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Isothermal gas compression and expansion processes by controlling actuators

Arpad TÖRÖK^{1*}, Michel FEIDT²

Abstract. This work is part of a program started several years ago, carried out until now by the publication of a series of articles [1, 2] and by the filing of several patent applications [3-5], program through which the authors wish to contribute to the fight for environmental protection, by proposing solutions for better use of available energy sources and for the reduction of chemical and, last but not least, thermal pollution of the environment. A first conclusion of this program is that at this stage of the state of the art, due to the fact that they were designed at a time when energy was cheap and the harmful effects on the environment were not yet so obvious, the industries and residential thermotechnical installations currently used, producing significant quantities of entropy, still pay a heavy price in lost exergy. Therefore, an important avenue of attack to achieve the proposed objectives consists of the modification, or progressive replacement, of the usual technical processes in existing thermodynamic installations, even those that use "green" energy, by new processes that consume less exergy. The most important changes, applicable to these installations, must aim at a favorable reduction in temperature differences between the elements of the installation and the environment in which they are located, as well as the use of thermodynamic transformations with minimal generation of entropy. In light of these observations, our research paid particular attention to isothermal transformations.

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

1. Introduction

Nowadays, the phenomena of environmental degradation, both from the point of view of its composition and its temperature, are becoming more and more worrying, which is strongly reflected in the concerns of civil society and those of the political environment, consequently they channel an increasingly important part of scientific and technical activity. The fight against chemical pollution is gaining ground by increasingly replacing fossil fuels with energy from non-polluting sources, and the fight against the threat of global warming is generating more and more environmentally friendly technological installations.

But the number of industrial technological installations, chimneys of all kinds, exhaust pipes, external modules of refrigeration and air conditioning installations, etc., which release huge amounts of thermal energy into the environment, decreases at too slow a rate. Part of the

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latest research of authors has focused in particular on the search for methods to reduce this type of pollution, simultaneously with the application of solutions to maximize the available exergy. One of the solutions which meets these two wishes is to replace the polytropic thermodynamic transformations which take place in the machines and installations used, with transformations as close as possible to isothermal transformations.

The use of the term "**isothermal transformation**" has become somewhat abusive in numerous technical works in recent years describing thermodynamic transformations in real installations of the state of the art. Because an ideal isothermal transformation (with unitary energy conversion efficiency), during which the temperature of the gas remains permanently equal to that of the ambient environment ($\Delta T=0$) cannot take place, this expression is also used for transformations during which the difference ΔT between the temperature of the gas and that of the environment varies between 0 and a certain maximum value, and the average polytropic coefficient of this transformation approaches the unit value (the appreciation of the degree of proximity being quite arbitrary). For example, in the **quasi-isothermal transformations** that occur in solid piston compressors, into which coolant is injected [18], or in liquid piston compressors with metal inserts [22], for compression ratios greater than 50, for cycle times of the order of a second (or even tens of seconds), the temperature of the working gas increases by 25-30°C at the time of suction, up to 50-80°C and even more, at the time of discharge. In many of these cases, a closer to reality expression is used: "**nearly isothermal**".

In this article, we will use this syntagma only in a much more restrictive sense, used in particular from theoretical studies of Finite Time Thermodynamics, namely for transformations for which the average temperature of the gas in the control volume is uniform and constant (for real installations, variations of only a few percent around the average value are accepted). Furthermore, in accordance with the EGM minimum entropy generation principle, we will focus on transformations in which the difference ΔT between the instantaneous average temperature of the gas and the average instantaneous temperature of the medium in which the transformation takes place is constant, assuming for real installations, a difference of a few percent.

2. Theoretical bases of the isothermal transformation

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 [6]. Carnot is undoubtedly the precursor of all subsequent developments in Thermodynamics applied to machines, systems and processes. He introduced the crucial concept of cycle and can be considered the originator of the notion of efficiency. [7]. Finite-time thermodynamics (F.T.T). appeared for the first time in an article by Curzon and Ahlborn [8], where they observed that during a finite power cycle, the efficiency corresponding to the maximum power becomes lower than the Carnot limit [7] and were among the first to calculate how the efficiency of real processes is limited by finite rate constraints. Subsequently, Berry and co-workers published pioneering studies on limiting process variables for finite-time or finite-

rate processes [9]. Already in 1948, Tolman and Fine [10] concluded that an increase in the internal entropy of a system reduces the available exergy. The principle of minimum entropy generation (EMG) or entropy production rate was considered by Bejan to be of greatest interest [11, 12] and considered an extremely useful engineering tool. EGM is the method that combines in simple models the basic concepts of heat transfer, fluid mechanics and thermodynamics. These simple models are used in the optimization of real (irreversible) devices and processes, subject to finite dimension and time constraints. According to the Gouy-Stodola theorem, in a system, the destroyed exergy is proportional to the total entropy generation rate [13]. If engineering systems and their components are to operate in a way that minimizes exergy destruction, their design must begin by 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 the principles of heat transfer and fluid mechanics. By simply varying one or more physical characteristics of the system, the analyst can bring the design closer to the operation characterized by the generation of a minimum entropy subject to finite dimension and finite time constraints [11].

3. Quasi-isothermal compressors

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) [3].

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 or liquid components into the device structure [14, 15, 16]

- the introduction, during processing, of certain lubricants, or liquid drops in suspension, or sprayed in intermittent jets [17, 18, 19]
- introduction of aqueous foam [20]
- the introduction of metal inserts in the case of liquid piston devices [21-23],
- the use of special enclosure configurations [24]
- recovery of exhausted thermal energy [25]
- mechanical control of piston speed [26-27]
- control of the flow of the working fluid, in the case of the liquid piston [28-31]

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 [1, 2] (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**.

4. The endoreversible isothermalizer

An ideal, perfectly isothermal compression, in which the gas and its surrounding environment (which surrounds it, solid or liquid) have the same temperature (ambient temperature T_{amb}), throughout the process, is only possible in the hypothetical case of a process of infinite duration. The heat transmitted to the gas by the piston can only be evacuated in the presence of a temperature difference ΔT between the two environments. This type of irreversibility cannot be avoided. This temperature difference ΔT can remain unchanged, and the process can become isothermal, if at each instant of the transformation, 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 [32] 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: $C_{GT}(t) = \sum hi(t)Ai(t)$, 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, confirmed by experimental results, for an imposed energy efficiency: $\eta=W_{ideal}/W_{real}$ (where W_{ideal} is the mechanical work consumed for the isothermal 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:

- 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}

- 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} , for this being necessary to continually modify the speed of movement of the piston (isothermal trajectory). Any other compression 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.

- 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. 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 (T_{iz} being the isothermal temperature 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, or outside it, by 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} .

We have therefore reached the conclusion that, for the creation of an isothermalizer, the essential condition is to ensure the volume occupied by the working gas has an isothermal speed

of variation, whatever the process by which this variation of volume is carried out: in the case of solid piston compressors, by accordingly modifying the speed of the piston, in the case of those with liquid piston, by correspondingly modifying the flow rate of liquid introduced/evacuated into/from the enclosure, into the in the case of those with a gas piston (fixed volume enclosures), by accordingly modifying the flow rate of gas entered/expelled, in the case of rotary devices, with a single enclosure, by accordingly modifying the instantaneous speed of the rotor, during a cycle.

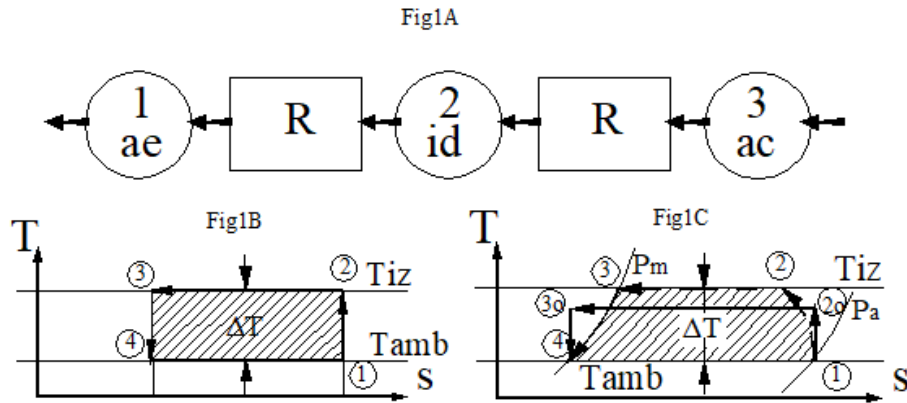


Fig.1A: AIA stages. 1=adiabatic expander 2=densifier, 3=adiabatic compressor, R=reservoirs. Fig.1B: T-s diagram of the evolution of the AIA.

Fig.1C: T-s diagram of a polytropic-isothermal-isobaric evolution

But it must be added that, in the case of the majority of quasi-isothermal compressors and expanders of the state of the art, in the case of a reduced general heat transfer coefficient C_{GT} , only thanks to this modification we obtain, at acceptable energy efficiencies, very low power densities, and at higher power densities, the T_{iz} working temperatures obtained are high. It is therefore necessary that the introduction of the isothermal speed be accompanied by procedures for increasing the C_{GT} and procedures for evacuating heat from the device enclosure, procedures taken from the state of the art or new ones more efficient procedures. The use of these processes provides additional possibilities to modify, in a controlled manner, the volume occupied by the gas in the enclosure, therefore additional possibilities for controlling the isothermal speed.

5. Isothermal speed

In the theoretical [32, 33] and practical [1-5] works mentioned previously, the imposition of the condition for carrying out an isothermal transformation led us to a first order differential equation (isothermal velocity equation), in which the only unknown is the function $v_{iz}(t)$: variation over time, during a cycle, of the speed of movement of the piston. The solution (analytical or numerical) of this equation, followed by the realization of an actuation device which requires that the compressor actuator realizes the movement relation described by this function, leads to obtaining a process in which the average temperature of the gases remains close to the value T_{iz} throughout the duration t_{iz} of the compression phase. The energy consumed

for this compression is minimal compared to any other movement relation $v(t)$ where the duration of the compression phase is equal to t_{iz} . The results obtained are used to create installations applying this process, by introducing cams into the kinematic drive chain [26, 27].

In the current state of the art, there are other theoretical studies, as well as a series of installations based on theoretical and experimental results [28-31], which propose methods to obtain with a consumption of reduced energy, compressions with a high compression ratio. These methods start from the calculation of the average values $\Sigma hi(t)Ai(t)$, or other process parameters, measured before or during the movement of the piston and modify the speed of the solid piston by mechanical devices, and in the case of liquid piston, by modifying the flow rate of the working liquid, using variable section nozzles, the modification commands being preset according to the results obtained by solving certain optimization algorithms.

The piston driving method based on the solution provided by the resolution of its equation of motion can give results in the case of low complexity installations. For example, for a cylindrical enclosure which has a diameter much larger than its length, we can consider the total heat transfer coefficient (thus the instantaneous heat transferred by the gas to its environment) constant throughout the compression (in particular, in the case of a narrow pressure range), the mechanical power supplied to the piston must therefore be constant (easily obtained by driving it by a direct current electric motor, supplied at constant voltage, equipped with a regulator which maintains the constant excitation current).

6. Obtaining the isothermal speed

Our proposed invention [3] avoids solving the differential equation of motion and transfers the task of finding the isothermal velocity $v_{iz}(t)$, 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 isothermalizers. 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 the results obtained 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 and this way of moving the piston (keeping the ΔT difference constant), 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 rarefiers, 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.

The **automatic and real-time adjustment** of the actuator power is based on the instantaneous measurement of the main parameters of the gas state using sensors of small dimensions, sufficiently sensitive and with a sufficiently fast response time. The collected signals are transmitted to a computing device, which produces an appropriate response, transmitted to the actuator. For example, if the gas temperature is measured at enough points to determine an instantaneous average temperature T_{mi} , the processor (calculation and regulation device) compares the measured value with the prescribed value T_{iz} , calculates the difference and sends to the actuator an acceleration or deceleration command to achieve equality $T_{mi} = T_{iz}$. Additionally, temperature sensors mounted in certain areas of the device can transmit signals which, after processing, cause commands to be transmitted to flow regulators on the coolant lines. In this way, an isothermal evolution can be achieved, whatever the temperature of the external environment, the temperature of the compressor components, the thermal sponge, or the heat fluids. Additionally, the collected information can be used to adjust lubricant flow rate, spray coolant flow rate, etc.

It is preferable to have a variant of the control system in which the role of gas temperature measuring transducers (slower, more difficult to mount and more sensitive to the temperature of nearby elements) is taken over by a pressure transducer. The basis of this change is the observation that at any moment, between the instantaneous values of the volume V_i , the pressure p_i and the average temperature T_{mi} , there is a well-established relationship, depending on the molecular structure of the working gas, given by the gas law. For example, if we can consider that the real gas model can be chosen for this gas, we will use the relation $p_i V_{m,i} = Z(p, V_m) R T_{mi}$, where V_m is the molar volume of the gas in the cylinder, R is the molar constant of gases, and Z is the compressibility factor (equal to 1 only for ideal gases), depending on both pressure and volume, its values being well known and tabulated. Therefore, to obtain the equality $T_{mi} = T_{iz} = \text{constant}$, there must be a direct correlation between p_i and V_i . Also, for each volume V_i there exists a well-defined position of the piston: $X_i = f(V_i)$. Consequently, a calculation relationship can be deduced: $X_i = f(p_i)$. It is preferable to choose pressure as the

control quantity, because its value is the same throughout the mass of gas and a single sensor is sufficient for its measurement. There are also simple, inexpensive and sufficiently accurate pressure transducers that give extremely fast response (e.g. piezoelectric transducers).

As an example, in **Fig.3** 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

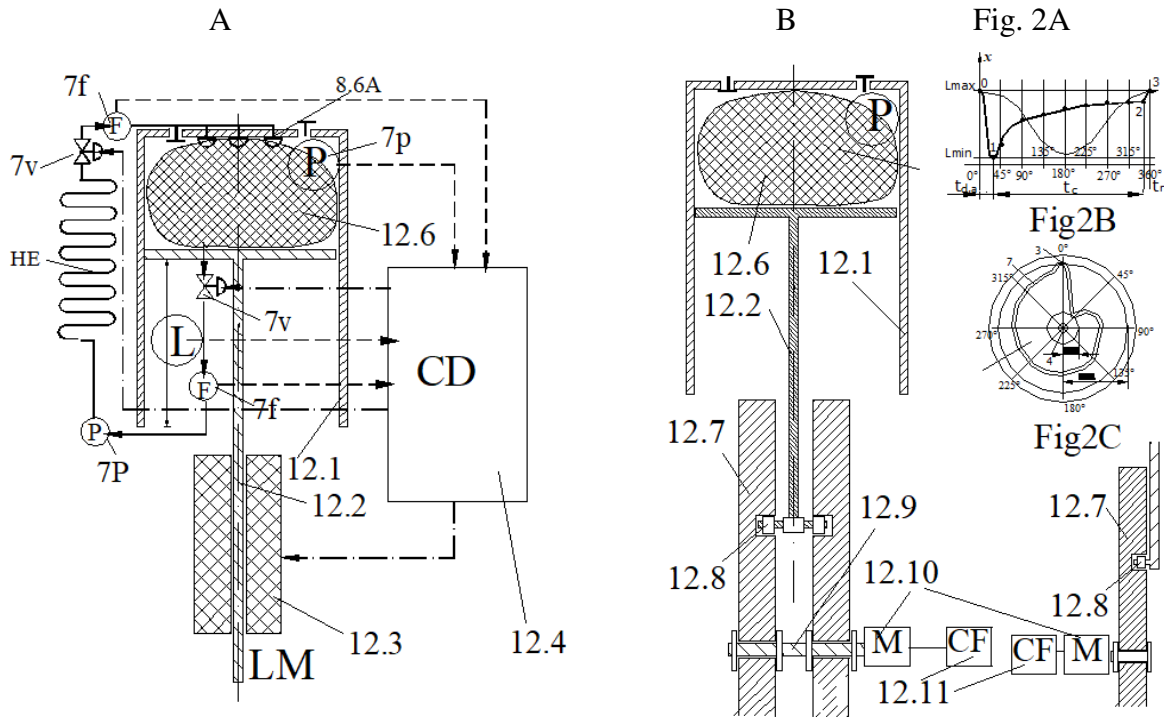


Fig.3. A : Isothermalizer with linear motor. B: Isothermalizer driven by constant speed motor and profiled channel disc

12.4, DC, which constantly compares the pressure measured by the pressure sensor 12.5, with that corresponding to the working temperature T_{iz} and the volume V_i at each moment. The calculation of the instantaneous volume is based on the position L_i 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 the most sensitive regulation systems 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 wound rotor, synchronous motors powered by a frequency converter, DC servomotors for actuating valves, stepper motors, etc. The cost of these types of drives, especially for high flow and high power density isothermalizers, is quite high and maintenance costs are not negligible. For this reason, drives using AC 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 successively in the design phase the two 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 control system: signal transducers, actuators and controller with the algorithm corresponding to the respective prototype and application. 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.3B** 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.2A** is represented, as a function of time, the shape of a curve $v_{iz}(t)$, and in **Fig.2B**, 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, shown in **Fig.2C**, 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.

7. The thermal sponge

The power consumed in an isothermal process depends, in addition to the initial working gas pressure and the compression ratio, on the isothermal temperature (initially established, depending on the desired energy efficiency) and the rate at which excess/required heat is removed/absorbed from the working gas, a rate which depends on the configuration of the device. An important objective of the invention described is to provide a series of design techniques for isothermalizer configurations, which will ensure competitive power densities.

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 solid piston, for example in the area located at the highest elevation of the device, area where, due to convection, there is a tendency to accumulate gases with a temperature above average, and the discharge of the compressed gas can be carried out by it replacing with the liquid coming from the discharge pipe. 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 S_i (**Fig.4**), which is in direct contact with the working gas, approximately the same throughout the compression phase, and their degree of deformation is constantly controlled by the position of the piston, each position of the piston corresponding to a different shape of the sponge, 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.

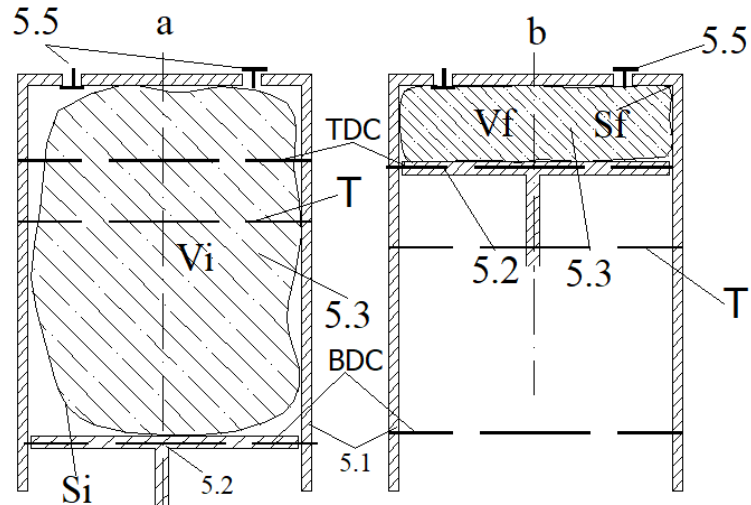
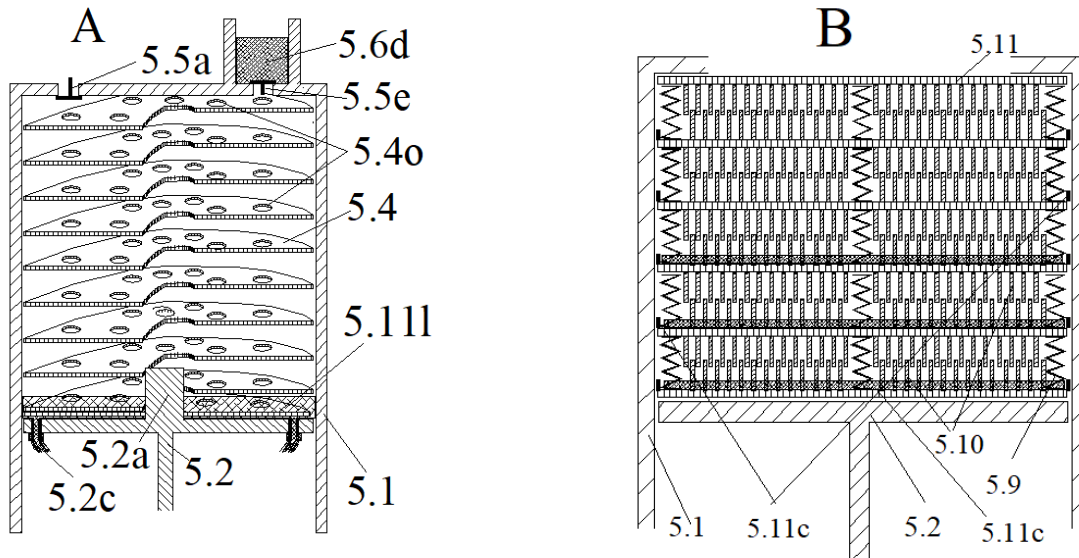


Fig. 4. Deformable thermal sponge. 5.1: cylinder, 5.2: solid piston, 5.3: deformable sponge
a: with the piston at BDC, b: with the piston at TDC



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

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. In most rotating devices, thermal sponges made of solid elements are more difficult to produce, but liquid elements of the sponge can be introduced and evacuated, both from the entry phase and during the thermodynamic transformation, in the form of a jet, in the form of drops, in the form of a spray, in the form of foam, etc., or simply by free flow. When they are

introduced/evacuated during the thermodynamic transformation, they can combine the role of liquid piston with that of cooling/heating agent.

8. Cooling systems in solid piston isothermalizers

When higher power density is desired (higher gas flow rates and implicitly higher piston speeds), in addition to the speed regulation system, isothermalizers are also equipped with a cooling/heating system, which introduces in their enclosure, during compression, through devices provided for this purpose, (for example, through sprinklers), a flow of heat transfer liquid, which is evacuated through another pipe and transported, using a pump, to a heat exchanger. If the flow rate of liquid discharged is equal to that supplied, the liquid agent is only a heat transfer liquid, but in case of inequality, the liquid agent is also a piston. Thanks to the existence of the thermal sponge, which makes the distance between any gas particle and the nearest heat-absorbing surface very small, it is no longer necessary to spray the heat transfer agent introduced, which in greatly improves the construction of the device and greatly reduces power consumption. It is sufficient for the introduced liquid to be distributed effectively over all heat-absorbing surfaces, and to drip onto these surfaces under the action of gravitational forces. Also, this type of device lends itself very well to cooling by water foam. The regeneration of the foam can be done by introducing gas (gas piston) at a pressure slightly higher than the instantaneous pressure of the gas, in strategically placed trays, which are part of the composition of the thermal sponge and which contain a mixture of liquid with surfactants.

The inlet and outlet flow rates of the liquid agent are determined by a regulator, by commands sent to the servomotors which operate the hydraulic valves mounted on the inlet and outlet pipes and by commands correlated to the commands sent by the regulator to the piston actuator.

9. Liquid piston isothermalizers

In liquid piston compressors and ionic compressors of the prior art, the compressor piston is a liquid agent supplied by a hydraulic pump, the liquid having exceptional lubricating and sealing properties, minimizing the volumes of dead gases, a two-way transport agent of thermal and mechanical energy. The disadvantage of these systems lies in the fact that if non-deformable thermal sponges, made up solely of solid elements, are introduced into the cylinder, they will at a given moment be covered by the liquid whose level increases, so that from this moment on, the heat absorption surface area of the sponge decreases (just when the amount of heat to be absorbed is greater).

One way to overcome this drawback is to make a compressor in which a thermal sponge is mounted with N fixed horizontal plates (**Fig.5**), parallel, approximately equidistant, with a surface area slightly less than that of the horizontal section through the cylinder. On the edges of the plates not welded to the side walls, they are provided with skirts 7.3a, so that pockets of gas are formed under each of the plates (practically, N elementary compressors are formed). This multiplies the absorption surface of the thermal sponge approximately N times. The plates are

arranged in such a way that after the liquid is introduced into the cylinder, all plates are covered with a layer of liquid, this becoming a liquid component of the thermal sponge, as it takes over at from the upper surface of the plate, the role of absorbing excess thermal energy. Penetration, in continuation, of the liquid agent, leads to an increase in the thickness of the liquid layers, without significantly affecting the absorption surfaces, but by further increasing the pressure in the cylinder (the liquid thermal sponge continuing its role of piston). An additional increase in the thermal absorption surface is obtained if in each elementary densifier, in its upper part, different types of metal inserts, strips, metal networks, or, if different types of vertical metal elements are mounted on the ceiling. Due to natural convective currents, as it is compressed, the gas tends to stratify, therefore in areas where the instantaneous temperature can reach higher values, the absorption surface of the thermal sponge is greater. In **Fig.5A**, the thermal sponge is completed by vertical telescopic fins, which move, simultaneously with the level of the liquid, in each elementary compressor, using a vertical rod 7.20.

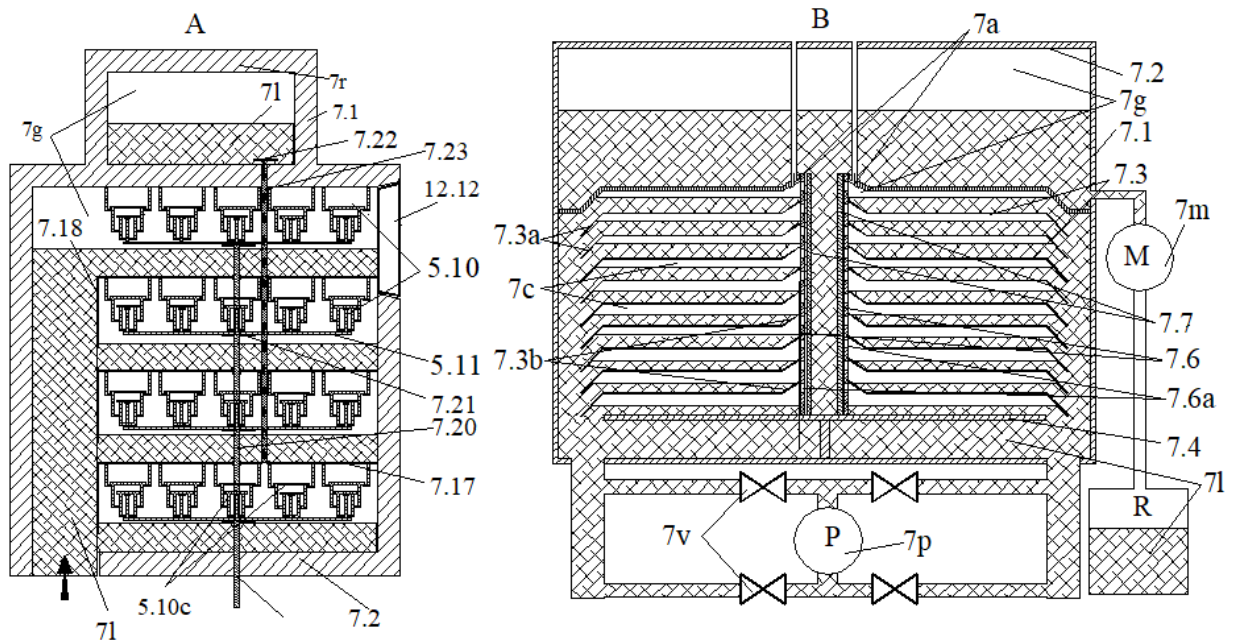


Fig.5. *Liquid piston isothermalizers.*

A: with thermal sponge with telescopic fins, B: with double-acting piston

The isothermalizer in **Fig.5B** is a double-acting liquid piston isothermalizer, obtained by assembling two isothermalizers similar to those described in Fig.5A. It is composed of two identical liquid piston isothermalizers, a parallelepipedic shell 7.1, fixed vertical walls 7.6 provided with gas outlet holes for each elementary isothermalizer 7c, movable vertical walls 7.7, stuck to them, but which can slide laterally, provided with similar evacuation holes, a solid thermal sponge, formed by the horizontal plates 7.3a provided outside the peripheral skirts 7.3a, intended for the division of the liquid piston, and towards the inside, stuck to the vertical wall 7.6, with skirts oriented towards the upper part. The liquid piston 7l is actuated by the hydraulic pump 7p, with variable flow rate thanks to the actuation of a regulating valve, controlled by a

regulation system, the imperative of which is to provide the liquid piston with the isothermal speed v_{iz} . Thanks to the 7v valve system, the liquid agent is directed alternately to the two main isothermalizers, where it is divided into elementary pistons.

Cooling of the liquid and, indirectly, of the densifier plates and the circulated gas can be done by continuous or intermittent recirculation of the liquid agent. An increase in heat transfer surfaces can be obtained if the pipe through which the gas is admitted to the densifier is powered by a foam generator. Also, at the right time, foam or compressed gas can be introduced directly into the liquid of each elementary compressor, using thin conduits.

10. Rotary isothermalizers

Isothermal transformations can also be carried out with rotary devices, from rotary compressors of the state of the art, by applying the procedures described in the invention. Also, these transformations can also be carried out with rotating devices which, in the state of the art, are most often used as liquid pumps, or as internal combustion engines, if appropriate measures are taken to ensure sealing. between enclosures of different pressures.

The isothermalizer depicted in **Fig.6A** is a variation of a vane compressor, characterized in that it uses a single vane in the rotor. It is composed of a stator (hollow cylinder 6.2), inside which rotates around its central axis, the rotor 6.1. In this configuration, the rotor cylinder is also hollow and its diameter is greater than the radius of the stator. A pocket is mounted in the rotor, generally parallelepipedic 6.4, into which the parallelepipedic blade 6.3 is inserted. Typically, the rotor is tangent to the interior surface of the stator wall.

The blade can slide over the entire height of the notch, and when its tip touches the wall of the stator, it divides it into two chambers, sealed together. This extreme position of the blade is ensured by the centrifugal force generated by its rotation, as well as by the pressure of the fluid 6.11 enclosed between the blade and the bottom of the notch (a lubricant, which is also a cooling agent of the gas and which is introduced, using a pump, through a flexible pipe).

Thanks to these constructive components, the device described is a polytropic rotary compressor which can achieve good performance in certain specific applications. Like any polytropic compressor, it can perform isothermal compression operations when its angular speed is equal to the isothermal angular speed $\omega_{iz}(t)$ throughout a rotation but, this speed is low even for large temperature differences ΔT . It is possible to achieve higher speeds and, therefore, higher compressed air flow rates by reducing the polytropic coefficient of the transformation. In the densifier of **Fig.6A**, this objective is achieved by abundantly injecting a liquid thermal agent, which has both the role of thermal sponge, when the liquid inlet flow rate is equal to the outlet flow rate, and that of liquid piston, when the inlet flow is higher. The liquid is sprayed through the nozzles 6.9b, mounted at the end of the sprinklers 6.9a, supplied by pipe 6.9. The flow rate of each sprinkler can be modified using the 6.9v adjustment devices, mounted at the inlet of the main pipe 6.9, or on each sprinkler. The rotor cylinder can also be filled with coolant, directed towards the sprinklers mounted in its wall. In addition, part of the liquid 6.11 is driven by the rotor blade and discharged, together with the compressed gas, onto the discharge pipe and after

its separation in the pipe 6.7c, it is collected in the tank 6.10, which is part a cooling circuit with the HE heat exchanger and the 6.7M pump. All actuators of the flow control devices, as well as those that determine the angular speed of the rotor, are controlled by a central DC device which receives signals from the piezoelectric pressure transducers 6.8 mounted in the working enclosure. The central device is programmed so as to ensure equality between the mechanical power given to the gas and the thermal power given by the gas to its environment.

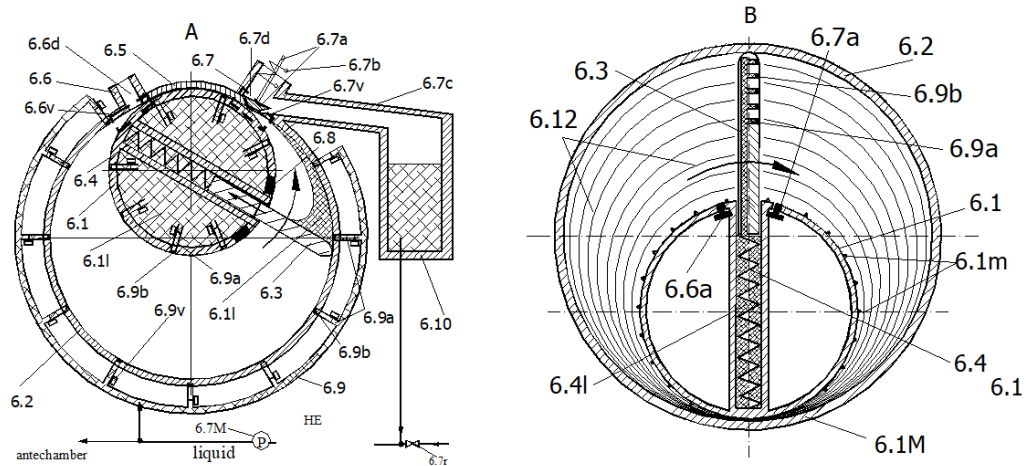


Fig.6. Isothermalizer with paddle in the rotor.

A: with isothermal angular velocity and spray cooling, B: with thermal sponge

Fig.6B shows how a solid sponge, composed of almost parallel plates, can be implemented in such an isothermalizer. The plates are cylindrical sheets 6.12, each having a notch along a generator, with an opening slightly larger than the width of the blade, having unequal diameters, values between the stator diameter and that of the rotor, mounted between these two cylinders, so that, compared to the assembly of Fig. 6A, the central axis of the rotor is moved towards the central axis of the stator, in the plane which contains them, by a distance equal to the total thickness of all these plates, leaving none of the spaces for gas leaks. By simply rotating the rotor around its axis, the cylindrical plates of the thermal sponge are driven into a rotational movement at equal angular speeds. The cooling liquid 6.4l circulates through the rotor blade 6.3, through the pipes 6.9a and the sprinklers 6.9b, from where it is injected between the sponge plates, the stator liquid being evacuated, at the using a pump, through an orifice made in the lower cover of the stator.

In applications [3-5], some procedures are presented for transforming other rotary compressors into isothermalizers: vane compressor, compressor with cams, gear compressor, scroll compressors, etc.

11. 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, makes it possible to achieve any $T(V)$ trajectory (variation of temperature as a function of piston position) desired, including

the **isothermal trajectory**, whether the response performance of the controllers is appropriate and whether the heat sources/sinks with which the devices interact meet certain conditions.

The time required to reach this trajectory depends on the constructive configuration of the device and how the device reacts to the demands of the cooling system. These two aspects are particularly complex and require numerous attempts on the test bench. 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 polytropic devices of the state of the art, large volumes of gas without exceeding a small temperature difference.

These devices seek to make the most of the exergy available in available energy sources, as well as those existing in the environment. Higher efficiencies are obtained when all thermal energy exchanges take place with optimal temperature differences (often, the smallest possible). The replacement of compressors and expanders 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.

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