The concept of technical combination of water supply systems and low-temperature air flow generation

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Abstract. The concept concerns the field of design and development of technical systems with an increased level of operational adaptability, in particular, the creation of air compression systems to lower its temperature before being supplied to a technological facility using water pressure in the water supply system of this facility. The innovation lies in filling the water storage tank with a water mass that simulates the operation of a liquid piston, with an increase in which air is compressed in the tank while cooling it based on the "adiabatic throttling" effect before being forced into the injection line, while the full working cycle is realized during two alternating cycles of filling the water storage tank due to hydropower with a water mass filling the capacity and electricity consumed by the pump during the period of the operating cycle. The low-temperature air flow is generated by means of a pneumatic accumulator (receiver), which is pre-filled with compressed air during the displacement of air from the working cavity of the water storage tank by a liquid piston, and then the compressed air is forced back into the upper piston space of the water storage tank in the "adiabatic expansion" mode.

1 Introduction

The profitability of open and closed–ground vegetable production is determined, on the one hand, by the technology of their cultivation, and, on the other, by the level of energy consumption of vegetable storages. In practice, it has been established that a significant energy-saving effect for vegetable storages can be achieved, for example, by introducing a rational systemic thermophysical approach, since it makes it possible to take into account the influence of a large number of factors on the thermal and air storage regime, or by creating alternative heat and cold supply systems that save energy and water resources while creating the necessary microclimatic conditions. Thus, it is possible to extend the shelf life of products by 2-3 times, as well as reduce electricity consumption by 40-50%. In this regard, a significant part of innovations in the field of vegetable storage is aimed at creating and maintaining a balanced microclimate inside vegetable storages, since the main factor affecting the safety of vegetables, as well as reducing energy consumption, is the structural

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and operational optimization of refrigeration systems as the main consumers of electricity [1, 2].

One of the State priorities in scientific areas is the technological reproduction of natural processes in the form of technical systems integrated into natural circulation by building harmonious interaction between humans and the environment, which underlies the improvement of the effectiveness of state scientific and technical policy, as well as ensuring the technological independence and competitiveness of the state [3].

In this conceptual aspect, we can propose a model of a technological system (Figure 1) for long-term storage of vegetable products due to the energy-efficient use of ambient air. We are talking about the development of dome-shaped buried storage projects. The technical result of the proposed innovation is to reduce product losses and reduce energy consumption, and simplifying the design naturally entails a reduction in operating costs.

The proposed design of the technological system is focused on the process of cold supply of dome-shaped buried vegetable storages, in particular, on preparation, transportation, and supply of compressed air to the receiver pipe for further cooling and transfer to the supporting soil mass using a spreading pipeline line – soil cooling. The presented technological system can be used in various regions, regardless of climatic conditions, for small businesses with a loading volume of vegetable storages from 20 to 1000 tons. The main task of the project development was to optimize the temperature regime of vegetable storages at different times of the year, as well as improve the technological performance of the low-potential heat generation system and modernize specialized equipment for transporting and accumulating temperature backgrounds [4, 5].

The cooling supply of the "stratodesic dome", that is, the storage of the buried type, simulates the process of a natural air compression refrigerating machine using a single-stage piston type power plant as a compressor. A technical feature in the project is the use of a liquid compression method – the injection principle of compressing air in containers with water before distributing it to technological facilities. When formalizing the calculation procedures, factor prerequisites were taken into account: refrigerant – natural (atmospheric) air; cooler – heat exchanger – frostable bearing soils that ensure thermal stabilization of the room for stored vegetable products; expander – a two-circuit technical structure - "pipe in a pipe" based on electromagnetic communication with the receiver; the process of "cold charging" of the bearing soil is implemented for account of the Cayette effect. It should be noted that the cooled room was considered during mathematical formalization as a dome-shaped object of a recessed type, characterized by minimal heat losses compared to vegetable storages of a standard type with flat enclosing surfaces [5, 6].

It has been proven that refrigeration equipment can operate efficiently without damaging the environment if it uses natural refrigerants, such as air. Along with electricity, gas and water, compressed air is considered the fourth energy carrier in the field of industrial production, supplying energy to serial technological processes. The use of compressed air as a refrigerant is reliably due to its inexhaustible energy functionality, the implementation of which is possible through cost-effective technical solutions, and, at the same time, the power transmitted by it with a small weight is inadequately large [7]. However, the use of air is technologically limited due to its low specific heat, which is approximately 1 kJ/kgF. The problem is the insufficient productivity and low cooling characteristics of refrigeration units – the power limit is up to 100 kW. Thus, in order to obtain the required cooling effect, the compressor needs to compress such a large volume of air that its energy potential can be technically realized and transformed in the production sector for the needs of each specific consumer. This project presents a methodological framework that makes it possible to formalize a number of patterns underlying or arising from the structural and functional relationships of a technological system in small business [7, 8].

2 Materials and methods

The functional capability of the compressor is assessed by performance or flow rate. This is a qualitative indicator, and it refers to the amount of gas pumped by the compressor per unit of time, expressed by the volume that this gas will occupy at suction pressure and suction temperature. At the same time, the smaller the volume of gas that remains in the working chamber and cannot be displaced from it with the minimum volume of this chamber and high pressure in it, the greater the portioned filling can be expected with the maximum volume of the working chamber and low pressure in it [9]. The considered ratio of pressures and volumes of gas is confirmed by well-known thermodynamic dependences, including the Clapeyron-Mendeleev equation of state for an ideal gas and the Van der Waals equation for a real gas, which implies that compression in the usual sense of the word can be achieved either by increasing pressure or decreasing temperature. In principle, this statement was a prerequisite for research into the production of compressed gas – atmospheric air – by combining the processes of increasing pressure and lowering temperature in the working chamber of compressor units.

According to the equation of state of an ideal gas, the specific volume of compressed air is V, m^3:

$$V = RT / P \tag{1}$$

where P is the pressure of compressed air, Pa; V is the volume of compressed air, m^3; R is the universal gas constant, J / mol K; T is the thermodynamic temperature, K.

Or real gas, the specific volume of compressed air V, m^3 is:

$$V = zRT / P \tag{2}$$

where z is the compressibility factor.

Currently, multistage reciprocating compressors are usually used to compress gas to medium and high pressures. However, the technological perspective is expressed in projects of systems with pre-compression in a single-stage reciprocating compressor and final compression of the gas in the working chamber above the liquid piston. Compressors with liquid pistons are widely known in the range of modern technology. They differ from other types of compressors in their high efficiency, simplicity of design, cost-effectiveness of maintenance and repair, and high reliability. A significant advantage of a compressor with a liquid piston is the intensive cooling of the compressed gas, and reducing the piston speed as the pressure increases brings the compression process closer to thermodynamically efficient. The absence of leaks and overflows in its working chamber is also a positive and important factor. All this makes it possible to increase the indicator and isothermal efficiency, as well as the compressor supply coefficient [10].

Thus, the creation of a compressor with high efficiency when operating in a wide range of pressures of the air injected into the receiver should be considered appropriate. At the same time, there are tasks to study the comparative effectiveness of various air supply methods, to develop a mathematical model and a methodology for calculating the process of filling the receiver and the upper piston space of the operated tank with air. The installation of an experimental installation for studying the processes of filling and emptying the container during compression showed the need for experimental studies of the process of filling the receiver and the working cavity with air. But the basic role remains for the mathematical formalization of the cyclic technical complex – "working cavity – pipeline – receiver" (WCPR).

Technologically, the working process with air, which occurs in one cycle of a compressor with a liquid piston, lasts two cycles and is divided into 4 stages: air supply to the receiver (air compression), the process of air expansion (adiabatic throttling), air intake (capture), air ejection (injection) to technological facilities. At the same time, the workflow with water is two-stage: filling the tank with water and gravitational removal of water from the inner cavity of the operated tank.

A mathematical model of the workflow implemented in a compression chamber, a combined single-stage volumetric energy machine, underlies the algorithm for investigating the effect of the installation's operating mode and parameters of the compressed gas on the dynamics and variability of the observed process. In addition, when modeling with a high level of adequacy, the analysis of the influence of the operating parameters and properties of the compressed gas on changes in its pressure, temperature and volume (density) is a guaranteed reliable procedure, the results of which will be used both in the creation of new design solutions for compression machines and in their further modernization.

It should be borne in mind that the formation of a plan and algorithm for designing a highly efficient compression machine is based on an understanding of how close the proposed technical solution is to the parametric base of an ideal analogue, and what should be considered before starting calculation procedures.

During the comparative analysis of an ideal compressor and a compressor with a polytropic (real) process taking place in it, attention was focused on the following provisions: a liquid piston is a continuous medium obeying Newton's law of friction and Hooke's law; the gas state in the working chamber of the hydraulic compressor at any given time is characterized by current mass values, intense parameters – pressure and temperature, as well as an extensive parameter – volume; the state of the upper layer of the liquid is determined by the values of volume and temperature.

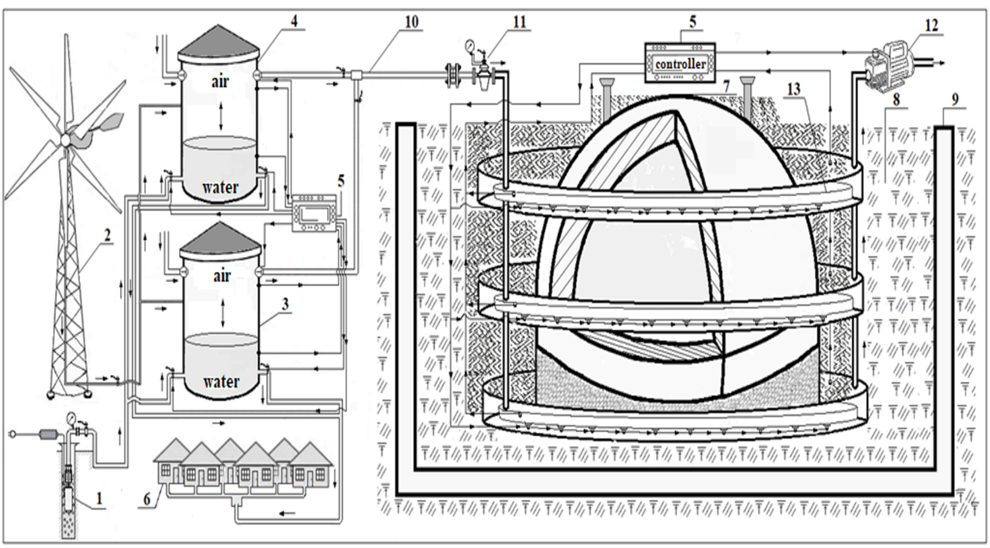

Fig. 1. Technological scheme of an energy-saving underground dome-shaped vegetable storage. 1 – submersible centrifugal pump; 2 – wind compressor; 3 – tank 1 (hydraulic compressor); 4 – reserve tank 2; 4 – controller; 6) technological facilities; 7 – dome–shaped underground vegetable storage; 8 – exploited soil mass; 9 - insulated partition; 10 – pipeline; 11 – compressed air dryer; pre–vacuum pump; 13 - receiver tube.

It is advisable to consider the working chamber as an open system in which thermodynamic processes are accompanied by both stationary and complex non-stationary thermo-gas dynamic phenomena. It is believed that the compressed air in the system can be taken as an equilibrium working fluid with a variable mass. In accordance with the principle of operation of a compressor with a liquid piston, air compression in the working chamber and its supply to the pneumatic accumulator receiver does not exceed the pressure range with a lower limit equal to atmospheric air pressure and an upper limit corresponding to boost pressure, limited only by the strength characteristics of the tank and receiver. Deviation in the compressor operation mode when air accumulates in the pneumatic accumulator receiver is not allowed

The control and innovation specifics of the proposed project lies in the fact that the predicted technical result involves not only effective energy saving when compressing air with a compressor with a liquid piston by combining it with a water supply system, but also expanding the functional orientation of the compressor to volumetric cooling of compressed air before feeding it into the injection line to technological facilities. For this purpose, the technical solution provides for integration with the air compression functionality of the function of generating its low-temperature flow based on the "adiabatic throttling" effect.

The proposed technical solution relates to the field of design and development of technical systems with an increased level of operational manufacturability, in particular to the creation of air compression systems to lower its temperature before being supplied to a technological facility, and the improvement in manufacturability was achieved by using water pressure in the water supply system of this facility.

To date, a method and device for its implementation are already known, where the soils under the supports of gas pipelines, oil pipelines, power transmission lines and other structures in the North in hard-to-reach areas are considered as a technological object, for which a cooled air stream is generated. The interest in the existing analogue is caused by the fact that when freezing the soil, a combined system of a wind turbine, an air compression device and a device for actually generating a low-temperature flow before feeding it to a heat exchanger located at a technological facility is used. The technical result declared in the description of the invention to the patent [12] is beyond doubt, but the kinetic energy of moving air is unstable and depends on a number of landscape and climatic factors, which, of course, is provided for in the technical solution by connecting the compressor unit to the power supply network. It is quite difficult to predict how often the pulse mode of energy transfer can be implemented, and how this will affect the operation of the "vortex tube", where high speeds of air divided into hot and cold streams are required. In addition, the profitability of innovating wind energy equipment into proven technological systems – namely, "technological" ones – remains open for almost all industries, requiring an intensive analytical approach.

A peculiar and interesting method is implemented by the device described in the patent for the utility model "Compressor with a liquid piston" [13], which consists in generating a high-pressure gas flow with an increased degree of compression, when the process being implemented tends from adiabatic to isothermal, which is achieved by replacing the moving parts of the compressor with liquid pistons, as well as technical combination in the working spaces of the tanks of the compression mechanism and the device for cooling the compressed gas and working fluid. According to the authors, this technical solution increases the reliability of the compressor with a liquid piston in conditions of high compression ratio and increases the efficiency of the compressor. It is advisable to comment on the disadvantage of this technical solution, taking into account not only the level of manufacturability of the invention at the time of its introduction into the production cycle, but also taking into account the development prospects, as well as adaptation to the technological needs of the user of innovative services. When it comes to increasing the efficiency of a compressor, it is necessary to specify which one is meant – mechanical, isothermal, adiabatic or polytropic. In this particular case, it is most acceptable to talk about the efficiency of a mechanical and isothermal compressor. But in both cases, when calculating efficiency, the designer must take into account the presence of an additional energy-consuming device that ensures the combined functioning of the entire gas compression system transported to the consumer: The mechanical efficiency must take into account losses in the pump pumping the working fluid between the tanks, and the isothermal efficiency will not be reliable without taking into account the power consumed by heat exchangers – coolers of gas or the same working fluid, which also applies to the pumping pump as a consumer of electrical or other energy. With such a functional combination, it is quite difficult to categorically talk about an increase in the efficiency of the compressor. Moreover, the statement about "increased reliability of a compressor with a liquid piston in conditions of high compression ..." raises a question when interpreting the definition of "large", since it is somewhat abstracted from the technical definition of this indicator as the ratio of the volume of the tank at the moment when the piston is in the lower dead position to the volume in the upper dead position. Rather, here we can mean a "slightly increased" degree of compression due to a partial increase in the gas entering the upper piston space of the tank, which is simultaneously cooled by a heat exchanger installed there. The physical essence is clear and there is no doubt – "cold compression", at the same time, the feasibility study in relation to direct supercharging with adequate energy costs remains an open question. Assessing the possibility of developing operational adaptability in terms of the functional adaptation of the prototype technical solution to expanded technological needs, it should be noted that there is a low probability of implementation through the combination of a compressor and a heat exchanger – cooler proposed by the authors of the process of supplying the consumer with gas at a temperature below ambient temperature. Cooled gas-air media are relevant and in demand in a number of industries in various sectors of the Russian national economy. The functional scheme of the device under study aggregates all implementing actuating elements, but inside the reservoir heat exchange with alternating gas-water contact is of the local "capsule" type, contributing to a targeted increase in the density of the gas trapped during the intake. To generate a low-temperature gas stream and transport it to the consumer, more powerful heat exchangers or modernization of the structure as a whole would be required.

Thus, the purpose of the research presented in this publication is to create the foundations for mathematical formalization of increasing the degree of functionality of the compression process and improving the operational adaptability of the device implementing this process by technically combining a water supply system with a low-temperature air flow generation system.

In accordance with this goal, a method has been developed for generating a high-pressure gas stream with an increased degree of compression, when the process under implementation tends from adiabatic to isothermal, and water piston compression is carried out by filling a sealed water storage tank with running water, from which the displaced air enters a pneumatic accumulator receiver and is pressurized until the water storage tank is released from water distributed to consumers. The gravitational release of the tank from water creates a vacuum in its internal cavity, where, at a technologically determined moment, forced air is returned from the receiver, which ensures the manifestation of the effect of "adiabatic throttling", which causes a sharp cooling of the air mass in the working cavity of the water storage tank. Due to the higher density of cold air, the vacuum state of the container is not completely leveled, which makes it possible, due to the pressure drop, to transport an additional portion of air from the environment into the container, the enthalpy of which does not significantly affect the temperature regime of the cooled internal volume. The next time the water storage tank is filled, the air cooled to a temperature below ambient temperature is squeezed into the main line for transportation to the consumer – to the technological facility, after which the water is gravitationally removed from the tank, distributing it to consumers in the same way. In this case, the inner cavity of the container is filled with air from the environment; then the process of compressing the air mass is repeated. In a gas compression device containing a water storage tank, a liquid piston, a pipeline, control sensors, switch taps and an automatic control system, according to the technical solution, the provided pneumatic accumulator receiver is fixed inside the water storage tank below the minimum water level technologically established for constant heat removal from the receiver during primary compression of the air mass in the upper piston space, and a controlled throttle to ensure a forced return of air to the upper piston space of the water storage tank.

It is the proposed set of specialized equipment and its placement in the inner cavity of the water storage tank that provides, according to the technical solution, an increase in the degree of implementation of a set of functions to fulfill the intended production tasks.

3 Results and discussion

In the presented system, the interaction of the working fluid with the environment occurs due to three types of interactions: thermal, the measure of which is the amount of heat transferred,Q; mechanical, the measure of which is the perfect working fluid of the work, δA; mass, the measure of which is the mass quantity of the transferred substanceM_i and the energy associated with it E_i [14, 15].

To assess the state of compressed air in the working cavity of a hydraulic compressor, the first law of thermodynamics was used for open thermodynamic systems of variable mass, since, being a special case of continuum equations, it is applicable to both reversible and irreversible stationary and nonstationary thermodynamic processes [15, 16].

It is necessary to analyze, as a matter of priority, the system of equations (3), which includes the equations of energy balance, the balance of mass of the working fluid (air), and the equation of state in thermal and caloric forms:

$$\begin{cases} dU = \sum_{i=0}^{n} dE_i + dQ - \delta A \\ M = \sum_{i=0}^{n} dm_j \\ P = P(\rho, \mathrm{T});\ U = U(\rho, T) \end{cases}, \tag{3}$$

where dU – change in internal energy, J; dE_i – an infinitesimal (elementary) amount of energy carried by an elementary mass dm_i, J; δQ – the amount of heat supplied to the working fluid, J; δA – the work performed by the working body, J; ρ – air density, kg/m^3; P – air pressure, Pa; T – thermodynamic air temperature, K.

When air flows from the atmosphere into the working cavity of the compressor and from it to technological facilities, the value of the specific amount of energy $dE_{i\mathrm{sp}}$ numerically equal to the specific enthalpy h_i compressed air in a given compressor volume. The enthalpy h_i of a compressible air medium characterizes the energy of an expanded thermodynamic system: working cavity-pipeline-receiver (WCPR) [16, 17].

Thus, the first law of thermodynamics for open thermodynamic systems of variable mass with the introduction of a natural argument – time t in general, it can be expressed in terms of the equation of the rate of change of the total internal energy of the working fluid (air):

$$\frac{dU}{dt} = \sum_{i=0}^{n} h_i dm_i + \frac{dQ}{dt} - \delta A\ , \tag{4}$$

where n – the number of external and internal drains or tributaries of the transferred mass.

The elementary value of the transported air mass dm_i:

$$dm_i = V_{j\text{воз}} d\rho_{jair} + \rho_{jair} dV_{jair}, \tag{5}$$

where dV_{jair} – an elementary change in the volume of air at j time, m^3; $d\rho_{jair}$ – elementary change in air density at j time point, kg/m^3; ρ_{jair} – the instantaneous (current) density of air at j time point, kg/m^3; V_{jair} – the instantaneous (current) volume of air at j time point, m^3.

The equation of the rate of change in the density of compressed air ρ_{jair}:

$$\frac{d\rho_{air}}{dt} = \frac{1}{V_{jair}}\left(\sum_{i=0}^{n} dm_i - \rho_{jair}\frac{dV_{jair}}{dt}\right), \tag{6}$$

The equation of the rate of change in the mass of the air flow M_{jair}:

$$M_{jair}(t) = \sum_{i=0}^{n} dm_i, \tag{7}$$

where M_{jair}– the instantaneous (current) mass of air at j time point, kg.

Since enthalpy is an additive quantity, then in expression (4) the value $\sum_{i=0}^{n} h_i dm_i$ it consists of an elementary amount of energy input $\sum_{i=0}^{n} h_{isum} dm_{isum}$ and an elementary amount of diverted energy $\sum_{i=0}^{n} h_{ires} dm_{ires}$ from compressed air:

$$\sum_{i=0}^{n} h_i dm_i = \sum_{i=0}^{n} h_{isum} dm_{isum} - \sum_{i=0}^{n} h_{ires} dm_{ires}, \tag{8}$$

The elementary amount of energy entering compressed air during external and internal mass exchange processes:

$$\sum_{i=0}^{n} h_{isum} dm_{isum} = \sum_{i=0}^{n} h_{i\mathrm{int}} dm_{i\mathrm{int}} + \sum_{i=0}^{n} h_{ivap} dm_{ivap}, \tag{9}$$

where $h_{i\mathrm{int}}$ – enthalpy of intake air from the atmosphere, J/K; h_{ivap} – enthalpy of water vapor entering the working cavity during evaporation of the liquid piston, J/K; $dm_{i\mathrm{int}}$ – the elementary mass of the intake air, kg; dm_{ivap} – the elementary mass of evaporated water, kg.

In turn, the elementary amount of energy removed from the compressed air:

$$\sum_{i=0}^{n} h_{ires} dm_{ires} = \sum_{i=0}^{n} h_{i\mathrm{pum}} dm_{i\mathrm{pum}} + h_{icon} dm_{icon} + h_{idis} dm_{idis}, \tag{10}$$

where $h_{i\mathrm{pum}}$ – enthalpy of the injected air from the working cavity, J/K; h_{icon} – enthalpy of compressed air condensed into water, J/K; h_{idis} – enthalpy of compressed air dissolved (separated from the main mass) in water, J/K; $dm_{i\mathrm{pum}}$ – the elementary mass of the injected air, kg; dm_{icon} and dm_{idis} – elementary masses of condensed and dissolved air, kg.

The interaction of compressed gas with the walls of the working cavity and the mirror of the liquid piston is defined by boundary conditions of the third kind, characterizing the law of heat exchange between the environment and air. To describe this law, we use the Newton-Richman equation (Newton's law of cooling) and write down the rate of change of convective heat transfer [11, 15]:

$$\frac{dQ_{\mathrm{wal}}}{dt} = \alpha_{\mathrm{wal}} S_{\mathrm{wal}} (T_{\mathrm{avewal}} - T_{jair}), \tag{11}$$

$$\frac{dQ_{\mathrm{liq}}}{dt} = \alpha_{\mathrm{liq}} S_{\mathrm{liq}} (T_{\mathrm{aveliq}} - T_{jair}), \tag{12}$$

where dQ_{wal} – the elementary amount of heat supplied from the walls of the working cavity to the air, J; dQ_{liq} – the elementary amount of heat supplied from the liquid piston to the air, J; α_{wal} и α_{liq} – coefficients of heat transfer to compressed air from the walls and mirrors of a liquid piston, W/m^{2}K; S_{wal} и S_{liq} – the set surface area of the walls of the working cavity and the mirror of the liquid piston, m^2; T_{avewal} и $T_{\mathrm{ave}liq}$ – the average current temperature of the walls of the working cavity and the upper (working) layer of water, К; T_{air} – the current temperature of the compressed air in the working cavity, К.

The elementary work of the process performed by compressed air δA is defined as:

$$\delta A = P_{jair} dV_{jair}, \tag{13}$$

where P_{jair} – instantaneous (current) air pressure at j time, Pa.

For the design calculation of compressed air pressure, taking into account that air compression occurs isothermically and without leaks, the equation of state is written as:

$$P_{\text{ini}}V_{\text{ini}} = P_{jair}(V_{jair} - dV_{jair}), \tag{14}$$

where P_{ini} – air pressure at the beginning of the compression process, Pa; V_{ini} – the volume of air at the beginning of the compression process, m^3.

Based on a generalized mathematical model of unsteady thermodynamic processes in an open system, the equation of the rate of change of the specific internal energy of compressed air is generally represented as follows:

$$\frac{dU}{dt} = \frac{1}{\rho_{air}V_{jair}}\left(\sum_{i=0}^{n}((h_{isum} - h_{ires}) - U)dm_i - \left(\frac{dQ_{\text{wal}}}{dt} + \frac{dQ_{\text{liq}}}{dt}\right) - P_{jair}\frac{dV_{jair}}{dt}\right., \tag{15}$$

Since the internal energy of a real gas is a function that depends not only on the temperature (for an ideal gas), but also on the density of the given gas, the equation of state in caloric form (3) can be used to write the equation of change in the internal energy of air [11]:

$$\frac{dU}{dt} = \left(\frac{\partial U}{\partial \rho_{air}}\right)_{T_{air}}\frac{d\rho_{air}}{dt} + \left(\frac{\partial U}{\partial t}\right)_{\rho_{air}}\frac{dT_{air}}{dt}, \tag{16}$$

where $\left(\frac{\partial U}{\partial t}\right)_{\rho_{air}} = C_V$ – mass isochoric heat capacity of air, J/kgK.

Therefore, the equation of the rate of change of air temperature:

$$\frac{dT_{air}}{dt} = \frac{1}{C_V}\left(\frac{dU}{dt} - \left(\frac{\partial U}{\partial \rho_{jair}}\right)_{T_{air}}\frac{d\rho_{jair}}{dt}\right), \tag{17}$$

As a result of substituting equations (6) and (15) into equation (17), it is possible to obtain the equation of the rate of change in air temperature $T_{air_}$in the expanded form [11]:

$$\frac{dT_{air}}{dt} = \frac{1}{C_V M_{jair}}\left(\sum_{i=0}^{n}((h_{isum} - h_{ires}) - U - \rho_{jair}\left(\frac{\partial U}{\partial \rho_{air}}\right)_{T_{air}}\right)dm_i - \left(\frac{dQ_{wal}}{dt} + \frac{dQ_{liq}}{dt}\right) - \frac{dV_{jair}}{dt}\left(P_{jair} - \rho_{jair}{}^{2}\left(\frac{\partial U}{\partial \rho_{air}}\right)_{T_{air}}\right), \tag{18}$$

where $M_{jair} = \rho_{jair}V_{jair}$ – the mass of air at j time, kg.

From the expression for the specific internal energy of air (15), it is easy to calculate the value of $\frac{\partial U}{\partial \rho_{air}}$, which obeys the equation of state for an ideal or real gas (air). If we adhere to the initial basic assumptions of the mathematical model, then $\left(\frac{\partial U}{\partial \rho_{\text{воз}}}\right)_{T_{\text{воз}}}$ =0. Then equation (18), taking into account (9), (10), will be written as:

$$\frac{dT_{air}}{dt} = \frac{1}{C_V\rho_{jair}V_{jair}}\left(\sum_{i=0}^{n}(h_{isum}dm_{isum} - h_{ires}dm_{ires} - Udm_i) - \left(\frac{dQ_{wal}}{dt} + \frac{dQ_{liq}}{dt}\right) - P_{ini}\frac{V_{wcpr}}{V_{wcpr} - dV_{jair}}\frac{dV_{jair}}{dt}\right., \tag{19}$$

In turn, the heat exchange of compressed air with the walls of the pipeline and receiver in the pipeline-receiver system will be determined by the rate of the amount of heat removed due to heat exchange with water (liquid piston) through the walls of the pipeline and receiver:

$$\frac{dQ_{pip}}{dt} = \alpha_{pip}(T_{\text{ave}pip} - T_{jair})S_{pip}, \tag{20}$$

$$\frac{dQ_{res}}{dt} = \alpha_{res}(T_{\text{averes}} - T_{jair})S_{res}, \tag{21}$$

where dQ_{pip} is the elementary amount of heat supplied from the walls of the pipeline to the air, J; dQ_{res} is the elementary amount of heat supplied from the walls of the receiver to the air, J; $T_{\text{ave}pip}$ and T_{averes} are the average current temperature of the walls of the pipeline and receiver (determined experimentally), K; S_{pip} and S_{res} are the set temperature the surface area of the walls of the pipeline and receiver, m^2; α_{pip} and α_{res} – the coefficient of heat transfer to compressed air from the walls of the pipeline and receiver, W/m^{2}K.

Thus, the rate of the amount of heat removed determines the heat exchange interaction between the compressed air and the cooling objects in the entire WCPR system at the end of the compression process:

$$\frac{dQ}{dt} = \frac{dQ_{res}}{dt} + \frac{dQ_{pip}}{dt} + \frac{dQ_{liq}}{dt} + \frac{dQ_{wal}}{dt}, \tag{22}$$

The internal energy of the overflowed mass in the WCPR system, which is at relative rest at the end of compression:

$$U = C_V T_{\mathrm{pum1}} dm_{pum1}, \tag{23}$$

where T_{pum1} is the current temperature of the injected air entering the receiver from the working cavity, K; dm_{pum1} is the elementary mass of air exiting the working cavity through the regulator, kg.

Taking into account the first stage of the operation cycle of a combined volumetric energy machine (the primary compression stage), which is close to ideal (p→0), expression (19) can be used to obtain an equation describing the rate of change in air temperature as a result of external influence over a short period of time in the WCPR system. Considering that $\sum_{i=0}^{n} h_i dm_i = 0$:

$$\frac{dT_{air}}{dt} = \frac{1}{C_V \rho_{jair} V_{jair}} \left[-\left(\frac{dQ_{res}}{dt} + \frac{dQ_{pip}}{dt} + \frac{dQ_{liq}}{dt} + \frac{dQ_{wal}}{dt} \right) - P_{ini} \frac{V_{wcpr}}{V_{wcpr} - dV_{jair}} \frac{dV_{jair}}{dt} - C_V T_{\mathrm{pum1}} dm_{pum1} \right], \tag{24}$$

Since the processes of compression and expansion occurring in energy machines must be considered as real, therefore, using the Van der Waals equation of state, the equation of the total specific internal energy of one mole of air is algebraically transformed as:

$$U = U_{\mathrm{id}} - \frac{a}{V_{jair}} = C_V T_{\mathrm{pum1}} - \frac{a}{V_{jair}}, \tag{25}$$

where a - the Van der Waals constant, which takes into account the forces of mutual attraction of molecules, Nm^4 / mol^2 (determined experimentally).

The rate of air temperature change in the WCPR system at the first stage:

$$\frac{dT_{air}}{dt} = \frac{1}{C_V \rho_{jair} V_{jair}} \left[-\left(\frac{dQ_{res}}{dt} + \frac{dQ_{pip}}{dt} + \frac{dQ_{liq}}{dt} + \frac{dQ_{wal}}{dt} \right) + \sum_{i=0}^{n} (h_{ivap} dm_{ivap} - h_{icon} dm_{icon} - h_{idis} dm_{idis}) - P_{ini1} \left(\frac{V_{wcpr}}{V_{wcpr} - dV_{jair}} \right)^m \frac{dV_{jair}}{dt} - \left(C_V T_{ivi1} \left(\frac{V_{wcpr}}{V_{wcp} - dV_{jair}} \right)^{m-1} - \frac{a}{V_{jair}} \right) dm_{pum1} \right], \tag{26}$$

Or

$$\frac{dT_{air}}{dt} = \frac{1}{C_V \rho_{jair} V_{jair}} \left[-\left(\frac{dQ_{res}}{dt} + \frac{dQ_{pip}}{dt} + \frac{dQ_{liq}}{dt} + \frac{dQ_{wal}}{dt} \right) + \sum_{i=0}^{n} (h_{ivap} dm_{ivap} - h_{icon} dm_{icon} - h_{idis} dm_{idis}) - P_{ini1} \left(\frac{V_{wcpr}}{V_{wcpr} - dV_{jair}} - \rho_{jair}^{2} \left(\frac{\partial U}{\partial \rho_{air}} \right)_{T_{air}} \right)^m \frac{dV_{jair}}{dt} - \left(C_V T_{ivi1} \left(\frac{V_{wcpr}}{V_{wcp} - dV_{jair}} \right)^{m-1} - \frac{a}{V_{jair}} - \rho_{jair} \left(\frac{\partial U}{\partial \rho_{air}} \right)_{T_{air}} \right) dm_{pum1} \right], \tag{27}$$

The rate of change of air pressure in the WCPR system in general [13]:

$$\frac{dP_{air}}{dt} = \left[\frac{1}{\rho_{jair}V_{jair}}\left(\sum_{i=0}^{n}((h_{isum}-h_{ires}) - U - \rho_{jair}\left(\frac{\partial U}{\partial \rho_{air}}\right)_{P_{air}}\right)dm_i - \left(\frac{dQ_{res}}{dt} + \frac{dQ_{pip}}{dt} + \frac{dQ_{liq}}{dt} + \frac{dQ_{wal}}{dt}\right) - \frac{dV_{jair}}{dt}\left(P_{ini1}\left(\frac{V_{wcpr}}{V_{wcpr}-dV_{jair}}\right)^{m} - \rho_{jair}{}^{2}\left(\frac{\partial U}{\partial \rho_{air}}\right)_{P_{air}}\right)\right] / \left(\frac{\partial U}{\partial P_{air}}\right)_{\rho_{air}}, \quad (28)$$

where m is the polytrope index.

The values of the derivatives $\left(\frac{\partial U}{\partial P_{air}}\right)_{\rho_{air}}$ and $\left(\frac{\partial \mathrm{U}}{\partial \rho_{air}}\right)_{\mathrm{P}_{air}}$ can be found from the equation of state in the caloric form U=f(p,p), depending on the type of equation of state used (Clapeyron, Dupree-Abel, Van-der Waals, Girn, Bogolyubov-Mayer, etc.).

The first stage of the operation cycle of a combined volumetric energy machine (the primary compression stage) should be considered close to ideal, i.e. subject to the assumptions of the proposed mathematical model:

$$\begin{cases} U = \frac{C_V}{R^{/}}\left(\frac{P_{jair}}{\rho_{jair}}\right) \\ \left(\frac{\partial U}{\partial \rho_{air}}\right)_{P_{air}} = \frac{C_V}{R^{/}}\left(-\frac{P_{air}}{\rho_{air}{}^{2}}\right) \\ \left(\frac{\partial U}{\partial P_{air}}\right)_{\rho_{air}} = \frac{C_V}{R^{/}}\left(\frac{1}{\rho_{jair}}\right) \end{cases}, \quad (29)$$

where $R^{/}$ is the individual gas constant, J/kgK.

Thus, the presented open thermodynamic WCPR system at the first stage (the primary compression stage) includes the following equations:

a) for conditions close to ideal:

1. The rate of air temperature change in the WCPR system:

$$\frac{dT_{air1}}{dt} = \frac{1}{C_V\rho_{jair}V_{jair}}\left[-\left(\frac{dQ_{res}}{dt} + \frac{dQ_{pip}}{dt} + \frac{dQ_{liq}}{dt} + \frac{dQ_{wal}}{dt}\right) - P_{ini}\frac{V_{wcpr}}{V_{wcpr}-dV_{jair}}\frac{dV_{jair}}{dt} - C_V T_{\mathrm{pum1}} dm_{pum1}\right], \quad (30)$$

2. The rate of change of air pressure in the WCPR system:

$$\frac{dP_{air1}}{dt} = \frac{R^{/}}{C_V V_{jair}}\left(-\left(\frac{dQ_{res}}{dt} + \frac{dQ_{pip}}{dt} + \frac{dQ_{liq}}{dt} + \frac{dQ_{wal}}{dt}\right) - P_{ini1}\frac{V_{wcpr}}{V_{wcpr}-dV_{jair}}\frac{dV_{jair}}{dt}\left(\frac{(C_V+R^{/}}{R^{/}}\right)\right), \quad (31)$$

3. The rate of change in the density of compressed air $\rho_{\text{воз}}$:

$$\frac{d\rho_{air1}}{dt} = \rho_{jair}\frac{dV_{jair}}{dt}\frac{V_{wcp}}{V_{wcpr}-dV_{jair}}, \quad (32)$$

b) for conditions close to real (a polytropic process):

1. The rate of air temperature change in the WCPR system:

$$\frac{dT_{air}}{dt}=\frac{1}{C_V\rho_{jair}V_{jair}}\Bigg[-\left(\frac{dQ_{res}}{dt}+\frac{dQ_{pip}}{dt}+\frac{dQ_{liq}}{dt}+\frac{dQ_{wal}}{dt}\right)+\sum_{i=0}^{n}(h_{ivap}dm_{ivap}-h_{icon}dm_{icon}-h_{idis}dm_{idis})-P_{ini1}\left(\frac{V_{wcpr}}{V_{wcpr}-dV_{jair}}-\rho_{jair}{}^{2}\left(\frac{\partial U}{\partial\rho_{air}}\right)_{T_{air}}\right)^{m}\frac{dV_{jair}}{dt}-(C_VT_{ivi1}\left(\frac{V_{wcpr}}{V_{wcp}-dV_{jair}}\right)^{m-1}-\frac{a}{V_{jair}}-\rho_{jair}\left(\frac{\partial U}{\partial\rho_{air}}\right)_{T_{air}})dm_{pum1}\Bigg],\quad(33)$$

2. The rate of change of air pressure in the WCPR system:

$$\frac{dP_{air1}}{dt}=\Bigg[\frac{1}{\rho_{jair}V_{jair}}(\sum_{i=0}^{n}((h_{isum}-h_{ires})-(C_VT_{ivi1}\left(\frac{V_{wcpr}}{V_{wcpr}-dV_{jair}}\right)^{m-1}-\frac{a}{V_{jair}})-\rho_{jair}\left(\frac{\partial U}{\partial\rho_{air}}\right)_{P_{air}})dm_i-\left(\frac{dQ_{res}}{dt}+\frac{dQ_{pip}}{dt}+\frac{dQ_{liq}}{dt}+\frac{dQ_{wal}}{dt}\right)-\frac{dV_{jair}}{dt}\left(P_{ivi1}\left(\frac{V_{wcpr}}{V_{wcpr}-dV_{jair}}\right)^{m}-\rho_{jair}{}^{2}\left(\frac{\partial U}{\partial\rho_{air}}\right)_{P_{air}}\right)\Bigg]/\left(\frac{\partial U}{\partial P_{air}}\right)_{\rho_{air}},\quad(34)$$

3. The rate of change in the density of compressed air $\rho_{воз}$:

$$\frac{d\rho_{air1}}{dt}=\frac{1}{V_{wcpr}-dV_{jair}}\left(\sum_{i=0}^{n}dm_i-\rho_{jair}\frac{dV_{jair}}{dt}\right),\quad(35)$$

The mathematical formalization of the primary air compression stage made it possible to optimize the technical solution in terms of structural and functional design and algorithmize the method implemented in it per cycle in two alternating cycles of filling the tank with water, simulating the operation of the piston during compression.

The device consists of: a controller 1 designed to dispatch the technological process implemented by the presented device; a water storage tank 2 filled with water from a well 3 by a pump 4 having a pipeline connection to an inlet pipe 5; shut-off valves 6 and 7; water supply lines 8 to the consumer; an electromagnetic valve 9 for communicating the inner cavity of the water storage tank with an air injection line 10 to technological facilities; an electromagnetic valve 11 for communicating the inner cavity of the storage tank with ambient air; an adjustable electric throttle 12 for controlling the air supply from the pneumatic accumulator 13 to the inner cavity of the storage tank; an electromagnetic valve 14 for communicating the inner cavity of the storage tank with the pneumatic accumulator; a liquid piston 15 for compressing air in the upper piston space 16; an upper water level sensor 17 and a lower water level sensor 18 in a water storage tank; an electric contact pressure gauge 19.

The device works as follows. Before starting the device, the controller 1 prepares the water storage tank 2 for filling it with running water, which is drawn from the well 3 by a pump 4 having a pipeline connection to the inlet pipe 5, where an electric shut-off valve 6 is installed, which in its initial position separates the inner cavity of the tank 2 and the pump 4. The controller 1 also switches to the closed position: the shut-off electric valve 7, disconnecting the tank 2 from the water supply line to the consumer 8; the solenoid valve 9, disconnecting the tank 2 from the air supply line to technological facilities 10; the solenoid valve 11, disconnecting the tank 2 from the ambient air; adjustable electric throttle 12, preventing premature escape of air from the pneumatic accumulator 13 into the working cavity of the tank 2. Only the solenoid valve 14 remains in the open position, which provides communication between the inner cavity of the tank 2 and the pneumatic accumulator 13 when air is squeezed out of the inner cavity of the tank 2 by a liquid piston 15.

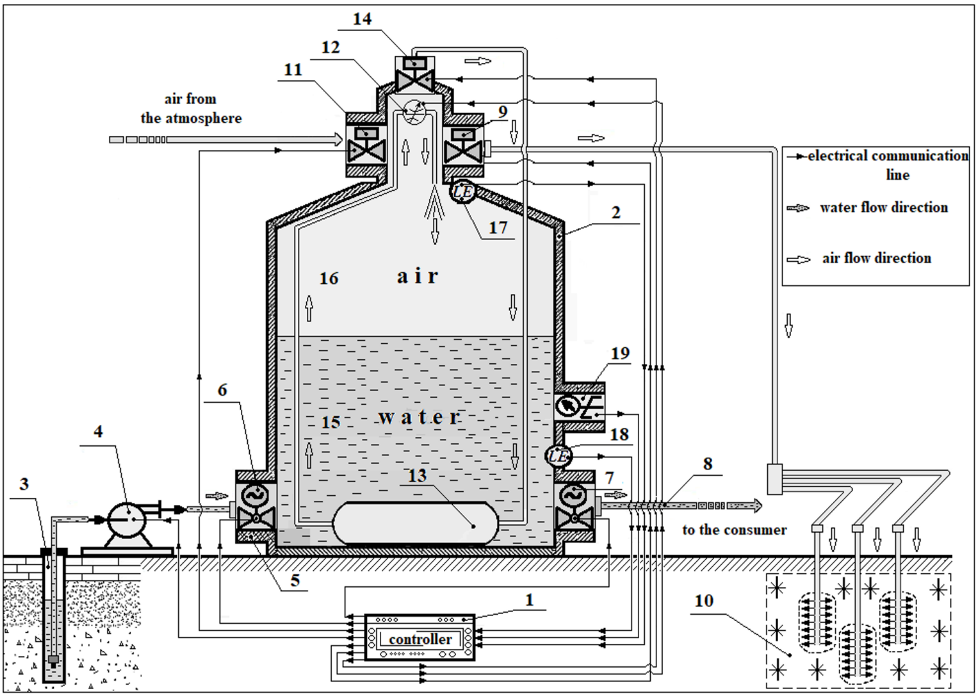

Fig. 2. Structural and functional diagram of the method of technical combination of a water supply system with a low–temperature air flow generation system and a device for its implementation

At the signal of the controller 1, the pump 4 is switched on and at the same time the shut-off electric valve 6 is opened. The working cavity of the tank 2 begins to fill with water from the well 3. As the tank 2 is filled, air is compressed in the piston space 16 and pumped into the pneumatic accumulator 13. Compression and injection of air will be carried out until the moment when the upper water level sensor 17 gives a signal to the controller 1 to stop the pump 4, close the shut-off electric valve 6, move the shut-off electric valve 7 to the open position and close the solenoid valve 14. Next, the compressed air pressure is released to 2 atm to the technological facilities 10. As a result, water begins to be gravitationally removed from the tank 2 into the water supply line to the consumer 8. When the tank 2 is freed from water, its internal cavity is evacuated, which will continue until the electric contact pressure gauge 18 sends a signal to the controller 1 to create a technologically provided vacuum in the surface space, after which the controller 1 opens the adjustable electric throttle 12 and forces compressed air from the pneumatic accumulator 13 into the already evacuated piston space 16, which ensures the manifestation of the effect of "adiabatic throttling", which causes a sharp cooling of the air mass inside the working cavity of the tank 2. The process of emptying the tank 2 of water will continue until the lower water level sensor 18 is triggered, the signal from which will serve as a command for the controller 1 to turn the electric shut-off valve 7 and the adjustable electric throttle 12 into a closed state, but at the same time to open the solenoid valve 11 to resume communication of the inner cavity of the tank 2 with ambient air. Since the air mass returned from the pneumatic accumulator 13 to the piston space 16 has a lower temperature relative to the ambient air and, accordingly, a higher density, at the moment of opening the solenoid valve 11, an additional portion of air will be transported into the inner cavity of the tank 2, which will fix the electric contact pressure gauge 19, after that, the controller 1 will close the solenoid valve 11, but open the solenoid valve 9, thereby disconnecting the communication of the working cavity of the tank 2 from the ambient air, but connecting the tank 2 to the air supply line to the technological facilities 10.

Next, in accordance with the functional algorithm, the controller 1 proceeds to the second cycle of filling the water storage tank 2: the pump 4 is simultaneously switched on and the shut-off valve 6 is switched to the open state; water from the well 3 begins to be pumped into the tank 2, squeezing air cooled to a technologically established level from the piston space 16 into the injection line air supply to technological facilities 10. The process will continue until the upper water level sensor 17 is triggered, after which the controller 1: closes the solenoid valve 9, opens the solenoid valve 11, turns off the pump 4, switches the shut–off valve 6 to the closed state, and the shut-off valve 7 to the open state, providing gravitational removal of water from the tank 2 to the water supply line to the consumer 8. The operation of the lower water level sensor 18 signals the completion of the second cycle of the technological cycle of the device for implementing the presented method, and in accordance with the functional algorithm, the controller 1 begins to prepare the water storage tank 2 for the next similar cycle.

The repetition of cycles will continue either until the targeted shutdown of the production process, or until the emergency de-energization of electrical or electronic equipment, or until the entire system of technical support for the operation of the innovative device goes into emergency mode.

The following parameters of the proposed device are set: the volume of the compressor's working cavity V_{worcav} = 221,4 m^3; the mass of water in the tank M = 213,3 tons when filled to a level of 8 m; pipeline diameter d_{pip} = 40 mm; pipeline length $L_{тр}$ = 16 m (taking into account the correction factor 1.6); pipeline volume V_{pip} = 0,02 m^3; receiver length L_{res} = 2 м; diameter of the receiver $d_{рес}$ = 1 m; volume of the receiver V_{res} = 1,03 $м^3$; surface area of the receiver $S_{res} \approx 3$ m^2 [18].

The workflow begins with the water rising from the minimum threshold of 1 m. The initial volume of air in the WCPR system is V_{ivi} = 194,3 m^3, and the final volume of air is V_{ini} = 8,1 m^3. The stage of primary compression is considered at n.o.: initial air pressure P_{ivi} = 1 atm; initial air temperature in the working cavity T_{ini}= 20 ° C.

The final air pressure in the receiver is P_{ini}= 30 atm. The temperature of the upper layer of the liquid piston increases from no more than plus 4 to plus 5 ° C. The compressibility of the upper layer of the liquid piston is not more than 1%. The air in the receiver is cooled at an average temperature of water from the well from plus 4 to plus 6 ° C.

A submersible centrifugal unit ECV 10-160-75 EDV 65-219 (160 m^3/h, pressure 75 m, casing diameter 250 mm) is recommended for filling the tank. The primary compression stage takes place in a time of t = 1.2 h with a water filling volume of V =186.2 m^3 up to a level of 8 m.

The volume of air at the beginning of the expansion process will be V_{ivi} = 8.1 m^3 when the pressure in the technical chamber is relieved to 2 atm. Pressure equalization in the working cavity and at the outlet of the water from the tank will occur closer to ≈ 4 m, at which point the volume of air will reach V_{ult} =109.4 m^3, while the volume of remaining water is V = 112 m^3. The exhaust pipe with a diameter of D = 200 mm is located at a height of 0.5 m from the Ground surface.

The volume of water at the beginning of the expansion process is V = 213.3 m^3. The container is emptied to a level of 1 m in a volume equal to V = 186.2 m^3. The average water consumption is Q = 905 m^3/s. Emptying to 1 m in t = 0.2 h =12 min.

A decrease in temperature after throttling by $\Delta T = 9{,}7\ ^{o}$ C, and a decrease in air volume due to cooling by about 2.5 times.

4 Conclusions

The technical result, expressed in energy savings (reducing the payback period to 2-3 years) when compressing air with a compressor with a liquid piston and expanding the compressor's functional focus on volumetric cooling of compressed air before feeding it into the injection line to technological facilities (increasing profitability to 60%), is achieved by combining a water supply system with a low-temperature air flow generation system. based on the "adiabatic throttling" effect, which ensures, when using the presented method, the implementation of a full working cycle during two alternating cycles of filling the water storage tank due to the hydropower of the mass of water pumped from the well and the electricity consumed by the pump during the cyclic period of operation of the innovative device.

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