ISOTHERMAL GAS COMPRESSION AND EXPANSION PROCESSES BY CONTROLLING ACTUATORS

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ABSTRACT

The authors of the paper propose not only a revolutionary apparatus for the compression/expansion of gases and vapors, but also a new way of approaching energy problems in current thermodynamic installations, emphasizing the determining role of treating the exergetic aspects of the processes. The **isothermalizer** is made available to users, a particularly powerful tool capable of bringing innovative solutions to many of the concerns of researchers determined to create a cleaner planet. Designers and users of thermodynamic systems now have the opportunity to choose, at any time, its energy and exergetic efficiency and the price they have to pay to achieve the proposed objective. Thus, in a gas compression problem (similarly, in the problems of their transport and storage, liquefaction, energy storage through the CAES process, etc.), the energy efficiency is established precisely, by choosing the isothermal velocity of the gas (so of the isothermal temperature), and the power density, by choosing the total surface area of the thermal sponge, as well as the flow rate and the mode of distribution of the cooling agent.

Keywords: isothermal transformation, finite temperature differences, energy efficiency, exergy efficiency, irreversible transformations, entropy generation

1. INTRODUCTION

The second law of thermodynamics states that energy has both quantity and quality, and that all real processes occur in the direction of decreasing quality of energy, in the sense that thermal energy at a higher temperature is degraded when transferred to a body at a lower temperature (Dincer and Cengel, 2001). Finite-time thermodynamics (F.T.T). appeared for the first time in an article by Curzon and Ahlborn (1975), where they observed that during a finite power cycle, the efficiency corresponding to the maximum power becomes lower than the Carnot limit (Feidt, 2009) and were among the first to calculate how the efficiency of real processes is limited by finite rate constraints. Already in 1948, Tolman and Fine concluded that an increase in the internal entropy of a system reduces the available exergy. The principle of minimum entropy generation (EMG) or the rate of entropy production was considered by Bejan (1996, 2016) of the greatest interest and regarded as an extremely useful engineering tool. According to the Gouy-Stodola theorem, in a system, the exergy destroyed is proportional to the total rate of entropy generation (Tondeur and Kvaalent, 1987) If engineering systems and their components are to function so that exergy destruction is minimized, their design must begin with minimizing entropy generation. To minimize the irreversibility of a newly designed system, the analyst must use the relationships between temperature differences and heat transfer rates, as well as between pressure differences and mass flow rates. The analyst must rely on heat transfer and fluid mechanics principles. Only by varying one or more of the physical characteristics of the system can the analyst bring the design closer to the operation characterized by minimum entropy generation subject to finite size and finite time constraints (Bejan, 1996).

In practical applications, it is necessary to obtain transformations as close as possible to those with an isentropic index (for complete conversion, without energy changes with the environment, of thermal energy into mechanical energy and vice versa), or with isothermal index. (in which the internal energy of the gas does not change). Obtaining transformations close to isentropic transformations is possible using high speeds of process deployment (to limit the duration of thermal energy exchange between the working gas and its environment). In the state of the art there are different types of devices which approach this desire. On the contrary, a transformation close to the isothermal, with a polytropic index close to the unit value, can only be obtained if the majority of the mechanical energy introduced instantly into the system by the moving parts and absorbed by the working gas in the form of thermal energy is immediately released into its environment, to be evacuated outside the enclosures. The time interval affected by this heat transfer must be sufficiently large (therefore, at lower speeds of the moving parts, which implies a reduction in power density), or the overall heat transfer coefficient between the gas and its ambient environment must be sufficiently large (which implies a large heat transfer surface, therefore higher consumption of materials) (Török et al, 2012, 2013, 2024).

2. QUASI-ISOTHERMAL COMPRESSORS

Reducing the polytropic index of compression and expansion processes was and remains a priority objective of the design of quasi-isothermal devices in the state of the art. The research carried out studied several procedures, and some of them were also implemented: the use of materials having a heat transfer coefficient between the gas and the component parts of the compressor as high as possible; increase in heat transfer surfaces by introducing additional solid components into the device structure (Heidari et al, 2014, US 2002, IT 2005); the introduction, during processing, of certain lubricants, or liquid drops in suspension, or sprayed in intermittent jets (Coney et al, 2002, US 2011, 2013); introduction of aqueous foam (Patil, 2019), the introduction of metal inserts in the case of liquid piston devices (US 2009, 2010, 2013); the use of special enclosure configurations (Saadat and Li, 2015); recovery of exhausted thermal energy (US 2007); mechanical control of piston speed (RO 2013 a, b); control of the flow rate of the working fluid, in the case of the liquid piston (Shirazi et al, 2013); the introduction of metal inserts in the case of liquid piston devices (Caleb et al, 2009; Rice et al, 2017; Saadat et al, 2012). In the state of the art, thanks to these processes, acceptable performances have been obtained in a fairly limited series of applications: for certain liquid piston compressors and expanders, including ionic ones, for certain types of reciprocating compressors and expanders with solid pistons, screw and scroll compressors. These performances are limited by the fact that the effect of these processes is felt only in the initial phase of the thermodynamic process, but after the advancement of the piston, in the phases where the heat introduced from the outside increases exponentially (Török et al 2012, 2013), due to the increase in the mechanical power of the piston, the heat exchange surfaces decrease. Furthermore, these processes do not modify the polytropic character of the transformation and only have the effect of increasing the **power density**.

3. THE ENDOREVERSIBLE ISOTHERMALIZER

The heat transmitted to the gas by the piston can be evacuated only in the presence of a temperature difference ΔT between the gas and its surroundings. An ideal, perfectly isothermal compression, where the gas and its surroundings have the same temperature (ambient temperature T_{amb}) throughout the process, is only possible in the hypothetical case of a process of infinite duration. Unlike the other types of irreversibilities, this one cannot be avoideded. But, this temperature difference ΔT can remain unchanged, and the process can become **isothermal**, if at each instant, the gas transfers to its medium a quantity of heat equal to that received by the change in volume. Other types of phenomena destroying the exergy of the system can appear and become significant, depending on the characteristics of the configuration used and the speed of the process.

In our investigations, we followed the example of Rubin (1979) and we found it useful that, in a first step of the investigation, we concentrate our attention on an ideal, **endoreversible** device, neglect internal irreversibilities and consider only the thermal exchanges between the gas and its environment, exchange with finite temperature differences. The choice of the working temperature T_{iz} of the gas (and, implicitly, of the temperature difference ΔT relative to its ambient temperature T_{amb}) at which we want the isothermal transformation to take place is a compromise between the quantity of energy consumed in addition to the ideal compression (at $\Delta T= 0$ and energy efficiency $\eta_{th}=1$) and the duration of a cycle (which dictates the power consumed). Furthermore, the value of ΔT is, at any instant, proportional to the quantity of heat exchanged by the gas with its environment, the proportionality factor being variable: $C_{GT}(t) = \Sigma hi(t)Ai(t),$ where $Ai\ (t)$ and $hi(t)$ are, at each instant t of the transformation, the values of the contact surfaces and the heat transfer coefficients of the elements i of the device, which are in contact with the working gas. The transformation undergone by the gas is isothermal ($T_{iz}=ct$), if the instantaneous power $Wi(t)$ of the piston is permanently equal to the instantaneous heat $Qi(t)$ transferred to the medium by the working gas.

We will call **isothermalizer** (**isocompressor/densifier** in the case of compression and **isodetenter/rarefier** in

the case of expansion), the device with cyclical operation, which is composed of one or more closed enclosures, with monotonically variable volume, containing gas at a monotonically varying pressure, whose temperature is constant for most of the duration of a cycle. According to the fundamental theoretical studies mentioned in the first paragraph, for an imposed energy efficiency: $\eta=W_{ideal}/W_{real}$ (where W_{ideal} is the mechanical work consumed for the ideal compression of the working gas at T_{amb}), the most energy efficient strategy, by which a quantity m_g of gas of pressure p_1 and temperature T_{amb} (the same throughout the mass of the gas), located in a temperature chamber $T_{amb,}$ is brought to a pressure p_2 and a temperature T_{amb} in a time interval is a three-step process:
1. an isentropic compression of the entire quantity of gas, in a time interval $\Delta t \rightarrow 0$, up to the pressure p_3, corresponding to a working temperature T_{iz}
2. an isothermal compression, in a time interval $\Delta t=t_{iz}$, up to a pressure p_4, which corresponds to the entropy of the gas in the state (p_2, T_{amb}), during which the average temperature of the gas remains T_{iz}. The gas law requires, for this transformation, a change in the speed of the piston (isothermal trajectory). Any other compression (any other trajectory) between the state (p_3, T_{iz}) and the state (p_4, T_{iz}) is carried out either in a time interval $\Delta t > t_{iz}$, or requires higher energy consumption. In this process, the higher the T_{iz}, the lower the T_{iz} and vice versa. At high values of T_{iz}, isothermal transformations with high energy consumption are obtained.
3 an isentropic expansion of the entire quantity of gas, from the average temperature T_{iz}, to the average temperature T_{amb}, from the pressure p_4, to the pressure p_2, in a time interval $\Delta t \rightarrow 0$.

In conclusion, any positive displacement compressor can become an isocompressor, if the variation in the volume of each compressor enclosure occurs at isothermal speed, corresponding to the isothermal temperature T_{iz}. Furthermore, for any prior art compressor repeatedly performing the same compression process, the simple addition of a profiled cam (or other type of mechanical converter) in the kinematic chain, can transform the piston speed into a isothermal speed (which corresponds to a t_{iz} equal to the duration of the cycle, reduced by the times affected by the adiabatic jumps), without modifying the speed of the drive motor, this results in an energy gain of more pronounced as the final pressure is high. The three stages of the evolution of the AIA (adiabatic-isothermal-adiabatic) can be carried out inside the enclosure by the piston of the isothermalizer, by the corresponding variation of its velocity. The adiabatic jumps may occur, also outside it, in a combination of separate devices, according to **Fig.1**. For the endoreversible isothermalizer we will consider that the working gas is sucked in and discharged at the desired temperature T_{iz}.

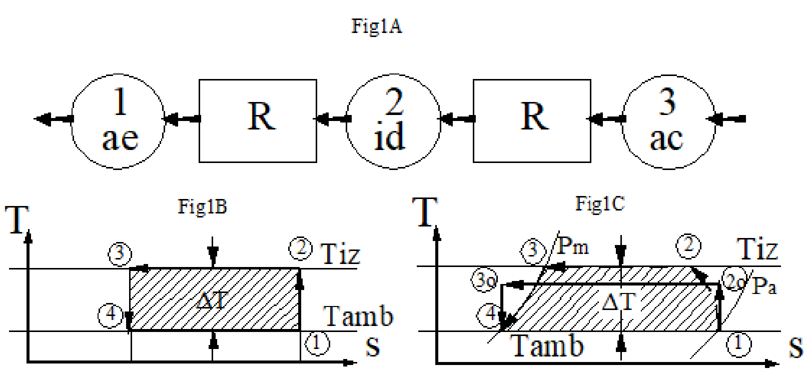


Figure 1: AIA stages. Fig.1A: 1=adiabatic expander 2=densifier, 3=adiabatic compressor, R=reservoirs
Fig.1B: T-s diagram of the evolution of the AIA. Fig.1C: T-s diagram of a polytropic-isothermal-isobaric evolution

In conclusion, for the realization of an isothermalizer, the essential condition is to ensure the volume occupied by the working gas an isothermal change rate, regardless of the process by which this volume change takes place: in the case of solid piston compressors, by the appropriate change of the speed of the piston, in the case of those with a liquid piston, by correspondingly changing the flow of liquid introduced/exhausted into/from the enclosure, in the case of those with gas pistons (enclosures with fixed volume), by correspondingly changing the flow of gas introduced/evacuated, in the case of rotating ones, with a single enclosure, by correspondingly changing the instantaneous speed of the rotor, during one cycle, and in the case of those with several enclosures, by adding a liquid piston for each of the enclosures. But, it must be added that, in the case of the majority of the compressors and quasi-isothermal expanders in the state of the art, only by this modification, if the general heat transfer coefficient C_{GT} is small, at acceptable energy efficiencies low power densities are obtained, and at higher power densities , high T_{iz} working temperatures are obtained. Therefore, it is required that the introduction of the isothermal speed be accompanied by procedures for increasing the C_{GT} and procedures for removing the heat from the apparatus enclosure through procedures taken from the state of the art, or through new, more efficient procedures. The use of these processes brings with it additional possibilities to change the volume occupied by the gas in the enclosures, therefore additional possibilities to control the isothermal speed.

3.1. Isothermal speed

The imposition of the condition to achieve an isothermal transformation leads to a first-order differential equation (isothermal velocity equation), in which the only unknown is the function $v_{iz}(t)$: the time variation, during one cycle, of the rate of change of the volume of the enclosure. The solution (analytical or numerical) of this equation, followed by the creation of an actuation system that requires the compressor actuation device to realize the motion relationship described by this function is one of the practical solutions for solving isothermal compression, indicated in a series of previous works.

Our proposed invention (Török et al, 2022, 2023) avoids solving the differential equation of motion and transfers the task of finding the isothermal velocity, to a feedback control system. Instead of determining, with an inherent margin of error, how the instantaneous power transmitted to the moving body should vary, we equipped the device with artificial intelligence and introduced a **closed-loop control device** into the system, which from information collected **over real time** by a series of transducers determines, independently of environmental parameters variations and other disturbances that may occur, the direction and the value by which the magnitude of the forces acting on the moving parts which influence the average temperature of the working gas must change at that moment. The procedure can be generalized, in order to apply it not only in the case of reciprocating machines, where the main controlled parameter is the speed of the piston, but to all volumetric devices, by introducing automatic regulation systems of the rotor angular velocity and/or of the inlet and outlet flow rates of the cooling fluids (transferring the piston function to these fluids). In addition, the invention also describes a gas piston isothermalizer, which isothermally compresses gases in a closed enclosure at constant volume, by controlling the flow rate of gas introduced, simultaneously with controlling the inlet and outlet flow rates of a heat transfer fluid. As in the case of rotary isothermalizers, carrying out the isothermal transformation is done by simultaneously adjusting several parameters.

These regulation systems become the main component of all isothermalisers. The shorter the response time and the smaller the difference between the actual and desired value of the process variable, the closer the transformations they control are to an isotherm. The use of real-time adjustment devices does not exclude a theoretical determination of the optimal trajectory and the use of this results during the design phase of the actuators, to reduce the error signal (SP-PV error) and increase the speed of the response of the controller. It should be noted that with this type of compressor configuration, even if the compression ratio is very high, it becomes useless to divide the compression process into several stages, with the aim of limiting the final temperature. However, it may prove useful to carry out compression in stages, without intermediate exchangers, for a more efficient distribution of the available power. Let us also point out that all the previous statements, made in the case of gas densifiers, remain perfectly valid in the case of rarefiators, with the exception of the fact that in their case the gas is colder than the thermal sponge and its constituent elements, therefore, the direction of heat transfer is from these to the gas.

As an example, in **Fig.2A** is shown the principle diagram of such a type of installation, carried out on the configuration of a solid piston compressor of the state of the art. In the figure, we have shown in solid lines the heat transfer liquid circuit, in broken lines the signals transmitted by the transducers, and in dotted lines (ACAD 10W100) the commands transmitted by the controller to the executive elements. In this case, the system is managed by the processor 12.4, DC, which constantly compares the pressure measured by the pressure sensor 12.5, with that corresponding (according to the gas law) to the working temperature T_{iz} and the volume Vi at that moment. The calculation of the instantaneous volume is based on the position Li of the piston at this instant, provided by the position transducer L, from which the volume occupied by the liquid is subtracted (calculated on the basis of the signals provided by the two flow meters). Depending on the result, the DC transmits the appropriate command to the drive system 12.3 (here based on a linear motor), setting in motion the piston 12.2, which moves in the cylinder 12.1. The linear rotating field is created by a cylindrical set of electrical coils and a set of permanent magnets (or a second set of coils) positioned on the piston rod, which moves along the axis of the cylinder. We have avoided the use of the classic system, from the state of the art, in which the return movement of the piston is ensured by a system of springs, a system whose optimal operation is conditioned by its resonance coefficient. Therefore, the return movement of the piston will be ensured by reversing the magnetic field, with a speed controlled by the controller, to ensure the greatest possible efficiency.

The excess heat from the gas is absorbed by a thermal sponge 12.6, deformable under the action of the piston and other constituent elements of the densifier, to be transferred to a flow of heat transfer liquid, which includes the variable flow pump 7P, the HE heat exchanger, 8.6A sprinklers mounted on the walls of the enclosure, 7v servo valves and 7f flow meters (one for the inlet flow, the other for the outlet flow). The modification of the heat flow taken up by the agent is done, simultaneously with the modification of the volume of the enclosure, by commands from the controller to the two 7v servovalves and possibly to the pump drive motor. Operation with maximum efficiency is ensured by the correlation of the two isothermal temperature regulation processes. Achieving these optimal $v_{iz}(t)$ trajectories requires the use of regulation systems that are as sensitive as possible and with the shortest possible reaction time, based, for example, on linear electric motors, on direct current electric motors with permanent magnets, or with a wound rotor , synchronous motors powered by a frequency converter, DC servomotors for valve actuation, stepper motors, etc. The cost of these types of drives, especially for high-flow and high-power-density isothermalizers, is quite high, and the maintenance costs are not negligible. For this reason, drives that use alternating current motors, single-phase or three-phase, asynchronous or synchronous, become preferable. The best compromise between the price and the performance of these drive systems is obtained by using in the design phase, successively, both solutions for driving the moving parts: the prototype of the isothermalizer designed for a specific application will be equipped with all the elements of the drive system tuning: the signal transducers, the actuators and the controller with the algorithm corresponding to the respective prototype and application.

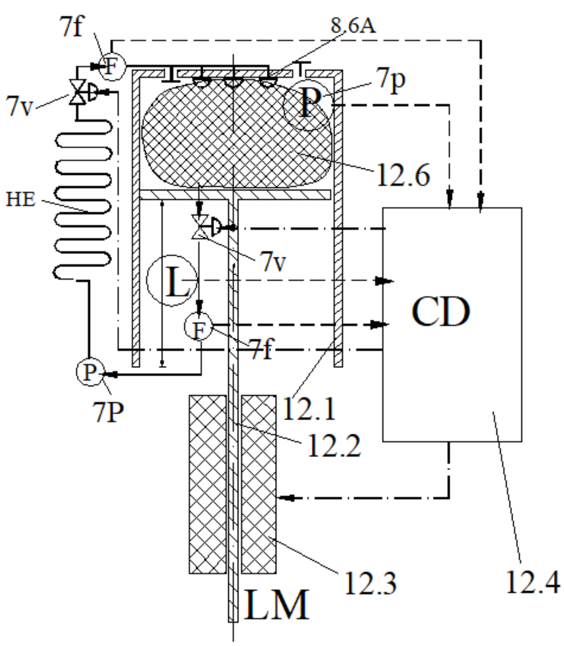
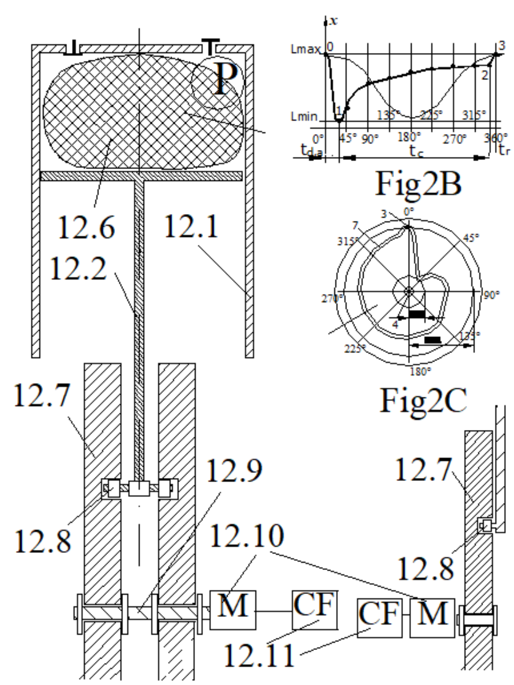


Figure 2: A: Isothermalizer with linear motor



B: Isothermalizer driven by constant speed motor and profiled channel disc

Then the system is tested on the test bench, being able to experiment for a number of different values of T_{iz}, therefore for different average piston speeds and different energy efficiency values. Depending on the results obtained, the optimal trajectory and the signal to be transmitted to the drive system are chosen. Following these experiments, the decision can be made to increase the absorbent surface of the thermal sponge, or to increase the flow rate of the heat transfer fluid. From the curve describing, for a cycle, the position of the piston as a function of time *L(t),* that is to say the isothermal trajectory, the dimensions and shape of the driving cam and the characteristics of its guide spring, or the trajectory of the profiled channel on one face of the drive disc, or the configuration of another mechanism chosen to achieve this trajectory, devices which, associated with a constant speed drive motor, will replace, for mass-produced isothermalizers, the DC motor and its automatic speed control system. **Fig.2.B** shows a solid piston isothermalizer, the piston being actuated by means of a disc 12.7 through the profiled channel of which the bearings 12.8 pass. The discs are driven by the axis 12.9 driven by the alternating current motor 12.10. In Fig.2B is represented, as a function of time, the shape of a curve $v_{iz}(t)$, and in Fig.2C, the way in which this curve is transformed into a closed curve (the profiled channel) whose trajectory depends on the angle of rotation of the motor. The mounting variant with a single disc, with the execution of the profiled channel on one side and with the coupling system to the motor shaft executed on the opposite side, (sketch on the left) is the variant that occupies the smaller volume. In the case of rotating devices which contain, for the compression/expansion of gases, a single working enclosure (for example, compressors with a single blade in the rotor), the maintenance of the temperature between two close limits is done by varying the angular speed of the rotor. in the time allocated to each compression cycle, and in the case of those comprising several enclosures of different volumes, each in different compression phases (lobe compressors, gear compressors, screw compressors, scroll compressors, etc.), the speed of the rotor is kept constant, but in each enclosure the flow of cooling/heating liquid and the flow of the liquid acting as a piston vary.

3.2. The thermal sponge

Acceleration of heat transfer between the gas and its environmental surrounding can be obtained by introducing a thermal sponge into the device responsible for the isothermal transformation. The type of sponge that the invention analyzed uses most often, due to its simplicity and effectiveness, is a deformable thermal sponge, mounted between the internal face of the piston (solid or liquid) and the cover of the cylinder, which deforms under the action of the mobile parts, its component elements having a surface in contact with the gas, as large as possible, but constant throughout the transformation. Due to its much greater heat capacity than that of the working gas, the total volume of elements which compose an effective thermal sponge can be very small, but it is important that this volume be distributed efficiently over a large area, throughout the gas volume. As a result, power density can increase by several orders of magnitude. In isothermalizers with a high compression/expansion ratio, the deformable thermal sponge can be supplemented by a non-deformable thermal sponge with a very large absorption surface, mounted in a small area outside the field of action of the piston. Such a deformable sponge is made up of one or more solid constituent elements (in many configurations, a liquid component is also introduced, in fixed quantity) of variable volume and/or position. The solid components of the thermal sponge have the total surface area *Si* (**Fig.3**), which is in direct contact with the working gas,

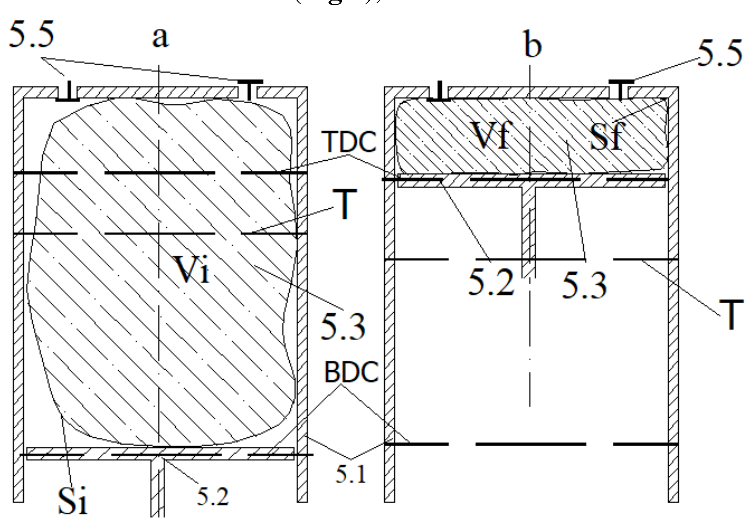
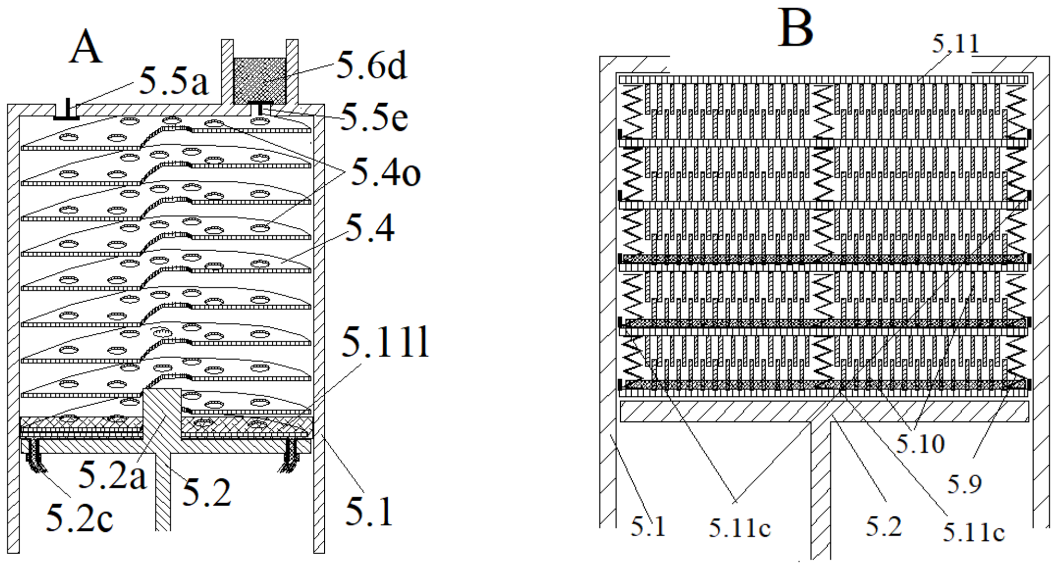


Figure 3. Deformable thermal sponge. 5.1: cylinder, 5.2: solid piston, 5.3: deformable sponge
a: with the piston at BDC, b: with the piston at TDC

approximately the same throughout the compression phase, and their degree of deformation is constantly controlled by the position of the piston, a property ensured by the elasticity of certain of its constituent elements, or by kinematic devices controlled by (or in correlation with) the movement of the piston. Liquid components of a thermal sponge installed in reciprocating devices can also play the role of a transport agent of excess thermal energy, if during the discharge and admission phases they are replaced by cooled components, or they can assume the role of a liquid piston, if during the thermodynamic transformation, the quantity of liquid introduced is different from that discharged. **Fig.4** shows two examples of deformable thermal sponges: Fig.A, with helical spring 5.4 with rectangular section and Fig.B, with horizontal flat plates 5.11 with vertical fins 5.10. The inlet and outlet flow rates of the liquid agent are determined by a regulator, by commands sent to the servomotors that actuate the hydraulic valves on the inlet and outlet pipes, and by commands that are correlated with the commands sent by the piston actuator regulator. Higher efficiencies are obtained when all thermal energy exchanges take place with optimal temperature differences (often, the smallest possible). The replacement of compressors and expaanders of the state of the art, by isothermalizers according to the invention, brings significant advantages, by reducing the energy consumed, by increasing the flow rates, by reducing the size of the tanks, by significantly reducing chemical and thermal pollution, etc.



Thermal sponge configurations.
Fig.4A: helical spring 5.4 with rectangular section, Fig.4B: horizontal flat plates 5.11 with vertical fins 5.10

4. CONCLUSIONS

Equipping devices designed for the compression and expansion of gases and vapors with real-time control systems, or with systems controlled by a processor, allows any trajectory for *T(L)* (variation of temperature as a function of piston position) to be obtained, including the **isothermal trajectory**, if the response performance of the regulators is adequate and if the heat sources/sinks with which the mechanical energy supplier/consumer interacts meet certain conditions. The time required to achieve this trajectory depends on the constructive configuration of the device and how the device reacts to the demands of the cooling system. Devices equipped with a thermal sponge with a large absorption surface and efficient cooling systems can compress/expand, even at speeds comparable to those of state-of-the-art polytropic devices, large volumes of gas without exceeding a small temperature difference.

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