Possibility of Thermomechanical Compressor Application in Desalination Plants

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Abstract. This article deals with estimation of thermocompressor operating possibility in desalination plant with mechanical vapour compressor. In this plant thermocompressor is used instead of commonly used centrifugal compressor. Preliminary analysis shows that such plant is able to operate, however, power consumption is 3.5-6.5 higher in comparison with traditional MVC plant. In turn, utilization of thermocompressor allows avoiding usual high-frequency drive of centrifugal compressor. Drives with frequency of 50 Hz are enough for thermocompressor when centrifugal compressor requires drives with frequency up to 500 Hz and higher. Approximate thermocompressor dimensions are estimated.

1. Introduction
Lack of the fresh water became a problem in recent years. According to researches, in 2025 about half of the states will suffer fresh water deficit [1]. Thus, generation of the fresh water is one of the main tasks which must be solved to ensure ecological safety of the Earth population. There are several ways of fresh water generation. One of the most promising is seawater desalination or waste water treatment. There are several ways to desalinate seawater. They include reverse osmosis (RO) [2], electrodialysis (ED) [3], freezing [4] and distillation [5-7]. Distillation plants in turn can be divided on Multi-effect distillation plants (MED) [8], multi-stage flash distillation plants (MSF) [9], multi-effect distillation plants with thermal vapour compression (MED-TVC) [10] and plants with mechanical vapour compression (MVC) [11]. Every type of has its own advantages and disadvantages. For example, MED plants can utilize waste heat of thermal power station, however, their efficiency is lower in comparison with RO plants. RO plants have a higher efficiency, however, they require often change of the membranes. ED plants are applicable on the water with relatively small salinity. Freezing plants are currently under development but in future they will allow using cold energy of liquid natural gas for desalination purposes. MVC plants have satisfactory energy consumption but they require careful design of the steam compressor which operation is crucial for plant.

Thermomechanical compressor is a device suited for gas compression by the heat adding in the closed volume [12]. From design point of view thermocompressor can be made with internal [13] or external regenerator [14] which can be represented in the form of annular gaps [15] and porous heat exchanger.

Thermomechanical compressor can be used in distillation plants as a substitution of mechanical compressor or as efficiency increase of other distillation plants.

2. Thermocompressor working principle
Technically thermocompressor consists of such elements as: piston with regenerator inside; piston rod; heat exchangers to compensate non-perfection of regenerator; input and output valves, cold and hot chambers. Principal scheme of the mechanical thermocompressor is presented on figure 1.

Working cycle of thermocompressor starts from high dead centre when both valves are closed and volume of the cold chamber is maximal (point 1). Piston starts to move downside pushing working fluid through regenerator to a hot chamber. Due to Amontons's Law of Pressure-Temperature, pressure is increasing because mean temperature of the working fluid inside the thermocompressor is increasing. When the pressure is high enough, output valve opens and working fluid with given pressure starts to be charged to a customer. When piston reaches low dead centre, output valve closes (point 3) and working fluid stops to be charged to a customer. Piston starts to move upside pushing working fluid back to a cold chamber. Pressure is decreasing due to decrease of the mean temperature inside the thermocompressor. When pressure drops low enough, input valve opens, charging new portion of working fluid inside the thermocompressor. Working cycle of thermocompressor is presented on figure 2.

![Figure 1. Principal scheme of the mechanical thermocompressor:](image)

Thus, for one working cycle thermocompressor rise the pressure of certain amount of gas $\Delta m$, which enters the TC through the input valve and leaves the TC from the output valve during one cycle.

![Figure 2. Working cycle of the thermocompressor.](image)
3. Thermocompressor coupled work with distillation plants
The main feature of the MVC plant is requirement of only one energy source which is electricity. Electric energy is transformed in thermal energy and potential energy of pressure which are further used to execute consequential phase changes seawater → steam → distillate. Design scheme of the plant with thermocompressor is presented on figure 3.

![Diagram of desalination plant with thermomechanical compressor](image)

**Figure 3.** Desalination plant with thermomechanical compressor 1 – evaporator stage, 2 – distillate chamber, 3 – brine chamber, 4 – water sprayer, 5 – circulation pump 6 – thermocompressor, 7 – mixer heat exchanger, 8 – water-distillate heat exchanger, 9 – water-brine heat exchanger

In this case steam enters thermocompressor instead of mechanical compressor. During displacement through regenerator both its temperature and pressure are raised. Let’s compare amount of energy necessary to raise the pressure in the same degree. Steam parameters of the last stage will be taken from the plant currently developed by Samara University together with “Metallist-Samara” OJSC. These parameters include: steam temperature 60 °C, steam pressure 20 kPa, required compression ratio is 1.6, steam flow rate is 0.111 kg/s. Other compression ratios are also added to compare the behavior of consumed work on bigger pressure range.

For the case of adiabatic expansion, amount of energy consumed by compressor will be determined as:

\[
N_{\text{comp}} = G \frac{k}{k - 1} R T_{\text{evap}} \left( \left( \frac{p^\text{II}}{p^\text{I}} \right)^{\frac{k - 1}{k}} - 1 \right),
\]

where \(N_{\text{comp}}\) – power consumption, \(G\) – steam flow rate, \(k\) – specific heat ratio, \(R\) – specific gas constant, \(T_{\text{evap}}\) – evaporation temperature, \(p^\text{I}\) – gas pressure in the input of the compressor, \(p^\text{II}\) – gas pressure at the output of the compressor.

For more real expansion, enthalpy difference is used:

\[
N_{\text{comp}} = G (i^{\text{II}} - i^\text{I}),
\]

where \(i^{\text{II}}\) and \(i^\text{I}\) – are enthalpies of steam before and after the compressor.

Finally, for the case of thermocompressor, it is necessary at first determine required temperature of gas to ensure required pressure:
$$T'' = T_{\text{evap}} \left( \frac{p'}{p''} \right),$$  \hspace{1cm} (3)  

where $T''$ is a temperature after thermocompressor.

In this case required power can be calculated as:

$$N_{\text{comp}} = G \cdot c_v \cdot (T'' - T_{\text{evap}}),$$  \hspace{1cm} (4)  

where $c_v$ – isochoric heat capacity of the steam.

Results of the power consumption comparison are presented on figure 4.

\[ \text{Figure 4. Comparison of the power consumption depending on compression ratio} \]

It can be seen that power consumption in thermocompressor is 3.5-6.5 higher and this relation is increasing with increase of compression ratio. However, despite this fact, thermocompressor has some advantages over centrifugal compressors which are commonly used in such desalination plants. These advantages include independence of compressor working parameters (frequency, dimensions) on compression ratio. In the meantime, operation of centrifugal compressor is determined by both dimensions and frequency and in case of aforementioned conditions its frequency reaches the values of 30000 r/min which negatively affects bearings.

Approximate dimensions of the proposed thermocompressor can be estimated. According to [12] relation of the flow rate to maximal mass contained in thermomechanical compressor depends on several parameters: temperature relation, regeneration efficiency, specific dead volumes. Working frequency also must be included in thermocompressor dimension analysis. Initial data is presented in table 1.

According to calculation non-dimensional mass output for given condition is about 0.18. That means that only about 18% of mass contained in thermocompressor will be charged per cycle.

Required mass flow rate can be determined as:

$$m_r = \frac{G}{f}. \hspace{1cm} (5)$$

Thus, maximal mass of gas inside the thermocompressor will be determined as:

$$m_{\text{max}} = \frac{m}{(dm / dm)} \hspace{1cm} (6)$$

Maximal mass in thermocompressor will correspond to position when cold chamber volume is maximal. This position also corresponds to an input pressure.
Table 1. Initial data for estimation

| Parameter                              | Value |
|----------------------------------------|-------|
| Input temperature $T_c$, K             | 333   |
| Hot chamber temperature $T_h$, °C      | 900   |
| Initial pressure $p_{in}$, kPa         | 20    |
| Required pressure $p_{out}$, kPa       | 32    |
| Regeneration efficiency $\eta_{reg}$   | 0.95  |
| Specific dead volume of hot chamber, $\varepsilon_h$ | 0.05  |
| Specific dead volume of cold chamber, $\varepsilon_c$ | 0.05  |
| Specific dead volume of regenerator chamber $\varepsilon_{reg}$ | 0.1   |
| Frequency $f$, Hz                      | 50    |

Solving of this equation along displaceable volume $V_d$ gives the value of 0.085 m$^3$, which is acceptable.

4. Conclusions
Application of the thermocompressor in desalination plants allows avoiding high-frequency centrifugal compressor which allows easing bearing loads. However, specific power consumption of thermocompressor driven MVC plant is higher in comparison with traditional MVC plant. Approximate dimensions of the thermocompressor for developed MVC plant are estimated and displaceable volume is equal to 0.085 m$^3$. More accurate investigation and optimization of the desalination plant working parameters are required for more accurate estimation of thermocompressor application potential in desalination plants.

Another direction towards which investigation can be carried out is numerical modeling of working process in software suited for fluid flow modeling such as ANSYS Fluent. Recent successes in both modeling of compressor operation [16-17] and piston machines modeling [18-19] allow assuming a possibility of successful modeling of working process in thermocompressor because it combines elements of both compressor and piston machines design.

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