle exhibits contradictions in terms of the first principle concerning the energy balance (heat and work) carried out in conventional thermal cycles (which do output mechanical work only by adding heat). This is because it is possible to do useful mechanical work via the thermal contraction of a working thermal fluid by extracting heat. Therefore, according to Equation (9), the added heat is

$$q_i = Cv \cdot \Delta u + T_2 \cdot \Delta s = Cv \cdot \Delta u + w_{o\exp} \tag{17}$$

and according to Equation (11), the extracted heat is

$$q_o = Cv \cdot \Delta u + T_1 \cdot \Delta s = Cv \cdot \Delta u + w_{ocont} \tag{18}$$

From Equation (15), the net output useful work is

$$w_o = w_n = T_1 \cdot | \Delta s | + T_2 \cdot \Delta s = (T_1 + T_2) \Delta s = w_{o\exp} + w_{ocont} \tag{19}$$

The energy balance of a conventional thermal cycle satisfies the following equation:

$$q_i - q_o = w_o = w_n \tag{20}$$

Thus, implementing Equations (17) and (18) into Equation (20) yields

$$q_i - q_o = w_o = Cv \cdot \Delta u + w_{o\exp} - (Cv \cdot \Delta u + w_{ocont}) = w_{o\exp} - w_{ocont} \tag{21}$$

It can be seen that the result of Equation (19) is different from the result of Equation (21). This difference indicates a discrepancy between the energy balances of conventional thermal cycles and thermal cycles based on thermal contraction. In addition, no fundamental laws of thermodynamics are violated when modeling energy balances based on the first principle. It is worth noting the fact of considering the steam engines that operated at atmospheric pressure on the high-pressure side and vacuum at the low-pressure side caused by the condensation of steam at a constant temperature [4–9], where useful mechanical work is done undergoing an isothermal decrease in entropy. This controversy has been assiduously accepted for more than two centuries—for example, by Newcomen (1664–1729) [4] and James Watt (1736–1819) [5]. Based on this controversy, RDACs operating at atmospheric pressure on the side of high pressure and in a vacuum on the side of low pressure have been used.

Consequently, the differences between output mechanical work definitions for thermal cycles that operate by expansion or expansion plus contraction are described below.

Thermal cycles that do work by expansion (conventional thermal cycles operating with open or closed processes):

$$w_o = q_i - q_o = w_{o\exp} \tag{22}$$

Thermal cycles that do work by expansion and contraction (based on RDACs operating with closed processes):

$$w_o = w_{o\exp} + w_{ocont} \tag{23}$$

2.7 Regenerative VTVT cycle

Figures 2, 3, and 4 show that a significant fraction of the heat extracted to generate the vacuum and work by the thermal contraction of the thermal working fluid in the discontinuous thermal cycle described above can be used or recovered in a heat regeneration strategy in the considered thermal cycle depicted in Figures 6(a) and 6(b). The final temperature of the expansion process is equal to the initial temperature, which means that the final internal energy is equal to the initial internal energy. At the final stage of the expansion process, this significant amount of heat is transferred to the cooling fluid as extracted heat using a recovery technique describe at Ramon Ferreiro (2023) [24]. Consequently, the cooling fluid contains a useful fraction of the transferred heat in the form of low-grade heat. The heat regeneration process aims to recover as much of this transferred heat as possible.

According to the T-s and p-V diagrams in Figures 6(a) and 6(b), the heat regeneration procedure is modeled as follows:

$$q_{i12} = q_{iext} + q_{reg34} \tag{24}$$

The regeneration factor (RF) is the ratio of the amount of heat regenerated (q_{reg34}) to the amount of heat extracted (q_{o34}) by cooling and is calculated as

$$RF = \frac{q_{reg34}}{q_{o34}} \tag{25}$$

$$q_{reg34} = RF \cdot q_{o34} \tag{26}$$

The external heat added to the cycle is achieved by combining Equations (14) and (15), which yields

$$q_{iext} = q_{i12} - q_{reg34} = q_{i12} - RF \cdot q_{o34} \tag{27}$$

Equation (27) shows that the higher the heat recovery (q_{reg34}), the lower the amount of external heat (q_{iext}) added to the cycle, which increases the ratio of the net work to the external heat added and, consequently, the thermal efficiency of the cycle.

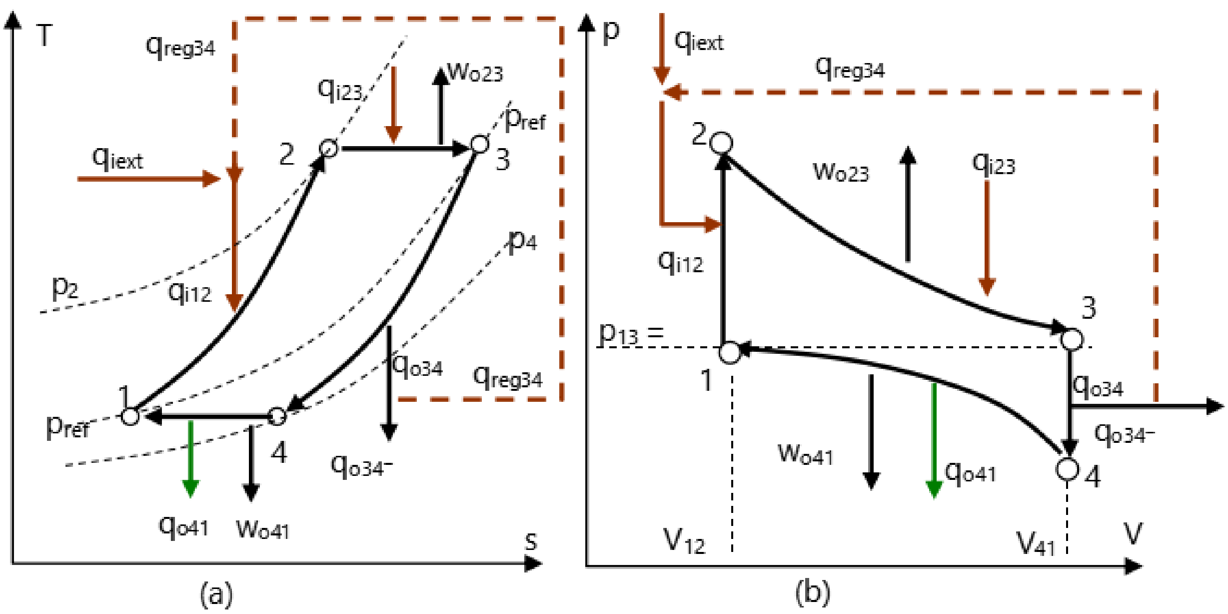

Figure 6. Regenerative VTVT thermal cycles depicting the thermodynamic processes carried out during each regenerative cycle: (a) a T-s regenerative diagram and (b) a p-V regenerative diagram.

2.8 Modeling performance criteria

In conventional thermal cycle analyses (for thermal cycles in which the useful work is due only to the addition of heat), the relationship between thermodynamic work and added heat is considered a performance criterion, while all types of energy losses (thermal efficiency) are neglected. In the case under study, there are other behavioral indices, including those described below.

The thermal efficiency of the ratio of the work achieved by adding heat to the added heat (isochoric plus isothermal) is given as

$$\eta_{\text{th\_exp}} = \frac{w_{o\exp}}{q_i} = \frac{w_{o\exp}}{q_{i12} + q_{i23}} \,. \tag{28}$$

The ratio of the work achieved by heat extraction to the extracted heat (isochoric plus isothermal) is given as

$$\eta_{\text{th\_cont}} = \frac{w_{ocont}}{q_o} = \frac{w_{ocont}}{q_{o34} + q_{o41}} \,. \tag{29}$$

Thermal efficiency, as the ratio of the work achieved by heat extraction to the work achieved by total heat addition, is given as

$$\text{Ratio } \frac{w_{ocont}}{w_{o\exp}} \,. \tag{30}$$

Thermal efficiency, as the ratio of the net work due to heat addition to the total added heat (isochoric plus isothermal), is given as

$$\eta_{th} = \frac{w_n}{q_i} = \frac{w_{o\exp} + w_{ocont}}{q_{iext} + q_{i23}} \,. \tag{31}$$

According to Equation (27), $q_{iext} = q_{i12} - q_{reg34}$, where $q_{reg34} = 0$ for a cycle without heat regeneration. Thus, $q_{iext} = q_{i12}$. Consequently, Equation (31) can be written as

$$\eta_{th} = \frac{w_n}{q_i} = \frac{w_{o\exp} + w_{ocont}}{q_{i12} + q_{i23}} \quad (32)$$

The regenerative cycle thermal efficiency is defined as the ratio of net work (due to the external heat added plus the heat extracted) to the added external heat and is given as

$$\eta_{th\_reg} = \frac{w_n}{q_i} = \frac{w_{o\exp} + w_{ocont}}{q_{iext} + q_{i23}} = \frac{w_{o\exp} + w_{ocont}}{q_{i12} - q_{reg34} + q_{i23}} = \frac{w_{o\exp} + w_{ocont}}{q_{i12} - RF \cdot q_{o34} + q_{i23}} \quad (33)$$

Equation (31) exhibits disruptive consequences for the thermal cycle that need to be explained. Useful work is done by the contraction of the thermal working fluid as a consequence of the extraction of heat by cooling the confined working fluid. Useful work is also done by regenerating the heat recovered from the extracted heat. Therefore, a high value of the heat regeneration factor combined with a low irreversibility factor (including thermal and mechanical irreversibility) can cause the useful work output to be greater than the external heat input. In this case, the PU can produce useful work without adding external heat to the thermal cycle. This means it is possible to feed the PU back from the output power via electrical heating and to take advantage of the excess power produced without needing to provide heat from an external source. This issue is addressed in the analysis of the results to estimate the achieved auto-power based on PU irreversibility and regenerated heat.

2.9 Case study of an RDAC-based PU

The disruptive PU to be prototyped is simulated using the thermodynamic model of a single RDAC—depicted in Figure 1 and described by Equations (8)–(16)—whose T-s and p-V diagrams are represented in Figure 2. The next task is to achieve performance results from cycle analyses that express the work done by the isothermal expansion and isothermal contraction for a discontinuous motion prototype depicted in the T-s and p-V diagrams of Figure 2, as well as performance parameters useful for evaluating the achievement concerning conventional thermal engines in which two units are needed to complete a single PU. Thus, the prototype based on the discontinuous motion PU depicted in Figure 1 is characterized by two RDACs.

The cycle represented in Figures 1 and 2 was calculated for a group of cases arranged as a function of top temperatures ranging from 400 K to 800 K while using helium and air as real working fluids (Lemmon E.W. et al., 2007) [25].

Table 2 shows the data related to the processed VTVT thermal cycles corresponding to the study cases with helium and air as the thermal working fluid. The temperature, pressure, specific volume, internal energy, and entropy are depicted in each state point of every cycle. Thus, according to the notation proposed in the table of symbols, the case studies are depicted in Table 3 for helium and air. Each case is studied as a function of the top and bottom temperatures and the reference pressure (the cycle initial pressure). The top temperature ranges from 400 to 800 K, while the bottom temperature is fixed at 300 K. The selected design pressure is 10 bars. The cycle is computed using the database referenced in Lemmon E.W. et al. (2007) [25]. The results of each case when helium and air were used as the thermal working fluid include state point temperatures (K), pressures (bar), specific volume (m^3/kg), internal energies (kJ/kg), and entropies (kJ/kg-K).

The cases depicted in Table 2 show the data corresponding to each state point of the cycle computation for helium and air as working fluids. The results of the cycle analysis were computed with the data in Table 2 and are depicted in Table 3.

Table 2. Data resulting from cycle VTVT analysis for cases when helium and air were used as working fluids corresponding to the T-s and p-V diagrams of Figure 2.

		Helium			T_H = 400K		Air		
sp	T(K)	p(bar)	v(m^3/kg)	u(kJ/kg)	s(kJ/kg.K)	p(bar)	v(m^3/kg)	u(kJ/kg)	s(kJ/kg.K)
1	300	10	0.5688	940.16	23.025	10.000	0.077968	338.29	3.1959
2	400	13.668	0.5688	1252.00	23.922	13.761	0.077968	410.66	3.4041
3	400	10	0.75731	1251.90	24.519	10.000	0.10451	411.16	3.4897

4	300	7.725	0.75731	940.07	23.622	7.209	0.10451	338.84	3.2817
		Helium		T_H = 500K			Air		
1	300	10	0.5688	940.16	23.025	10	0.077968	338.29	3.1959
2	500	17.335	0.5688	1563.9	24.618	17.515	0.077968	484.22	3.5682
3	500	10	0.94583	1563.5	25.678	10	0.13083	484.95	3.7190
4	300	5.596	0.94583	940.02	24.085	5.54	0.13083	339.18	3.3472
		Helium		T_H = 600K			Air		
1	300	10	0.5688	940.16	23.025	10	0.077968	338.29	3.1959
2	600	21.265	0.5688	1875.70	25.187	21.265	0.077968	559.66	3.7057
3	600	10	11.344	1875.10	26.624	10	0.15705	560.50	3.9093
4	300	4.494	11.344	939.99	24.464	4.462	0.15705	339.39	3.4002
		Helium		T_H = 700K			Air		
1	300	10	0.5688	940.16	23.025	10	0.077968	338.29	3.1959
2	700	24.667	0.5688	2187.5	25.667	25.013	0.077968	637.38	3.8254
3	700	10	13.229	2186.8	27.425	10	0.18323	638.26	4.0736
4	300	3.708	13.229	939.96	24.784	3.681	0.18323	339.55	3.4405
		Helium		T_H = 800K			Air		
1	300	10	0.5688	940.16	23.025	10	0.077968	338.29	3.1959
2	800	28.332	0.5688	2499.30	26.084	28.758	0.077968	717.48	3.9323
3	800	10	15.114	2498.40	28.118	10	0.20937	718.37	4.2190
4	300	3.114	15.114	939.94	25.061	3.095	0.20937	339.67	3.4836

The results represented in Table 3 indicate the function of high and low cycle temperatures T_H(K), T_L(K), as well as reference cycle pressure, denoted as Ref_pres (bar) and consist of input cycle heat, $q_i = q_{i12+23}$ (kJ/kg); regenerated heat, q_{reg34}; external heat added to the cycle, $q_{iext} = q_i - q_{reg34}$; output cycle heat, $q_o = q_{o34+41}$ (kJ/kg); expansion work w_{o23} due to added heat, $w_{oexp} = w_{o23}$ (kJ/kg); contraction work w_{o41} due to extracted heat, $w_{ocont} = w_{o41}$ (kJ/kg); net useful work ($w_{oexp} + w_{ocont}$) = ($w_{o23} + w_{o41}$); w_n = (kJ/kg); cycle regeneration factor = q_{reg34}/q_{o34}, RF (%); regenerative cycle thermal efficiency, η_{th_reg} (%); cycle thermal efficiency, η_{th} (%); thermal efficiency of the expansion process, η_{th_exp} (%); thermal efficiency of the contraction process, η_{th_cont} (%); ratio (w_{o41}/w_{23}); minimum cycle pressure, Min_p (bar); and maximum cycle pressure, Max_p (bar).

Table 3. Performance results of the cycle case studies corresponding to the cycle depicted in Figure 2 for helium and air as working fluids based on data from the case studies depicted in Table 2, where a regeneration factor (RF) of 0.5 or 50% has been assumed.

	Helium				
T_H(K)	800.0	700.0	600.0	500.0	400.0
$q_i = q_{i12+23}$(kJ/kg)	3186.3	2477.9	1797.7	1153.7	550.6
q_{reg34}	779.23	623.42	467.56	311.74	311.83
$q_{iext} = q_i - q_{reg34}$	2407.11	1854.52	1330.19	842.00	238.81
$q_o = q_{o34+41}$(kJ/kg)	2169.3	1774.5	1366.8	941.5	490.9
$w_{oexp} = w_{o23}$(kJ/kg)	1627.2	1230.6	862.2	530.0	238.8
$w_{ocont} = w_{o41}$(kJ/kg)	610.8	527.7	431.7	318.0	179.1

w_n=(kJ/kg)	2238.0	1758.3	1293.9	848.0	417.9
RF(%)	0.5	0.5	0.5	0.5	0,5
η_{th_reg} (%)	93.0	94.8	97.3	100.7	105.87
η_{th}(%)	70.2	71.0	72.0	73.5	75.9
η_{th_exp} (%)	43.4	45.9	48.0	49.7	51.1
η_{th_cont} (%)	28.2	29.7	31.6	33.8	36.5
Ratio(w_{o41}/w_{23})	0.38	0.43	0.50	0.60	0.75
Min_p(bar)	3.11	3.71	4.49	5.60	7.25
P_{ref}=Ref_p(bar)	10.00	10.00	10.00	10.00	10.00
Max_p(bar)	28.33	24.67	21.27	17.34	13.67
	Air				
T_H(K)	800.0	700.0	600.0	500.0	400.0
$q_i = q_{i12+23}$(kJ/kg)	608.6	472.8	343.5	221.3	106.6
q_{reg34}	151.48	119.48	88.44	58.31	28.93
$q_{iext} = q_i - q_{reg34}$	457.07	353.35	255.09	163.02	77.68
$q_o = q_{o34+41}$(kJ/kg)	465.0	373.4	282.4	191.2	98.1
$w_{oexp} = w_{o23}$(kJ/kg)	229.4	173.7	122.2	75.4	34.2
$w_{ocont} = w_{o41}$(kJ/kg)	86.3	74.7	61.3	45.4	25.7
w_n=(kJ/kg)	315.7	248.5	183.5	120.8	60.0
RF(%)	0.4	0.4	0.4	0.4	0.4
η_{th_reg} (%)	69.1	70.3	71.9	74.1	77.2
η_{th}(%)	51.9	52.5	53.4	54.6	56.3
η_{th_exp} (%)	32.1	34.1	35.6	36.7	37.7
η_{th_cont} (%)	18.6	20.0	21.7	23.7	26.2
Ratio(w_{o41}/w_{23})	0.38	0.43	0.50	0.60	0.75
Min_p(bar)	3.10	3.68	4.46	5.56	7.21
P_{ref}=Ref_p(bar)	10.00	10.00	10.00	10.00	10.00
Max_p(bar)	28.76	25.01	21.27	17.52	13.76

The results depicted in Table 3 regarding the thermal efficiencies (regenerative and non-regenerative cycles) have been computed assuming that the regenerated heat undergoes heat losses approaching 50% of the available heat. This is common if heat regeneration occurs directly during the same cycle and in the same PU. Such a drawback can be solved by using cascade coupling. The next section deals with this topic.

2.10 Self-powered power plant based on regenerative PUs coupled in cascade

According to the previous section, the case studies based on heat regeneration exceed 100% thermal efficiency when the high temperature of each cycle approaches low values between 600 and 400 K. In such cases, it has been assumed that the studied system is reversible. However, in practice, some irreversibility or losses are unavoidable in all real cases. This concept has been discussed by Ramon Ferreiro (2018) [26]. Such a concept is considered an absolutely disruptive phenomenon that deserves a more detailed and meticulous study to determine to what extent the thermal and mechanical irreversibility or losses cancel the superiority of thermal efficiency. It is evident that among the conditions necessary to satisfy the characteristic of operating without external heat input, it must greatly exceed the Carnot factor. This condition has already been demonstrated through references R. Ferreiro et al. [27], R. Ferreiro[28]. The present section clarifies this issue.

According to this phenomenon, it should be inferred that the thermal machine under investigation can operate without thermal energy from an external heat source. This means that this machine can provide useful work without creating or generating energy (which is impossible) and without providing heat from the outside. This means that it is self-sufficient, with certain limitations.

Given the expectation of improving heat regeneration techniques, as well as the goal of avoiding losses, a case study is proposed in which heat regeneration factors reach between 60% and 90% and the losses factor (LF) or in which irreversibility varies between 20% in the worst cases and 5% in the best cases.

A simplified scheme of such a self-powered power plant based on regenerative PUs coupled in a cascade is depicted in Figure 7. The figure depicts a power plant composed of several PUs coupled in a cascade that can recover as much released low-grade heat as possible. The recovered low-grade heat is subsequently used as a heat source for the downstream PUs, which operate with the lowest temperatures.

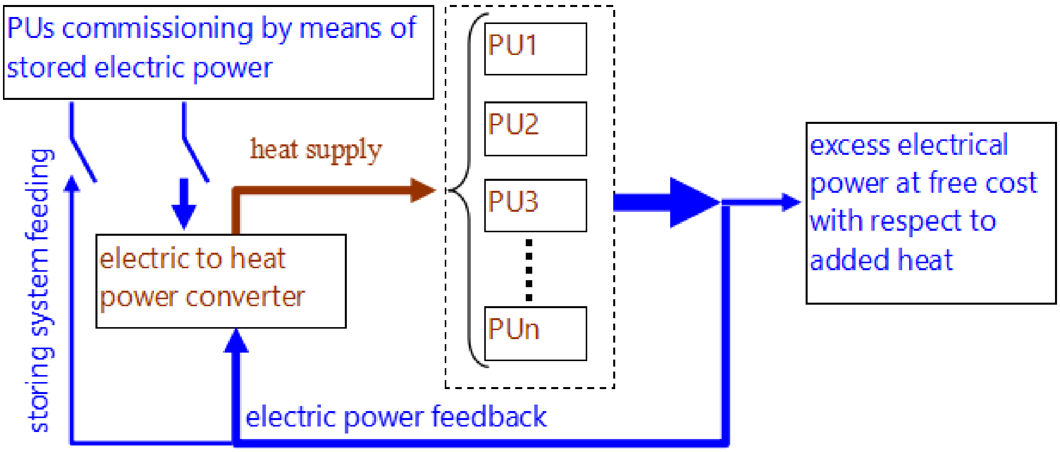

Figure 7. Self-powered power plant based on regenerative cascaded PUs powered with internal heat obtained from the output electric power.

Table 4. Analysis of the irreversible thermal efficiency study carried out in the cases of no losses (0%) and 20% losses to compare its consequences for the cases of regenerative thermal efficiencies between 0.6 and 0.9

	LF%=0; RF(0.6-0.9); He				LF%=0; RF(0.6-0.9); Air			
TH	0.6	0.7	0.8	0.9	0.6	0.7	0.8	0.9
400	115.00	125.70	138.80	154.50	94.90	107.10	123.00	144.50
500	108.00	118.20	129.50	143.10	90.20	101.30	115.40	134.00
600	104.60	113.20	123.30	135.00	87.00	97.20	110.10	126.90
700	101.60	109.50	118.80	129.70	84.60	94.20	106.20	121.80
800	99.40	106.80	115.40	125.50	82.80	91.90	103.30	117.90
	LF%=20; RF(0.6-0.9); He				LF%=20; RF(0.6-0.9); Air			
TH	0.6	0.7	0.8	0.9	0.6	0.7	0.8	0.9
400	92.00	100.56	111.04	123.60	75.92	85.68	98.40	115.60
500	86.40	94.56	103.60	114.48	72.16	81.04	92.32	107.20
600	83.68	90.56	98.64	108.00	69.60	77.76	88.08	101.52
700	81.28	87.60	95.04	103.76	67.68	75.36	84.96	97.44
800	79.52	85.44	92.32	100.40	66.24	73.52	82.64	94.32

Table 5. Percentage (%) of net self-energy for 20% losses, 15% losses, 10% losses, and 5% losses, where ηreg-100% = net self-energy relative to regenerative efficiency of 100%

LF%=20; % auto energy; RF(0.6-0.9); He					LF%=20; % auto energy; RF(0.6-0.9); Air			
TH	0.6	0.7	0.8	0.9	0.6	0.7	0.8	0.9
400	-8.00	0.56	11.04	23.60	-24.08	-14.32	-1.60	15.60
500	-13.60	-5.44	3.60	14.48	-27.84	-18.96	-7.68	7.20
600	-16.32	-9.44	-1.36	8.00	-30.40	-22.24	-11.92	1.52
700	-18.72	-12.40	-4.96	3.76	-32.32	-24.64	-15.04	-2.56
800	-20.48	-14.56	-7.68	0.40	-33.76	-26.48	-17.36	-5.68
LF= 15%; % auto energy; RF(0.6-0,9); He					LF= 15%; % auto energy; RF(0.6-0,9); Air			
TH(K)	0.6	0.7	0.8	0.9	0.6	0.7	0.8	0.9
400	-2.25	6.85	17,98	31.33	-19.34	-8.97	4.55	22.83
500	-8.20	0.47	10,08	21.64	-23.33	-13.90	-1.91	13.90
600	-11.09	-3.78	4.81	14.75	-26.05	-17.38	-6.42	7.86
700	-13.64	-6.93	0.98	10.25	-28.09	-19.93	-9.73	3.53
800	-15.51	-9.22	-1.91	6.68	-29.62	-21.89	-12.20	0.22
LF= 10%; % auto energy;; RF(0.6-0,9); He					LF= 10%; % auto energy;; RF(0.6-0,9); Air			
TH(K)	0.6	0.7	0.8	0.9	0.6	0.7	0.8	0.9
400	3.50	13.13	24.92	39.05	-14.59	-3.61	10.70	30.05
500	-2.80	6.38	16.55	28.79	-18.82	-8.83	3.86	20.60
600	-5.86	1.88	10.97	21.50	-21.70	-12.52	-0.91	14.21
700	-8.56	-1.45	6.92	16.73	-23.86	-15.22	-4.42	9.62
800	-10.54	-3.88	3.86	12.95	-25.48	-17.29	-7.03	6.11
LF= 5%; % auto energy; RF(0.6-0,9); He					LF= 5%; % auto energy; RF(0.6-0,9); Air			
TH(K)	0.6	0.7	0.8	0.9	0.6	0.7	0.8	0.9
400	9.25	19.42	31.86	46.78	-9.85	1.75	16.85	37.28
500	2.60	12.29	23.03	35.95	-14.31	-3.77	9.63	27.30
600	-0.63	7.54	17.14	28.25	-17.35	-7.66	4.60	20.56
700	-3.48	4.03	12.86	23.22	-19.63	-10.51	0.89	15.71
800	-5.57	1.46	9.63	19.23	-21.34	-12.70	-1.86	12.01

Efficiencies or net positive performance can be obtained without external heat input to differentiate ranges of heat regeneration factors and ranges of irreversibility or losses (both thermal and mechanical). Conclusive data are presented in Table 5.

The results in Table 4 correspond to the cases of 0% losses (100% reversible) and 20% losses in which the regenerative thermal efficiencies change from 0.6 to 0.9. Some of the results observed in Table 4 show that for top temperatures of the cycle approaching T_H = 400 K, in the case of RF = 0.9 at 0% losses, the regenerative thermal efficiency (η_{th_reg}) approaches 154.5% (for helium as the working fluid) and 144.4% (for air as the working fluid). However, for the case of RF = 0.9 and 20% losses, the regenerative thermal efficiency approaches 124.6% (for helium as the working fluid) and 115.6% (for air as the working fluid). Thus, even with 20% losses, the regenerative thermal efficiency achieved remains relevant when the regenerative factor approaches 0.9 or 90%.

2.11 The proposed alternative cascade coupling strategy

When the aim is to recover as much heat as possible and recover low-grade-heat rejected by a thermal cycle so that it can be returned to the same cycle, it must be considered that regenerating more than 50% of the available heat through heat exchange by countercurrent forced convection heat transfer is a technological challenge.

Therefore, it seems reasonable to try to enhance the heat-regeneration techniques based on cascade coupling using the released heat in the next downstream PUs. The proposed regenerative power plant consisting of several cascaded PUs without cycle regeneration consists of recovering the cooling released heat from all upstream PUs to be fed to the last downstream PU. This way, a substantial fraction of rejected heat can be effectively converted into useful work.

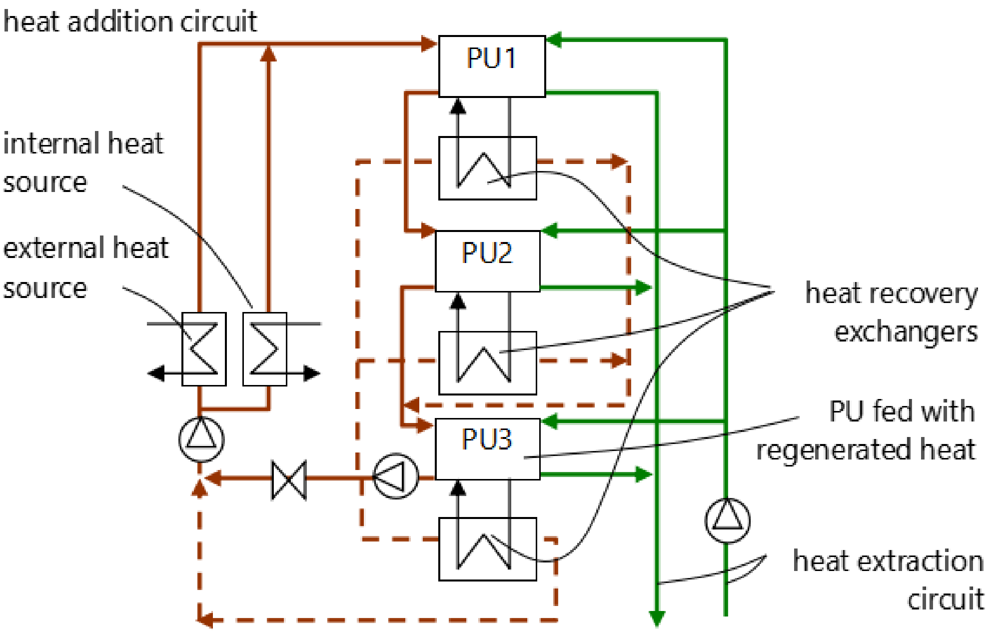

Figure 8. A cascade coupling-based power plant equipped with several PUs. The last downstream PU is fed with heat recovered from the upstream PUs, each of which comprises two RDACs, according to patent application number P202200035 and publication number ES 2 956 342 A2.

In practice, it is impossible to recover more than 50% of the available rejected heat to be regenerated by re-feeding on the same thermal cycle, even in optimal cases. It is more effective and practical to take advantage of the recovered heat to cascade the next PU downstream. In other words, non-regenerative cycles could effectively be coupled in a cascade, as depicted in Figure 8. Thus, when recovered heat is applied to downstream cascade coupling, heat regeneration may be solved by transferring the rejected heat recovered from the upstream PUs to the next downstream PU. The tentative scheme shown in Figure 8, in which the recovered heat is transferred to the next downstream PU (according to patent priority number P202200035), is important to achieve self-powered-based power plants.

3 Analysis of results

Newcomen and James Watt [4–5], among other authors, expressed the concept of reciprocating steam engines operating at atmospheric pressure in the high-pressure zone and with a vacuum in the low-pressure zone in contact with the condenser to obtain useful mechanical work due to the vacuum generated by cooling caused by heat extraction from the condenser in an open process. This concept has been applied in this research work in a closed process to model the VTVT thermal cycle.

Table 2 shows the data from the VTVT cycle analysis for helium as a working fluid. The pressure increased from 7.25 bars to 10.00 bars along the isothermal process 4-1 carried out at 300 K, while entropy decreased from 23.66 kJ/kg-K to 23.02 kJ/kg-K. When air was used as the working fluid, the pressure increased from 7.21 bars to 10.00 bars along the isothermal process 4-1 carried out at 300 K, while entropy decreased from 3.2817 to 3.196 kJ/kg-K. In the remaining cases in Table 2, the pressure and entropy changes are repeated in the same direction. Based on these results, we can make the following statement: “It is not possible to perform mechanical work by closed isothermal contraction due to a vacuum without a decrease in entropy and an increase in pressure” or “Doing mechanical work by closed isothermal contraction due to the presence of a vacuum leads to a decrease in entropy and an increase in pressure.”

The comparison of the results of each case in Table 3 as a function of the high and low temperatures of the cycle (T_H and T_L), as well as the reference pressure (p_{ref} or *Ref_p*), allowed us to evaluate the estimated operating conditions of each case of the preliminary study according to the selection of the design values of a disruptive prototype.

The most significant behavioral characteristics are as follows:

a) The ratio of useful work due to expansion and contraction increases as the top temperature of the cycle decreases, which contributes to overall efficiency.

b) The difference in thermal efficiencies between helium and air as thermal working fluids represents an increase in thermal efficiency greater than 20%. Thus, based on efficiency differences and assuming that the thermal working fluid has a durable life cycle (i.e., it is not consumed in the absence of leaks), helium is superior to air. Figures 11 (for helium as the working fluid) and 12 (for air as the working fluid) depict the thermal efficiencies of the regenerative and non-regenerative modes under a regenerative factor of 50% for helium and 40% for air. Helium is more efficient in both the regenerative and non-regenerative modes.

c) As depicted in Figures 9 (for helium) and 10 (for air), the efficiency of the work done by expansion is greater than that done by contraction when efficiency is calculated as the ratio of the work and the heat used to achieve it. In addition, such efficiencies decrease with the value of the top temperature of the cycle.

d) With regard to doing useful mechanical work associated with the decrease in entropy in the contraction-based heat engine, as the piston recedes along its return stroke, doing useful mechanical work at zero cost, the entropy of the thermal working fluid also decreases. This is similar to James Watt's steam engine, which operated at atmospheric pressure in the zone of high pressure and a vacuum in the zone of low pressure.

e) In general, as the top temperature of the thermal cycle decreases from 800 K to 400 K, the thermal efficiency of the cycle increases. This is contrary to conventional thermal cycles (which only perform expansion work). This tendency was observed regardless of whether helium or air was used as the thermal working fluid.

f) The heat regeneration factors (RFs) assumed in the case study (shown in Table 3) are 0.5 (50%) and 0.4 (40%), respectively, when helium and air were used as working fluids. When subjected to these values, the regenerative thermal efficiency for helium is 105.87. This means that the net output work of a regenerative cycle operating with helium undergoing a regeneration factor of 50% exceeds the amount of heat added: 105.87%-100% = 5.87%. That is, the amount of useful work delivered exceeds the amount of heat added to the cycle.

Table 5 shows the percentage of net self-energy (η_{reg}-100%) as a function of losses ranging from LF = 5% to LF = 20% for regenerative factors ranging from 0.6 to 0.9. Table 6 illustrates the case in which the thermal working fluid is helium and air, with LF = 10%, RF = 0.7, and T_L = 400 K (also represented in Table 5). The resulting self-power is 13.13%. Using air as a thermal working fluid in the same case yields a self-energy of -3.61%. Thus, when air is used as the working fluid, there is a deficit of 3.61% to obtain 100% of the added external heat. However, when helium is used as the thermal working fluid, there is excess energy of 13.1% over 100% of the added external heat. Based on the regeneration structure depicted in Figure 8, using helium as the thermal working fluid can generate an excess self-conversion of heat to electrical power of more than 25%.

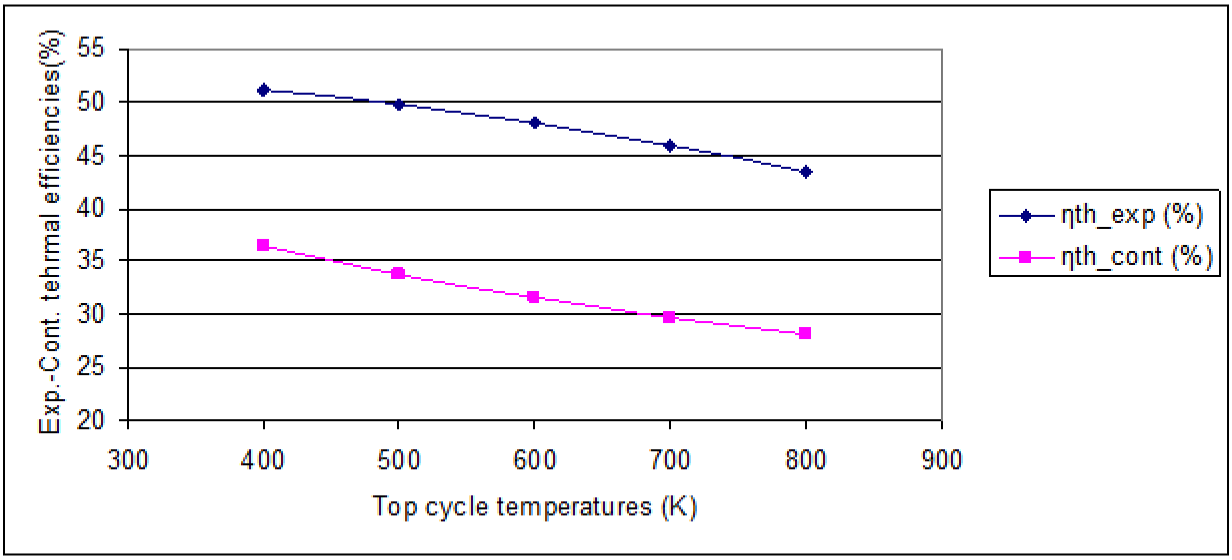

Figure 9. Expansion and contraction thermal efficiencies for helium as a working fluid

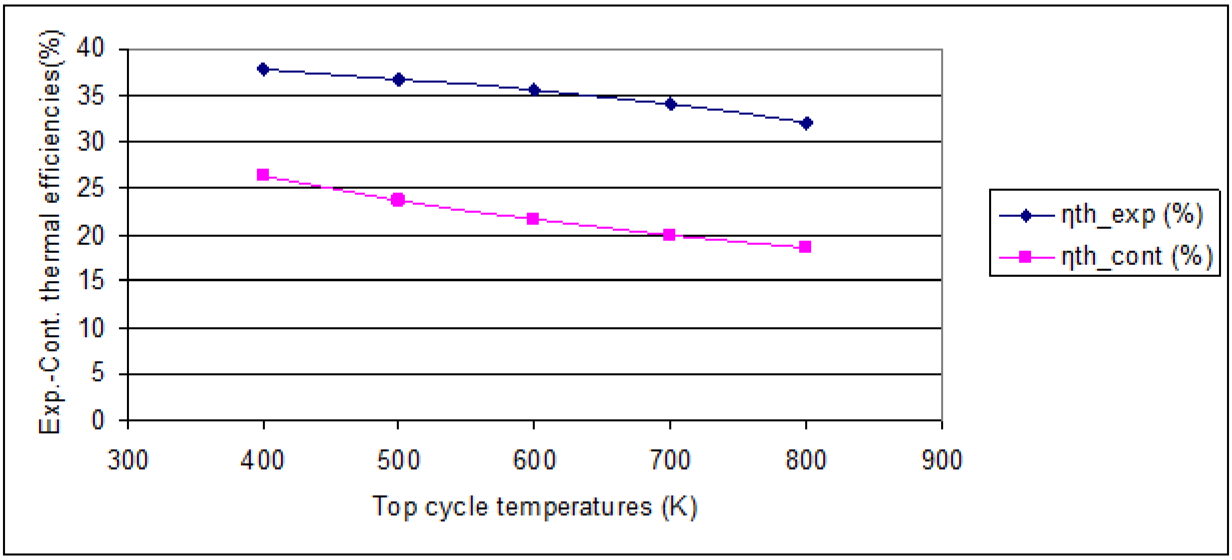

Figure 10. Expansion and contraction thermal efficiencies for air as a working fluid

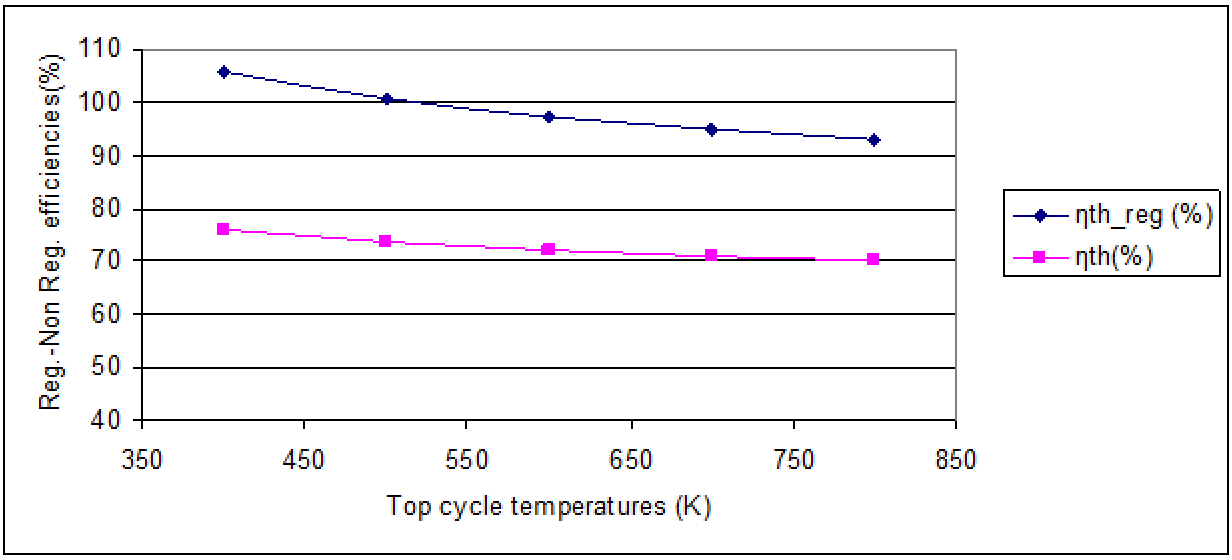

Figure 11. Regenerative and non-regenerative thermal efficiencies (%) for helium as a working fluid with a regeneration factor (RF) of 0.5 or 50%

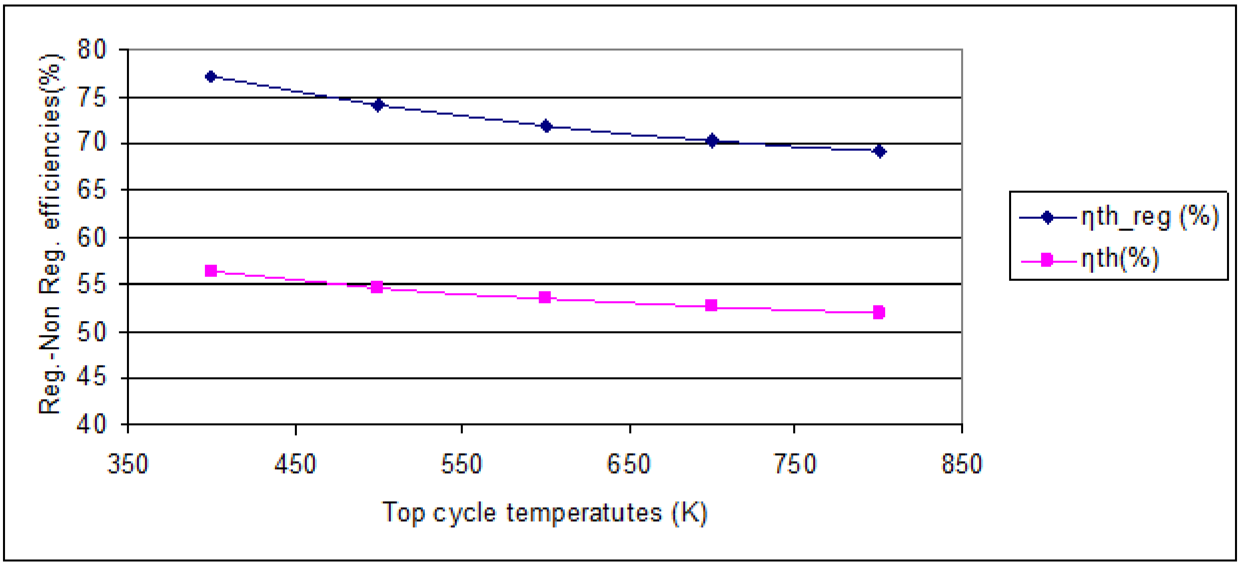

Figure 12. Regenerative and non-regenerative thermal efficiencies (%) for air as working fluid with a regeneration factor (RF) of 0.4 or 40%

4. Conclusions

This research aimed to integrate strictly isothermal closed processes within thermal cycles characterized by doing work via closed contraction processes by extracting heat at free cost and using it in a preliminary study to implement a disruptive thermal engine prototype.

According to the results, a design aimed at reducing irreversibility or any type of loss due to heat transfer and mechanical losses could be used in a prototype capable of providing net electrical power without needing external heat. Based on the cycle modeling procedure, the hypothetical concept of achieving self-power (i.e., the ability to convert heat into work without the additional external input of heat) is not related to the concept of free energy since no thermodynamic law is violated despite the achievement of disruptive and unexpected results.

The results provide abundant useful information about prototyping tasks based on the combination of various design options related to the requirements of efficiency, operating temperature, reference operating pressure, and the verification of the experimental results compared with expectations based on experimental observations and consistent theoretical estimates.

The estimates obtained based on loss factors due to irreversibility and regeneration factors assumed in the analysis of the results provide a guide for developing a prototype capable of regenerating an important fraction of low-grade heat. Such a prototype can generate an excess ratio of self-conversion of heat to electrical power of greater than 25%. This extraordinary result confirms the technical viability of real machines that exhibit the ability to provide more energy than they use, that is, second-class perpetual motion machines.

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