Mathematical modelling and numerical study of recirculation membrane and membrane-refrigerated systems of compressed air dehydration

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Abstract. Mathematical modeling and numerical investigation of the process of drying compressed air was carried out in a hybrid membrane-refrigeration system, which consists of a compressor with a receiver, a Peltier refrigerator, and a gas separation membrane module. This system allows you to get dry air, dried to a dew point of -50°C and below without loss of compressed air since only condensed water is removed from the system. It is shown that reduced energy consumption decreases with increasing compression pressure. A comparison of the hybrid membrane-refrigerator system with a single-module membrane system has been carried out, the advantages of the hybrid system are shown.

1. Introduction
The content of water vapor in compressed air adversely affects the pneumatic system. Poor drying air in the brake systems of transport can lead to a catastrophe [1]. Condensation of water vapor in the gas transmission system can lead to the rapid wear of expensive equipment [2], [3]. The concentration of water vapor in the air is carefully monitored in the production of microelectronic equipment, pharmaceuticals, and other industries. Various methods can be used to remove water vapor from compressed air: refrigeration [4] technologies, absorption [5], adsorption [6], membrane [7] and hybrid [8] technologies.

It is promising to consider a membrane-refrigerator system for removing water vapor from compressed air since membrane technologies allow to obtain low dew points. The advantage of membrane technologies is that there is no phase transition in the processes of membrane gas separation. However, to avoid condensation of water vapor inside the membrane module, the gas must be pre-cleaned of mechanical impurities and dried. The use of a refrigerator at the entrance to the membrane module will achieve the required concentration of water vapor in the product stream while reducing the cost of installation and energy costs. The study conducted in this paper and the proposed mathematical model make it possible to simplify and accelerate the numerical study of the compressed air drying systems that are being developed.

2. Formulation of the problem
In this paper, we consider a hybrid membrane-refrigerator system (figure 1), consisting of a compressor with a receiver, a desiccant on Peltier elements [9], a membrane module. Air enters the
system, is compressed adiabatically, heats up and enters the receiver. In the receiver, the gas is cooled to the initial temperature, the water vapor condenses, the partial pressure of the water vapor becomes equal to the pressure of saturated water vapor at room temperature. From the receiver, the gas is supplied to the cold area of the dryer on the Peltier elements, a condensate is formed, which is removed from the system by means of a cyclone separator, the gas dried to the dew point temperature of the refrigerator is heated to the initial temperature by means of a hot area of the dryer on the Peltier elements and fed to the entrance to the membrane module. In the membrane module part of the gas and water vapor penetrate the selective membrane. For most modern membrane materials, the permeability of water vapor is higher than that of other gases. Thus, dried compressed air is released from the membrane module. The concentration of water vapor in the product stream will depend on the cooling temperature in the refrigerator and on the parameters of the membrane module: the area and permeability of gases through the membrane. This system allows you to extract water from the humid air without changing the amount of nitrogen and oxygen.

![Figure 1. Hybrid membrane-refrigerator system.](image)

We assume that the air consists of nitrogen (N₂) – 78.00 vol. %, oxygen (O₂) – 21.00 vol. %, and water vapor (H₂O) – 1.00 vol. %. With the help of the system (figure 1) it is necessary to obtain air-dried to a dew point temperature of -50°C, the performance of the system should be at least 2000 l(STP)/h. With the help of numerical studies, it is necessary to determine the values of flows and concentrations of nitrogen, oxygen and water vapor at each point. This paper considers membranes from PVTMS and PPO, gas permeability for these membranes are shown in table 1.

| Membrane | N₂ | O₂ | H₂O |
|----------|----|----|-----|
| PVTMS    | 120| 430| 3300|
| PPO      | 55 | 261| 2610|

3. Mathematical model

If the partial pressure of the vapor exceeds the pressure of the dry water vapor in the process of cooling the gas, then condensate forms. The experimentally obtained pressure values of saturated water vapor at temperatures from -79.9 to 99.9°C are specified in GOST (all-Union State Standard) 8.524 - 85 [10]. According to [11], the saturated vapor pressure of the pure phase:

\[ P_s'(t) = 6.112 \cdot \exp \frac{17.62 \cdot t}{243.12 + t}, \]

where \( t \) is the saturated steam temperature in °C. The pressure of saturated water vapor in humid air is calculated by the formula:

\[ P_s = \left(1.0016 + 3.15 \cdot 10^{-6} \cdot p_h - \frac{0.074}{p_h}\right) \cdot P_s'(t), \]
where $P_h$ is the compression pressure of the compressor. Pressure $P_s$ is calculated in hectopascals.

### 3.1. Compressor and receiver

When the air in the compressor is compressed from the initial pressure $P_l$, (in this work $P_l = 1$ atm), to the high-pressure $P_h$, the partial pressure of water vapor and other components (nitrogen and oxygen) will increase in $\varepsilon = P_h / P_l$ times, $\varepsilon$ - compression ratio. After compression, the gas enters the receiver, which is cooled to the initial temperature $t_1$ at high pressure. If the partial pressure of the water vapor in the receiver exceeds the saturated water vapor pressure at the initial temperature, some vapors will condense. It is obvious that the partial pressure of water vapor after the receiver depends on the compression ratio $\varepsilon$ and the initial concentration of water vapor at point 2 (see figure 1). It is assumed that at points 3, 4 and 5 the sum of partial pressures of all components (nitrogen, oxygen and water vapor) is equal to the pressure after compression in the compressor $P_h$.

The gas flow $G_2$ under pressure $P_1$ with water vapor concentration $C_{2,H_2O}$, where $C$ is the volume fraction of gas components, enters the compressor. Depending on the concentration of water vapor at point 2 and the compression ratio $\varepsilon$ in the compressor, two cases are possible: the first - in the receiver there is no condensation and the second - in the receiver there is condensation.

Consider the first case. If the partial pressure of water vapor in the compression process in the compressor will increase in $\varepsilon$ times, but will not exceed the saturated vapor pressure $P_s|$ at a temperature $t = t_1$, then condensation will not occur:

$$P_{2,H_2O} * \frac{P_h}{P_l} \leq P_s|_{(t = t_1)}.$$  \hspace{1cm} (3)

The pressure of saturated water vapor $P_s|_{(t = t_1)}$ at the temperature in the receiver is determined by the formula (2). It is important to take into account that condensation occurs in the receiver, in the compressor condensation is unlikely, since the gas in the compression process is heated, as a rule, to (50° ÷ 60°)C [12].

Dividing the left and right parts of equation (3) of the inequality by $P_h$, given that the concentration of water vapor $C_{2,H_2O}$ in gas at point 2 is determined as the ratio of the partial pressure of water vapor $P_{2,H_2O}$ to the total gas pressure $P_l$. Then we get:

$$C_{2,H_2O} \leq \frac{P_s|_{(t = t_1)}}{P_h}.$$  \hspace{1cm} (4)

In this case, the condition may be formulated differently. If the concentration of water vapor at point 2 $C_{2,H_2O}$ is less than the ratio of the saturated vapor pressure $P_s|_{(t = t_1)}$ at the gas temperature in the receiver (room or initial temperature) to the high pressure after compression $P_h$, the condensation of water vapor after compression in the compressor does not occur, the flow $G_3$ does not change and does not change the concentration of nitrogen, oxygen, and water vapor $C_3$:

$$G_3 = G_2 \hspace{1cm} (5)$$

$$C_3^i = C_2^i, \hspace{0.5cm} i = 1 \ldots m. \hspace{1cm} (6)$$

In equation (6) and further in work have designated a number of components of gas for m and, have assumed that the last component – water vapors.

The second case – after compression in the compressor and cooling in the receiver, condensation of water vapor occurs. If the water vapor concentration $C_{2,H_2O}$ (at point 2) is greater than the saturated vapor pressure ratio $P_s|_{(t = t_1)}$ at $t_k$ in the receiver to the high pressure after compression $P_h$, the water vapor condenses after cooling in the receiver, thus changing the partial pressure of nitrogen and
oxygen. The change in the amount of water vapor due to condensation leads to a change in the volume concentrations of the gas components, and the ratio of the partial pressure of nitrogen to the partial pressure of oxygen remains the same as at point 2. The partial pressure of the water vapor becomes equal to the saturated vapor pressure at the initial temperature:

$$P_{3}^{H_{2}O} = P_{5}|_{t_{1}}.$$  \hspace{1cm} (7)

Taking into account that the sum of partial pressures of nitrogen, oxygen and water vapor is equal to the pressure after compression in the compressor $P_{h}$ and the ratio of partial pressures of nitrogen to oxygen does not change after compression and condensation of water vapor, we write a system of equations:

$$\begin{cases} P_{SC} + P_{3}^{N_{2}} + P_{3}^{O_{2}} = P_{h} \\ \frac{P_{2}^{N_{2}}}{P_{3}^{N_{2}}} = \frac{P_{2}^{O_{2}}}{P_{3}^{O_{2}}} \end{cases}. \hspace{1cm} (8)$$

Having solved the system (8) with respect to $P_{3}^{N_{2}}$, the partial pressure of nitrogen or oxygen can be written as:

$$P_{3}^{i} = \left( P_{h} - P_{5}|_{t_{1}} \right) C_{2}^{i} \frac{1}{1 - C_{2}^{H_{2}O}}, \quad i = 1 \ldots m - 1.$$

The change in partial pressure leads to a change in concentrations:

$$C_{3}^{i} = \frac{1 - P_{5}|_{t_{1}}}{1 - C_{2}^{H_{2}O}} C_{2}^{i}, \quad i = 1 \ldots m - 1,$$

$$C_{3}^{H_{2}O} = \frac{P_{3}|_{t_{1}}}{P_{h}}. \hspace{1cm} (11)$$

Since the amount of nitrogen and oxygen after compression is preserved, it is possible to determine the flow $G_{3}$ (at point 3):

$$G_{3} = G_{2} \frac{1 - C_{2}^{H_{2}O}}{1 - C_{3}^{H_{2}O}}. \hspace{1cm} (12)$$

Thus, the gas flow $G_{2}$ with the concentration of components $C_{2}^{i}$ enters the compressor, at the output from the receiver we obtain the flow $G_{3}$ and the concentration $C_{3}^{i}$, which are determined by the formulas (5) and (6) if the condition (4) is met and condensation has not occurred or by the formulas (10), (11), (12), if the condition (4) is not met and condensation has occurred.

The energy consumption for gas compression in the compressor during adiabatic compression is found from the formula:

$$W_{c} = \frac{G_{2}}{n_{a_{a}} \eta_{a}} R T \left[ \left( \frac{P_{2}}{P_{1}} \right)^{\frac{\kappa - 1}{\kappa}} - 1 \right], \hspace{1cm} (13)$$

where $n_{a_{a}} = 0.85$ is the efficiency of the compressor operating in the adiabatic compression mode; $R$, is the universal gas constant $[J/(mol \cdot K)]$; $T = 273$ K is the temperature of the gas $[K]$; $\kappa = 1.4$ is the
adimabatic index for air, in this paper air is considered in the approximation of the ideal gas, so the compressibility factor $\mu = 1$.

3.2. Refrigerator on Peltier Elements

The refrigerator on Peltier elements consists of cold and hot areas and performs two functions. The first is the cooling of the gas to temperatures at which water vapor condenses, the second is the heating of the gas drained in the cold region to the initial temperature.

The refrigerator in front of the membrane module is installed to reduce energy consumption. It is important to note that the temperature to which the gas is cooled must be positive so that no ice is formed, usually $3^\circ$C.

At the entrance to the refrigerator enters the gas flow $G_3$ under pressure $P_h$ with a concentration of water vapor $C_{3H2O}$. Depending on the concentration of water vapor at point 3 and the temperature to which the gas is cooled in the refrigerator, two cases are possible: the first – in the refrigerator there is no condensation and the second – in the refrigerator, condensation occurs.

Consider the first case. If the partial pressure of water vapor at the inlet to the refrigerator (at point 3) does not exceed the pressure of saturated water vapor $P_{3s}$ at the temperature to which the gas is cooled in the refrigerator $t_r$, condensation will not occur. Saturated water vapor pressure $P_{3s}$ at the cooling temperature in the refrigerator is determined by the formula (2). Similarly, the condition for the compressor can be written condition for the refrigerator. If the concentration of water vapor $C_{3H2O}$ (at point 3) is less than the ratio of saturated vapor pressure $P_{3s}$ at the cooling temperature of the gas in the refrigerator to the high pressure $P_h$, the condensation of water vapor after cooling in the refrigerator does not occur. The $G_4$ stream can be defined by formulas similar to (4), (5), (6).

The second case is the condensation of water vapor after cooling in the refrigerator.

If the concentration of water vapor $C_{3H2O}$ (at point 3) is greater than the ratio of saturated vapor pressure $P_{3s}$ at the cooling temperature of the gas in the refrigerator to the high pressure $P_h$, the water vapor condenses after cooling in the refrigerator, while changing the partial pressure of nitrogen and oxygen (similar to (7)). The change in the amount of water vapor due to condensation leads to a change in the volume concentrations of the gas components. At the same time, the ratio of the partial pressure of nitrogen to the partial pressure of oxygen remains the same as at point 3. The partial pressure of water vapor after cooling and condensation becomes equal to the pressure of saturated vapor at the cooling temperature of the gas in the refrigerator (8).

Similar to the equations for the compressor, the partial pressure of nitrogen or oxygen after cooling in the refrigerator can be obtained by rewriting equation (10). The sum of the partial pressures of all components is the gas pressure after compression $P_h$. The change in partial pressure leads to a change in concentrations according to equations (11), (12). Since the amount of nitrogen and oxygen after compression is preserved, it is possible to determine the flow $G_4$ (at point 4 equation (13)). Thus, the inlet to the refrigerator receives a gas stream $G_3$ with a concentration of components $C_{3i}$, at the outlet of the refrigerator we have a flow $G_4$ with a concentration $C_{4i}$, which are determined by the formulas (5) and (6), if the condition (4) is met, in the opposite case, the formulas (10), (11), (12), (13) are used.

After cooling, the gas is heated to the initial temperature by the hot area of the refrigerator.

From the obtained equations for the compressor and refrigerator, you can notice a pattern and associate the output parameters (flow and concentration) from the block with the input – flow, and concentration of water vapor. We introduce an auxiliary function $B_{out}$, which shows how many times the output stream from the block is less than the input stream. Any drying (refrigerator) unit, in which the drying process occurs by reducing the partial pressure of water vapor to the pressure of saturated vapor at a temperature in the dryer, can be calculated using the function:

$$B_{out} = \min\left(1, \frac{1 - C_{in}^{H2O}}{1 - F_{out}}\right),$$

(14)
where $C_{in}^{H_2O}$ is the concentration of water vapor at the inlet, $P_s$ is the pressure of saturated water vapor, its value is determined from equation (2) or GOST [10]. The values of flows and concentrations at the outlet of the unit can be recorded through the function B and the concentration of water vapor $C_{in}^{H_2O}$:

$$G_{out} = G_{in} \cdot B_{out},$$

$$C_{out}^{i} = \frac{C_{in}^{i}}{B_{out}},$$

$$C_{out}^{H_2O} = 1 - \frac{1-C_{in}^{H_2O}}{B_{out}}.$$

If the partial pressure at the inlet $P_{in}$ to the desiccant is less than the saturated vapor pressure at the temperature in the desiccant $P_s|t$, then condensation does not occur, the flows and concentrations do not change. If the partial pressure at inlet $P_{in}$ of desiccant is greater than the saturated vapor pressure at a temperature in desiccant $P_s$, condensation occurs. Cooling in the refrigerator drying module is due to the operation of Peltier modules, which are a heat pump. The power of the heat pump depends on the enthalpy of gas supplied to the refrigerator. The enthalpy difference between the initial (at point 3) and final (at point 4) States of the drained gas determines the energy consumption of the refrigerator $W_r$. Enthalpy is determined by the temperature of the gas and its moisture content. The dependence of moisture content on the partial pressure of water vapor and pressure $P_h$ is defined as:

$$d = 621.98 \frac{P_{H_2O}^3}{P_h - P_{H_2O}^3}.$$

The enthalpy of a mixture of gases is the sum of the enthalpy of air components, the specific enthalpy of moist air is the sum of the enthalpy of dry air and water vapor.

$$J = J_c + J_n \cdot \frac{d}{1000},$$

where $J_c$ is the specific enthalpy of dry air, [kJ/kg dry air]; $J_n$ is the specific enthalpy of water vapor, [kJ/kg vapor]. Given (18) and (19), we write the enthalpy of moist air as:

$$J = 1.006t + \frac{(2501+1.85d)}{1000}, \text{kJ/kg dry air.}$$

Energy consumption for the refrigerator is determined by the formula:

$$W_r = (J_4|_{(d_4,t_r)} - J_3|_{(d_3,t_1)}) \cdot G_3 \cdot \frac{P_h M}{R T},$$

where $J_4|_{(d_4,t_r)}$ is the enthalpy of gas mixture at the outlet of the refrigerator in point 4, $d_4$ is the moisture content in the air at point 4; $t_r$ is the cooling temperature in the refrigerator; $J_3|_{(d_3,t_1)}$ is the enthalpy of gas mixture at the inlet to the refrigerator at the point 3, $d_3$ is the moisture content in the air at point 3; $t_1$ is the initial temperature of the gas; $M$ is the molar mass of air, $T = t_1 + 273$ is the temperature of the gas entering the refrigerator.

3.3. Membrane module

At the entrance to membrane module enters the flow $G_A$ with a concentration of $C_A$ under pressure $P_h$. Due to the pressure difference in the high pressure region and in the low pressure region, part of the
gas penetrates through the selective membrane, forms a permeate flow (point 6), usually the permeate flow pressure \( P_l = 1 \) atm, not penetrated flow coming out of the membrane – the retentate flow (point 5). Separation occurs due to the difference in permeability of one gas compared to another for this membrane material (table 1).

The mathematical model "crossflow" was chosen for the calculation of the membrane module" [13], [14]. This model assumes that there is no pressure drop along the fiber, the permeability of the component is constant at any pressure and temperature, within the framework of the task, such assumptions are permissible since the concentration of water vapor is much less than the concentrations of nitrogen and oxygen. The model describes the process of separation in half-fiber membrane modules and has the form:

\[
\frac{dZ^i}{dS} = -\pi^i P_h (C_q^i - \gamma C_l^i),
\]

where \( q \) is the local flow in the high-pressure region, \( C_q^i \) is the concentration of the \( i \)-th component in the flow \( q \), \( S \) is the area element, \( \pi^i \) is the permeability of the \( i \)-th component, \( \gamma = P_l / P_h \), \( C_l^i \) is the concentration of the \( i \)-th component penetrated through the membrane.

The initial conditions for the equations (22) and (25) have the form:

\[
Z^i \mid_{(s=0)} = G_4 C_4^i.
\]

The Cauchy problem (22), (23), (24), (25), (26) is solved numerically by the fourth-order Runge – Kutta method. The given parameters are the power flow \( G_4 \), the concentration of components in this flow \( C_4^i \), the permeability of each component \( \pi^i \), the pressure \( P_h \) and \( P_l \), the area of the membrane module \( S \). Solving the Cauchy problem we find the value of flow and concentration in the retentate:

\[
Z^i \mid_{(s=S)} = G_5 C_5^i.
\]

For the membrane module and for other units, the flow balances and quantities of the substance and the condition of mixing (30), (31) are fulfilled:

\[
G_4 = G_5 + G_6,
\]
\[
G_4 C_4^i = G_5 C_5^i + G_6 C_6^i,
\]
\[
G_2 = G_1 + G_6,
\]
\[
C_2 = \frac{C_1^i + G_6 C_6^i}{G_1 + G_6}.
\]

The numerical solution for a hybrid membrane-refrigerator system is found by the iteration method.

4. Results and discussion
Numerical study of a hybrid membrane-refrigeration system (figure 1) at room temperature 20°C with the temperature of the cooling gas in the refrigerator 3°C concentration of water vapor on the login
1 vol.%, system capacity 2000 l(STP)/h, dewpoint of dried gas -50°C, pressure in the low pressure 1 atm. During the study, the compression pressure in the compressor varied from 5 to 10 atm. Calculations were carried out for membranes from PVTMS and PPO. Given the dependence of the energy consumption of the compressor \( w_c/G_5 \), refrigerator \( w_r/G_5 \) and the amount of the pressure \( P_h \).

Figure 2. The dependence of the energy consumption of the compressor and the refrigerator from the compression pressure \( P_h \).

Figure 2 shows that the given energy consumption for cooling in the refrigerator is less than the given energy consumption of the compressor. The reduced energy consumption of the compressor decreases with the growth of high pressure. The use of a membrane with a greater permeability of water vapor reduces the energy consumption of the compressor, as the permeate flow decreases.

It should be noted that when the concentration of water vapor in the flow of the power system, equal 1 vol.% and room temperature 20°C, condensation occurs in the receiver and the refrigerator, and the composition of the mixture after the receiver and the refrigerator is more dependent on the temperatures in the receiver and the refrigerator, and not on the compression pressure \( P_h \) (see equations (1) and (2)). Figure 3 shows the dependence of the energy consumption of the compressor and the refrigerator, provided that the room temperature is 20°C, the cooling temperature of the gas in the refrigerator is 3°C, the concentration of water vapor at the entrance to the system is 0.1 vol.%, system capacity 2000 l(STP)/h, dewpoint of the dried gas -50°C, pressure in the low pressure 1 atm for the membrane of PVTMS.

It can be seen from figure 3 that a 10-fold decrease in the input water vapor concentration slightly reduces the energy consumption of the compressor since the flow at the inlet to the compressor decreases slightly. The reduced energy consumption of the refrigerator with a decrease in the concentration of water vapor at the inlet is 10 times reduced by 1.2 times. At the same time, the energy consumption of the refrigerator is not linearly dependent on the pressure, it can be seen from equation (31).

The application of the PPO membrane with a water vapor permeability of 12.6 times less leads to the need to increase the area by 2.6 times to reach the required dew point in the dried airflow.

It is important to show that the use of a hybrid membrane-refrigerator system in which the permeate flow of the membrane module returns to the entrance to the system is energetically more profitable than the use of membrane technologies without a refrigerator and without recirculation flows.

The single-module membrane system consists of a compressor with a receiver and a membrane module. The air is compressed in the compressor to a pressure of and heated, then cooled to the initial temperature in the receiver, with the water vapors condensed, the compressed air is separated and drained in the membrane module. The disadvantages of such a system include the fact that the flow of the membrane module retentate (at the output of the system) contains about 95 vol.% nitrogen and only 5 vol.% oxygens. The energy consumption of the compressor in a single-module membrane system is greater than the total energy consumption of the hybrid membrane-refrigerator system, this is shown in figure 4.
Figure 3. The dependence of the energy consumption units in a hybrid membrane-refrigerating system with a membrane made from PVTMS compression pressure $P_h$ in various concentrations of water vapor in the feed flow: a) – compressor; b) refrigerator.

Figure 4. Dependence of total energy consumption of hybrid membrane-refrigerator system (system 1) and single-module membrane system (system 2) with membranes and PVTMS and PPO.

The calculation of energy consumption was carried out under the condition that the water vapor content in the feed stream of the system $C_{H_2O}^{H} = 1$ vol. %. The study obtained that for drying to a dew point of -50°C with the necessary flow of the gas is equal to 2000 l(STP)/h, besides using a single-membrane system, the need to expend more energy and use more membrane area than for the hybrid membrane-refrigerated system.

5. Conclusion
1. A hybrid membrane-refrigerator drying system for compressed air is proposed, which consists of a compressor with a receiver, a refrigerator, and a membrane module and has a recirculation flow. This system allows obtaining dry air-dried to the dew point -50°C and below without loss of compressed air, as only condensed water is removed from the system.
2. Mathematical models for calculating the hybrid membrane-refrigeration system are presented. Describes the processes of condensation in the compressor with the receiver, drying in the refrigerator and the separation of multicomponent gas (air) in the membrane module.
3. It is shown that mathematical models of the condensation process in a compressor with a receiver and drying in a refrigerator are identical and are described by a single mathematical model.

4. Since there is a separate mathematical model for each block, hybrid and drying systems can consist of any number of such blocks and have several recirculation flows.

5. Conducted numerical studies of the proposed system.

6. It is shown that reduced energy consumption decreases with increasing compression pressure. The high permeability of water vapor will reduce the area of the membrane in the membrane module and reduce the energy consumption of the membrane module.

7. A comparison of the hybrid membrane-refrigerator system with a single-module membrane system has been carried out, and the advantages of the hybrid system have been shown.

6. References

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