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# QUASI-IZOTHERMAL COMPRESSORS AND EXPANDERS WITH LIQUID PISTON

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**Abstract.** *This paper is a continuation of efforts made by the authors for developing new systems for compression and expansion of gases and vapors, systems in that the development of processes receives a pronounced isothermal character. In the theoretical analysis of the first part of the paper, we analyze the characteristics of polytropic transformations that occur in the installations for compression and expansion of the state of the art. Are highlighted, clearly, the conditions to be satisfied by these transformations to be as close as possible than an isothermal transformation, and the power limitations resulting from such conditions. Next, we review a number of practical achievements and theoretical studies that overcome these limitations by replacing the solid mobile organ via a liquid piston. We also propose several original types of compressors and expanders with liquid piston and a series of installations that take advantage of this type of driving.*

**Keywords:** *gas and vapor compression and expansion, quasi-isothermal processes, isothermal processes, liquid piston.*

## 1. THEORETICAL CONSIDERATIONS

### 1.1. Isothermal Processes

The kinetic-molecular theory demonstrated even as of its beginnings that the interaction between gas and mobile organ results in changing momentum of the gas molecules which collide with the mobile organ in motion. During the compression process, the mechanical kinetic energy of the mobile body is converted to mechanical potential energy of the gas / vapor system on the one hand (by increasing its pressure), and to thermal energy on the other hand (through temperature increase). During the expansion process the mechanical potential energy and part of the internal energy of the system, plus in almost all cases, the thermal energy absorbed from the external environment, turns into mechanical kinetic energy to trigger the mobile organ. In an ideal adiabatic compression, reversible, all the kinetic energy of the mobile organ is taken by the gas, resulting in a maximum increase of its temperature, while an ideal isothermal compression the thermal energy generated by the mobile organ is eliminated through the walls and through the cooling of the lubricant, while the gas temperature remains the same. For piston compressors, Stoicescu and Petrescu [1] have shown that, compared with the compression at a speed very close to zero, for high speeds of the piston, there is an additional increase in pressure on the piston.

It is widely recognized that in order to achieve economical compressions or to achieve performing thermal engines (especially from energy sources with low thermal potential such as renewable resources), isothermal compression is the preferred choice. For a given volume of gas, a higher temperature means higher pressure, hence a further increase of the mechanical work needed to continue the compression. As a result, for the compression of a gas whose temperature is slightly higher than that of the refrigerant, the mechanical work consumed is lower as the conversion is closer to an isothermal one. Notwithstanding that this compression process is always accompanied by a process of heat transfer: for each cubic meter of air isothermally compressed at ambient temperature, in a ratio of 1:100 (in order to introduce it into a 10 L tank) will need to rise with 10°C the temperature of 11 L of cooling water, and to recover then at the same temperature, through isothermal expansion, the mechanical energy used, the heat produced by burning 10g of diesel is required.

### 1.2. Isothermal and quasi-isothermal reversible transformations

According to the first principle of thermodynamics:

$$dU = \delta Q - \delta L = \delta Q - p \cdot dV \quad (1)$$

In an isothermal process  $dU = 0$ , hence:

$$\delta Q = \delta L. \quad (2)$$

On the other hand,

$$\delta Q_{rev}(t) = \sum_i [U_{gi}(t) \cdot A_i(t)] \cdot (T_S - T) \cdot dt = \frac{T_S - T}{R(t)} dt \quad (3)$$

where:  $T$  is the gas temperature, which in most cases is a variable dependent on a number of parameters, which in turn are dependent on the variable  $t$  (time reported to initial moment  $t_0$ );  $T_S$  is the temperature of the source (including in the source its border thereof, the walls of the device), and  $U_{gi}$ , respectively  $A_i$  are the global coefficient of thermal and respectively the surface of portion  $i$  (different from the other portions from the thermal transfer characteristics point of view) of the total surface separating the two systems. The portions which make up the total area differ from each other in thickness, by the material they are made of, by the manner in which they are cooled or heated, by contact duration with the working fluid, etc. We denoted by  $R(t)$  the overall thermal resistance of the system. The requirement for a transformation to be isothermal is that  $T(t) = T_0 = ct$ . If there is no temperature difference between the source and the system,  $T_S = T(t_0)$ , then  $\delta Q = 0$ , and  $\delta W = 0$  and therefore there cannot be a heat exchange nor a isothermal transformation. The heat exchange can only take place in the moments when this difference is not zero. In the thermal installations in current development stage there are no isothermal processes without phase change, the almost unanimous opinion being that this type of process is purely theoretical. Most common solution is the one in which the boundary of the source plays an active role meaning that after the establishment of a quasi-stationary regime, the wall temperature of the device is stabilized to an almost constant value: in case of compression greater than the one of the source, and in case of expansion lower than this. During a compression the wall of the device will deliver to the system a quantity of thermal energy in the first part of the cycle, as in the second part it will absorb from the system a larger amount. The difference between the two quantities is discharged into the environment during the entire cycle. In such a system, the gas passes through a polytropic transformation in which the polytropic exponent is variable: both for compression as well as for expansion at the beginning of the transformation its value is higher than the adiabatic exponent, it becomes equal to it when the gas temperature equals the wall temperature and decrease below its value towards the end of the transformation. In these circumstances, researcher's efforts were not

directed towards reaching an isothermal process, but to decrease as sharp as possible the polytropic exponent, and thus the differences between the gas temperature at the entrance and exit of the device.

Török and all [2] approached a different method, searching first of all, disregarding the configuration of the device used, the conditions that need to be met so that the respective thermodynamic transformation is a perfect isothermal one. They showed that in any positive displacement compressor a perfect isothermal compression can be achieved, if a positive temperature difference between the gas and the external environment is permanently maintained, if the piston starts its motion with a certain speed and if later the speed varies so that a certain equation of motion is respected (derived from the need to permanently respect the equality of mechanical power of the piston and the flow of thermal energy exhausted from the compressor. Indeed, from equations (2) and (3) follows

$$p \cdot \frac{dV(t)}{dt} \cdot dt = \frac{T_S - T_0}{R(t)} \cdot dt = (T_S - T_0) \sum_i [U_{gi}(t) \cdot A_i(t)] dt \quad (4) [2]$$

By splitting to  $dt$ , the first term is the instantaneous mechanical power, and the next will be the instantaneous thermal power exchanged with the external environment. For the transformation to be isothermal, the equality of the two powers must be maintained throughout the transformation. For the two powers the speed dependence of  $w$  (or of angular velocity  $\omega$ ) may be emphasized:

$$P(w) = (T_S - T_0) \sum_i [U_{gi}(w) \cdot A_i(w)] \quad (5)$$

thus resulting an equation whose unknown is the function  $w(t)$ , and by solving it one can find how to vary this speed to meet the condition imposed. For  $t = 0$ , we find the initial speed  $w_0$ : startup speed of the mobile organ. But  $P(w) = F(w) \cdot w$  so that the equation (5) can be decomposed into  $i$  simple equations:

$$P(w_i) = (T_S - T_0) \cdot U_{gi}(w_i) \cdot A_i(w_i), \quad (6)$$

one for each portion  $i$  of the total area which has a different thermal behavior from the other, from which we get  $i$  solutions  $w_i$ , each representing the speed of the piston for which isothermal transformation is obtained when the other portions are thermally insulated from the environment. The general solution of equation (6) is the sum of the  $i$  speeds. The method is particularly useful in assessing the contribution of each method used to improve the overall coefficient of thermal transfer. For

example, for machines with pistons, the gas volume at a given time will be:  $V = V_0 \pm x \cdot A = (x_0 \pm x) \cdot A$  such that:

$$\begin{aligned} P(x) &= p \cdot \frac{dV}{dt} = \frac{p_0 \cdot V_0}{V} \cdot \frac{dV}{dt} = \\ &= p_0 \cdot V_0 \cdot \frac{d}{dt} [\ln(x_0 \pm x)] = \frac{T_S - T_0}{R(t)} \end{aligned} \quad (7)$$

For the piston and the cylinder head we can consider the thermal resistance being constant in time, hence  $R(t) = R_0$ . The equation (7) becomes:

$$p_0 \cdot V_0 \cdot \frac{d}{dt} [\ln(x_0 \pm x)] = \frac{T_S - T_0}{R_0},$$

hence:

$$\begin{aligned} \ln(x_0 \pm x) &= \frac{T_S - T_0}{R_0 \cdot p_0 \cdot V_0} \int dt = \\ &= k_0 \cdot t + \ln x_0 \end{aligned}$$

where  $k_0 = \frac{T_S - T_0}{R_0 \cdot p_0 \cdot V_0}$  can be positive or negative, as the process is a compression or an expansion:

$$x = x_0(1 \pm e^{k_0 t}) \quad (8)$$

and

$$w_1(t) = \frac{dx}{dt} = \pm x_0 \cdot k_0 \cdot e^{k_0 t} = w_0 \cdot e^{k_0 t}$$

If we choose as the independent variable

$$\varepsilon(t) = V/V_0,$$

the equation becomes:

$$p_0 \cdot V_0 \cdot \frac{d}{dt} \ln \varepsilon = \frac{T_S - T_0}{R_0}$$

with the solution:

$$\varepsilon(t) = e^{k_0 t} \quad (9)$$

and the initial condition  $\varepsilon(0) = 1$ .

It notes the link between the two functions:

$$w_1(t) = w_0 \cdot e(t)$$

For the side surface of the cylinder, the area through which the system is exchanging thermal energy with the environment is changing as the piston moves:  $A_2(t) = \pi D(x_0 \pm x)$ , so:

$$\begin{aligned} P(x) &= \frac{p_0 \cdot V_0}{(x_0 \pm x) \cdot A} \cdot \frac{d[A \cdot (x_0 \pm x)]}{dt} = \\ &= \pi D(x_0 \pm x) \cdot U_{g2} \cdot (T_S - T_0) \end{aligned} \quad (10)$$

$$\begin{aligned} \frac{d(x_0 \pm x)}{(x_0 \pm x)^2} \cdot \frac{d}{dt} \left[ \frac{-1}{(x_0 \pm x)} \right] &= \\ &= \frac{T_S - T_0}{p_0 \cdot V_0} \cdot \pi \cdot D \cdot U_{g2} = k \end{aligned}$$

$$\frac{1}{(-x_0 \mp x)} = K \cdot t + K_0 \quad x = x_0 \pm \frac{1}{K \cdot t + K_0}$$

and

$$w_2 = \mp \frac{1}{K(K \cdot t + K_0)^2}$$

The optimal speed of the piston will be

$$w = 2w_1 + w_2$$

### 1.3. Irreversible isothermal transformations

To study the cumulative effect of several types of commonly occurring irreversible compression and expansion processes we used the equation [3]:

$$\begin{aligned} \delta W_{ir} &= p_{m,i} \left( 1 \pm \frac{a w}{\sqrt{3RT_{m,i}}} \pm \frac{b \Delta p_{lam}}{2 p_{m,i}} \pm \frac{\Delta p_f}{p_{m,i}} \right) dV = \\ &= p_{m,i} \cdot dV p_{m,i} \pm \frac{a w}{\sqrt{3RT_{m,i}}} \cdot dV \pm \\ &\quad \pm b \Delta p_{lam} \cdot dV \pm \Delta p_f \cdot dV = \\ &= \delta Q_{rev} \pm \delta Q_w \pm \delta Q_{lam} \pm \delta Q_{fr} \end{aligned} \quad (11)$$

where each of the terms in parenthesis explains quantitatively the contribution of one of the irreversibility of the system. It is also underlines that each of these terms has the size and the meaning of quantity of heat:  $Q_{rev}$  is the heat exchanged with the environment in a reversible process,  $Q_w$  is the heat caused by the influence of the finite speed of the piston  $w$  (during the compression, the gas in the vicinity of the piston has both the temperature and the pressure higher than the rest of the system, and smaller during the expansion),  $Q_{lam}$  is the heat produced by the gaso-dynamic friction within the fluid and at its passing through the valves or other types the constrictions (only part of this heat effectively to the increase of the temperature, the remainder being used to recover the potential energy lost due to a temporary increase in speed, but results in a pressure drop  $\Delta p_{lam}$ ) and  $Q_{fr}$  is the part of heat produced by the friction between the piston and the cylinder wall, which remains in the system [4, 5, 6, 7, 8].

In the case of an isothermal transformation after highlighting the correlations between the piston

speed and the other factors that determine the size of irreversibility [3]:

$$\frac{a \cdot w}{\sqrt{3RT}} = k \frac{w}{w_s} \quad \Delta p_f = A_f + w B_f$$

$$\Delta p_{lam} = \beta \cdot p_{m,i} \cdot k \cdot \left( \frac{A_p}{A_v} \right)^2 \cdot \frac{w^2}{w_s^2}$$

equation (11) becomes:

$$\delta Q_{ir} = \left\{ 1 \pm k \left( \frac{w}{w_s} \right) \pm \frac{kb}{2} \beta \left( \frac{A_p}{A_v} \right)^2 \left( \frac{w}{w_s} \right)^2 \right\} p_{m,i} \cdot dV \pm$$

$$\pm f(A_f + B_f w) \cdot dV =$$

$$= \delta Q_{rev} \pm \delta Q_w \pm \delta Q_{lam} \pm \delta Q_{fr} \quad (12)$$

In Figure 1 are the curves of isothermal expansion and compression in the presence of irreversibility caused by the finite speed  $w$  of the piston. The area between the curve of evolution of the instantaneous average pressure  $p_{m,i}$  and the pressure on the piston  $p_p$  is a measure of the additional mechanical work consumed during the

compression, respectively the lost work due to the irreversibility during the expansion. In the first case, the thermal equivalent of work should be disposed, but in the case of expansion the thermal equivalent of work, although it is extracted from the hot source, does not result in the creation of useful work.

Using a decomposition like the equation (6), we can solve the equation without taking into account the irreversibility, and after finding the speed equation  $w(t)$ , we solve successively the three equations corresponding to the conversion of the useful mechanical work  $p \cdot dV$  into heat in the case of the three irreversibility. The three speeds thus obtained shall be deducted from/added up to the total speed and the difference will show, after a few iterations, how the piston speed must evolve to achieve isothermal transformation. So, in order to achieve an isothermal process, in case of the compression, for each of irreversibility, the mechanical work consumed is higher compared to a reversible transformation, which corresponds to an additional heat that must be exhausted, and for expansion less heat is absorbed from the hot source and less mechanical work is produced.

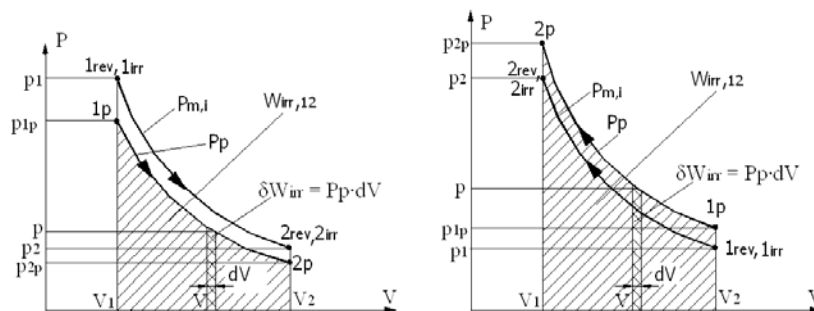


Fig. 1. Expansion and compression isotherms in the presence of irreversibility caused by the finite speed  $w$  of the piston [6].

## 2. ISOTHERMAL COMPRESSORS AND EXPANDERS

### 2.1. Compressors and expanders with solid piston

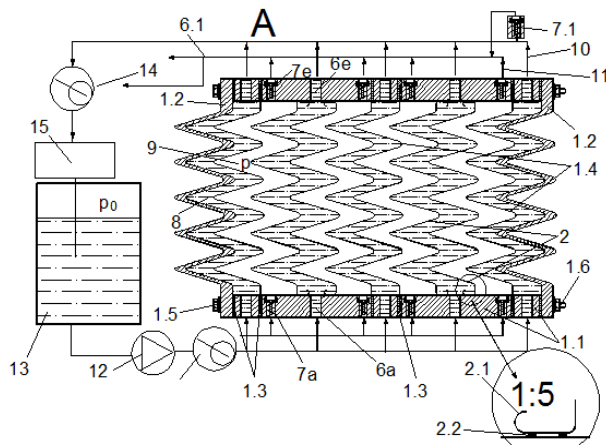
The theoretical analysis of the progress in the isothermal processes led us to the conclusion that, regardless of the configuration of the positive displacement device, the thermodynamic process can be one isothermal, or close to it (quasi-isothermal), if the mechanism that operates its mobile organ may cause the volume variation as close to the manner described by equations (4) and (5). The same analysis allows us to determine the speed of the process to achieve an isothermal transformation. For example, for compressing air in a cylindrical

reciprocating compressor made of cast iron with a diameter of 20 cm and a length of 30 cm, without using any cooling system, but with a temperature difference of 10°C between the working agent and the outside air, in order to achieve a compression ratio of approx. 7.39 ( $e^2$ ), the starting speed of the piston has to be approx. 0.0036 m/s, to make the compression ratio after 166 seconds, at which moment, the piston will have a velocity of 0.00049 m/s. For the actuator with profiled cams, this signifies a rate of rotation about 20 rot/hour, which means an air debit about 0.2 m<sup>3</sup>.

Equation (5) is the one that clearly suggests the following paths for getting some higher speeds and debits: the growth of the global transfer coefficient and the growth of the area through which occurs

the heat transfer to/from outdoors. The authors have already proposed [2], several methods and several device models in this respect. For example, for a device with the same volume as the one considered above, but with a diameter of 48.9 cm and a length of 5 cm, to which both the piston and the cylinder head are profiled (with needle wedge profiles with an opening angle  $6^\circ$ ), the calculation showed a 16-fold increase in the average speed for that can be obtained perfectly isothermal transformations, and if this device is completely immersed in a cooling bath, there is obtained a doubling of the speed (angular speed of about 10 rot/min). The need to eliminate the waste heat by circulating coolant was the one of causes who suggested the use of a liquid pump from this circuit as the main device for the production and modification of power needed to compress the gas. The devices that changes the fluid flow and pressure are easier to achieve than the devices with profiled cams, have a low inertia, responds quickly to commands and causes less vibration.

In Figure 2 there is shown a compressor of which inner space is divided by fixed profiled walls 1.4. The walls are separated from each other by the metallic frame 1.1 by whose assembling (by studs and nuts 1.5 and 1.6) obtain the side walls of the compressor, while the sealing between the frame and the wall is made of elastic sealants, which allows a rapid disassembly.



**Fig. 2.** Compressor driven by the pressure created by the liquid pump:

1.2 – side walls; 1.3 – seals; 1.4 – fixed walls; 1.5, 1.6 – joiners; 2 – mobile walls (pistons); 6a, 6e – liquid valves; 7.1, 7, 7e – gas valves; 8 – liquid; 9 – gas; 10, 11 – pipes; 12 – liquid pump; 13 – tank  $p = \text{ct.}$ ; 14 – hydraulic motor (turbine); 15 – heat exchanger [9].

Between each pair of the walls are placed two movable pistons each composed of a profiled wall, and a sliding system. Each frame are the valves 6a for the intake of working fluid between the two

pistons and the valves 6e, respectively for his exhaust. When the intake flow is higher than the outlet flow, the pistons are pushed towards the fixed wall, compressing a gas from this compartment and exhaust it after the exhaust valve 7e. is openned. When the exhaust flow rate is higher than the inlet flow, the pressure of the fluid drops abruptly (the liquids are essentially incompressible), which leads to the opening of the intake valve and the gas filling of the respective compartments. The compression of the gas from the compression chamber is performed only by the introduction of liquid by the pump 12, simultaneously with the limitation of the flow of exhausted liquid, by the hydraulic turbine 14.1 (or a hydraulic motor, or a control valve, etc.).

## 2.2. Compressors and expanders with liquid piston

The enhancement of surfaces bounding the gas system, through which occurs the heat exchange between it and the outside is limited by technical and economic difficulties. On the other hand, a careful analysis of (5) points out that the overall heat transfer coefficient growth is limited also. In fact, its value for any of the portions  $i$  of the boundary component is given by:

$$U_g = \frac{1}{R_g} = \frac{1}{\frac{1}{\alpha_i} + \frac{\delta}{\lambda} + \frac{1}{\alpha_e}} \quad (13)$$

where  $\alpha_i$  and  $\alpha_e$  are the individual transfer coefficients between the wall and the inside fluid respectively for the outside fluid,  $\delta$  is the wall thickness, and  $\lambda$  is the thermal conductivity of the wall. Most of the times, the wall thickness of the usually apparatus is small (of the order of millimeters, sometimes centimeters), and the thermal conductivity of the material used is high (on the order of tens, even hundreds W/mK), so the second term of the denominator can be neglected. When the outer wall is cooled by a liquid (especially when the device is fully submerged in a liquid pool), the third term in the denominator is also negligible, so that  $U_g \approx \alpha_i$ . For all technical gases, this term is rather small (10-40 W/m<sup>2</sup>·K) severely limiting the possibility of increasing  $U_g$ . So, the researches were again oriented towards the increasing the contact area between the inside gas and the cooling agent. Technical literature devoted to the study of heat transfer processes inside the compressor is large [12-31]. There are also numerous patents and patent applications that suggest practical solutions to solve

this problem [33-41]. Three main methods have been proposed to increase this area:

- the abundant spraying of the coolant during the compression process, or of the thermal agent during the expansion process. The spraying can be performed also before the suction (with the additional consumption of the work). The volume of exhausted fluid will contain a significant fraction of fluid under pressure, and its potential energy will be recovered in a hydraulic motor circuit [41];
- the compression of the gas introduced into the liquid in the form of bubbles, either by hydrostatic pressure of a liquid column, either by the pressure created through centrifugation, or by the pressure created by a liquid pump. In the first case, the circuit can be opened (and use the gravitational fall of a liquid contained in a pool) or closed (use a fluid pump to compensate the pressure losses). In the second case, the amount of energy required for centrifugation can be recovered in a hydraulic turbine, and in the latter case using a compression in pressure steps can obtain the high gas flow [40];
- the increase of the overall coefficient of heat transfer between the gas and the environment, through the inner absorbent mass in contact with the gas during the compression process and in contact with the liquid at the end of the compression and during the exhaust. The use of the liquid piston, among other undeniable advantages [30, 31], has the benefit that the liquid used for compression (a fixed amount, introduced through a solid piston or a variable amount introduced through a pump) penetrates the interstices of the absorbent network and can quickly absorb from it (its heat transfer coefficient is more greater than that of the gas) the heat collected during compression process. In the expanders, the superheated liquid delivers to the network the heat required by the gas for an isothermal expansion.

### 3. THERMODYNAMIC CYCLES WITH ISOTHERMAL COMPRESSION AND EXPANSION

One of the advantages of piston compressors and expanders with liquid is the possibility to obtain isothermal thermodynamic transformations which creates the opportunity to build thermal devices where the efficiency (or COP, for the reversed cycle) is close to the optimal (corresponding to a device that operates as a Carnot cycle). The engine in Figure 3.1 is a Carnot type engine made of a compressor and an expander, thermally insulated from the environment (insulation 1.7), facing each other, communicating through pipe 11, and having a well defined ratio between their volumes (compressor has a higher volume than the expander), predetermined based on the temperatures needed for isothermal compression

and expansion (different from the temperatures of hot and cold sources). The other prerequisite needed to achieve isothermal transformation is that the speed of the piston complies with the equations derived from parity at each moment of the mechanical power of each of the pistons with the thermal flux of the respective device exchanged with the environment. Figure 3,A shows the positions of the pistons at the end of each of the four phases of the engine cycle (Fig. 3,B):

- In the initial position (position 1 in the  $T$ - $s$  diagram), the compressor cylinder is empty, the position of its piston being at the end of the stroke and a gas volume  $V_1$  at pressure  $P_1$  and temperature  $T_1$  is in the expander. In the first phase of the cycle (curve 1-2), this gas expands isothermally (absorbing heat from the hot source) up to a volume  $V_2$  and a pressure  $P_2$ , pushing the exterior piston using the liquid, and keeping constant temperature. For the transformation to be isothermal, the piston has to start with a speed  $w_0$ , given by the structural characteristics of the expander, by the mass and the properties of the working gas and by the temperature difference between the gas and the hot source. During this phase, the piston compressor will remain motionless, while the one of the expander will move with the speed required to constantly maintain the product  $A \cdot w \cdot p$  ( $A$  is the piston surface at a given time) equal to the heat flux from the hot source to the gas. This equation leads to an exponential increase in piston speed (Fig. 3,C). In this phase of the cycle the engine receives heat from the hot source which gets converted into mechanical work to expand the gas, which in turn is sent out as useful mechanical work.

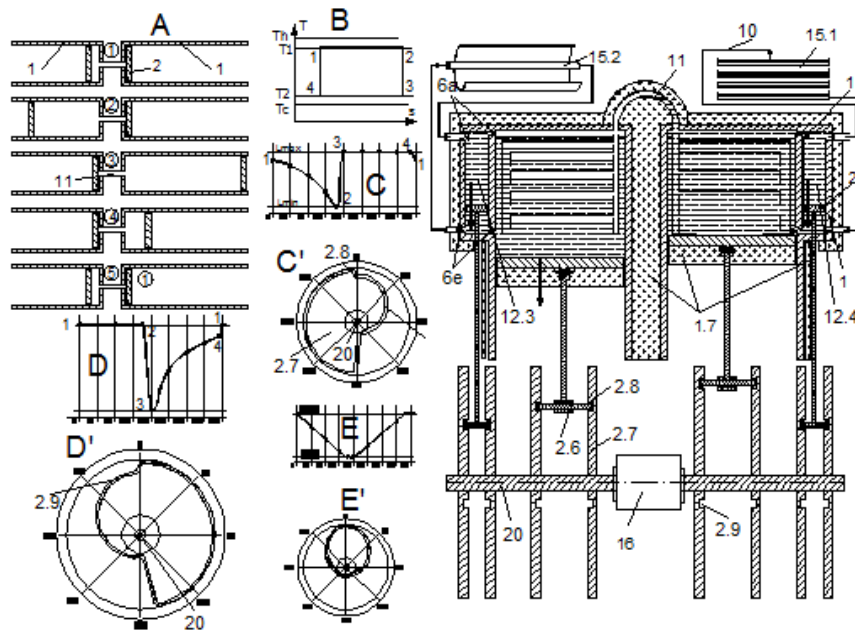
- In the second phase of the cycle, both pistons go in a very short time through the entire length of the cylinder, backwards. As a result, the gas is transferred into the compressor, and in the same time it expands to a volume  $V_3$  and a pressure  $p_3$ . Due to the high speed of the pistons, the quantity of heat exchanged with the environment is very small, so that the process can be considered adiabatic, the temperature dropping to the value  $T_2$  given by the equation of such a transformation (Fig. 3,B and C, curve 2-3). The adiabatic expansion in the compressor is done by transforming part of its internal energy into mechanical work, which is transmitted to the piston.

- In the third phase of the cycle, the expander piston remains at the end of the cylinder, while the piston of the compressor compresses the gas up to a pressure  $p_4$ . Having a higher piston speed in the beginning, it decreases so that the gas temperature remains constant (Fig. 3.1 B and C, curve 3-4).



– In the last phase of the cycle, the compressor piston crosses in a very short time the remaining length of the cylinder while the expander piston will

move into the position corresponding to volume  $V_1$ . The gas will be compressed adiabatically to a pressure  $p_1$ , its temperature increasing to the value  $T_1$ .



**Fig. 3.1.** Carnot engine developed with liquid pistons:

1 – cylinder; 2 – piston; 1.7 – thermal insulation; 11 – pipe; 2.6-2.9 – actuator;  
20 – ax; 15.1 – cooler; 15.2 – solar collector; 16 – electric generator.

In the last two stages of the cycle, the engine consumes part of the mechanical energy accumulated in the rotation of the flywheel in order to achieve the two compression stages.

Using the same configuration, the system can operate as a refrigerator, or as a heat pump by reversing the cycle. In the reverse cycle, the expander becomes the compressor, and through the isothermal compression of the gas from the volume  $V_2$  to the volume  $V_1$  external mechanical work is used and heat is yielded to the environment; the compressor becomes expander and through the isothermal gas expansion from the volume  $V_4$  up to the volume  $V_3$  heat extracted from the external environment is used and mechanical work is yielded to the machine shaft. Obtaining the variations in speed required to achieve such a cycle can be achieved through multiple methods, in Figure 3 the solution chosen is to engage the pistons using the rods 2.6 with bearings 2.8 at both ends moving through the profiled channels 2.9, dug into discs 2.7, which rotate around the pin 20. The profiles shape is obtained by transposing into polar coordinates the desired equation of motion (curves  $B'$  and  $C'$ ).

Figure 3 presents one way of building the engine, using the compressor and the expander shown in [40]. The liquid in the expander is heated continuously by a circuit which includes the solar tubes

15.2 installed in the linear focus of trough parabolic mirrors, and the compressor is 12.3 which extracts cooled fluid and introduces heated fluid during the intervals when the expander piston is motionless. The piston of this compressor is synchronized with the other pistons using profiled rotary discs mounted on the same shaft. The law of motion of the piston and the form of the profiled channel are shown in Figures 3.1,  $E$  and  $E'$ . The cooling circuit of the liquid in the compressor is made of a compressor 12.4 similar to that described above, and a cooler 15.1. The cooler in the figure is made up of a large number of trays made of perforated plates, or wire mesh, through which cold air suctioned from the lower part flows. The system described is easily adaptable for high power solar plants, using as working liquid molten salts at high temperatures.

#### 4. CONCLUSIONS

Although there is recent discoveries, the compressors and the expanders with liquid piston became the most powerful of devices in this category. They are able to achieve compression of gases and vapors with a low consumption of mechanical energy and the expansion is achieved by the most efficient use of the heat extracted from the heat source. For this reason they are the most



suitable equipment the valorization of renewable energy and for the energy storage facilities.

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