

# Parametric optimization of the MVC desalination plant with thermomechanical compressor

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**Abstract.** This article deals with parametric optimization of the Mechanical Vapour Compression (MVC) desalination plant with thermomechanical compressor. In this plants thermocompressor is used instead of commonly used centrifugal compressor. Influence of two main parameters was studied. These parameters are: inlet pressure and number of stages. Analysis shows that it is possible to achieve better plant performance in comparison with traditional MVC plant. But is required reducing the number of stages and utilization of low or high initial pressure with power consumption maximum at approximately 20-30 kPa.

## 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], electro dialysis (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 desalination plant 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.

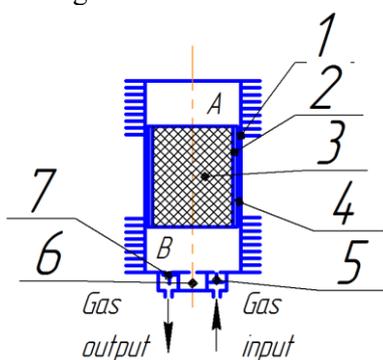
Thermomechanical compressor is a device suited for gas compression by the heat adding in the closed volume [12]. From design point of view thermocompressor can be made with internal [13] or external regenerator [14] which can be represented in the form of annular gaps [15] and porous heat exchanger.

Thermomechanical compressor can be used in distillation plants as a substitution of mechanical compressor or as efficiency increase of other distillation plants. Principal possibility of thermomechanical compressor operation within desalination plant was justified in article [16].

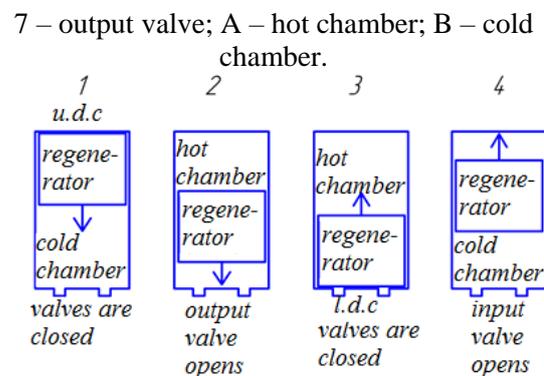


**2. Thermocompressor working principle**

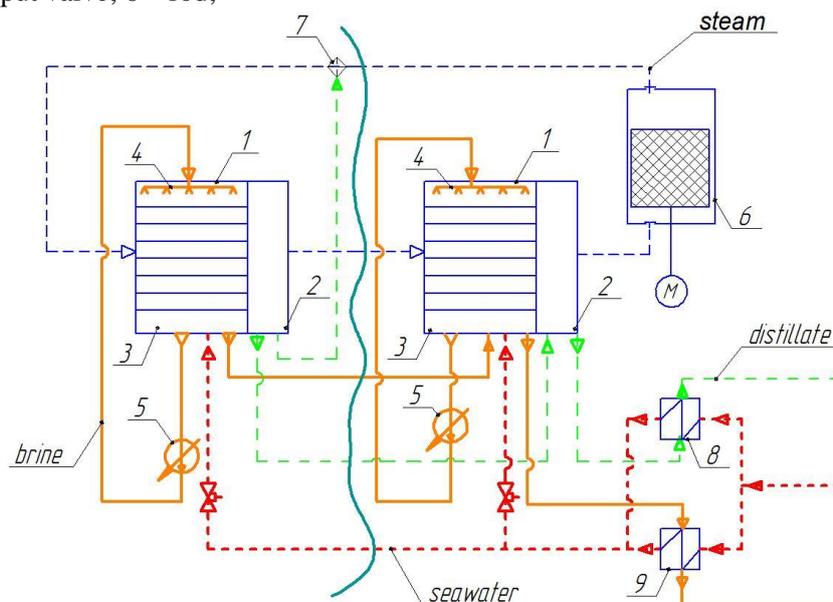
Working cycle of thermocompressor starts from high dead centre when both valves are closed and volume of the cold chamber is maximal (point 1). Piston starts to move downside pushing working fluid through regenerator to a hot chamber. Due to Amontons's Law of Pressure-Temperature, pressure is increasing 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:  
 1 – thermocompressor shell; 2 – displacer-regenerator shell; 3 – regenerator; 4 – gap; 5 – input valve; 6 – rod;



**Figure 2.** Working cycle of the thermocompressor.



**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

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.

### 3. The rmocompressor 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  $\rightarrow$  steam  $\rightarrow$  distillate. Design scheme of the plant with thermocompressor is presented on figure 3.

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 oC, steam pressure 20 kPa, required compression ratio is 1.6, steam flow rate is 0.111 kg/s.

Preliminary calculations show that at given configuration of desalination plant power consumption of the thermomechanical compressor is higher in comparison with traditional mechanical compressor. So main purpose of given research is to find such parameters and configuration of desalination plant where relation of power consumption of thermomechanical compressor to power consumption of mechanical compressor is minimal [16]. For further analysis let's investigate influence of different factors on thermomechanical compressor power consumption. Drinkable water output is kept constant as 2 m3/hour. Thermomechanical power consumption in ideal case is equal to:

$$N_{comp} = Q = G_{st} c_v (T_{out} - T_{in}) \quad (1)$$

Amontons's law states that in constant volume

$$\frac{p}{T} = const \quad (2)$$

Substituting (2) into (1) gives:

$$N_{comp} = Q = G_{st} c_v T_{in} \left( \frac{p_{out}}{p_{in}} - 1 \right) \quad (3)$$

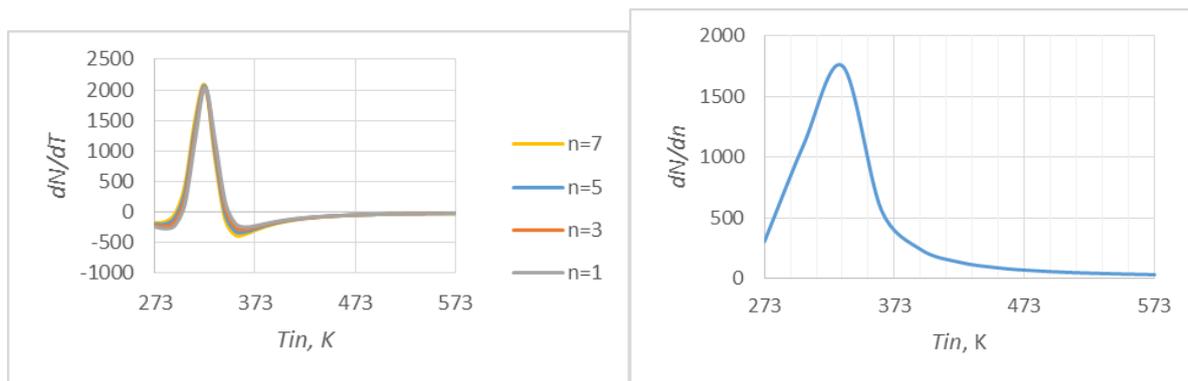
Dependency of saturation pressure on temperature is quite well described by empiric equation:

$$p_{sat} = 33.05 \cdot 10^{-5} (T - 273)^2 - 3.29 \cdot 10^{-2} (T - 273) + 0.993 \quad (4)$$

Previous researches show that optimal temperature difference in such evaporator-condenser is about 2 K [17]. So total temperature difference will be equal to 2n, where n is a number of stages. Considering that, thermomechanical power consumption will be equal to

$$N_{comp} = \frac{G_{wo}}{n} c_v T_{in} \left( \frac{33.05 \cdot 10^{-5} (T_{in} + 2n - 273)^2 - 3.29 \cdot 10^{-2} (T_{in} + 2n - 273) + 0.993}{33.05 \cdot 10^{-5} (T_{in} - 273)^2 - 3.29 \cdot 10^{-2} (T_{in} - 273) + 0.993} - 1 \right) \quad (5)$$

Let's analyse dependency of consumed energy of thermomechanical compressor on two factors: initial pressure and number of stages. Initial pressure is connected with initial temperature by steady dependency because parameters on this point are taken from saturation line. So, initial pressure can be replaced by initial temperature as comparison criteria. To estimate dependency of power consumption on initial temperature it is necessary to find derivatives of the equation (5). Dependency of the isochoric heat capacity on the temperature and pressure was neglected for first calculation but must be taken into account in further calculations. Plot of the  $dN_{comp}/dT_{in}$  derivative is presented on figure 4. Similarly, derivative of power consumption by number of stages was also calculated and is presented on figure 5.



**Figure 4.** Dependency of the derivative  $dN_{comp}/dT_{in}$  on temperature and number of stages

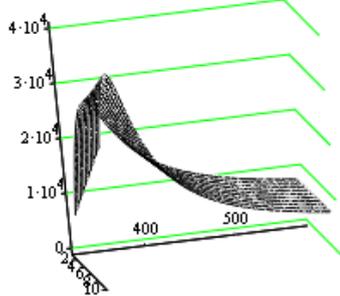
**Figure 5.** Dependency of the derivative  $dN_{comp}/dn$  on temperature

Analysis of the plot from figure 4 shows that derivative has a range of positive values between 287 and 346 K. So, conclusion can be made that maximal work of thermomechanical compressor will be at inlet temperature equal to 346 K, and it is highly recommended to avoid this temperature. Calculation of  $dN_{comp}/dn$  derivative shows that it almost does not depend on number of stages and depends only on temperature. Highest value of the derivative is achieved at temperature 333 K, so influence of the number of stage change will be maximal at this temperature. So, derivative analysis allow to make two main conclusions: inlet temperature must differ from 346 K and number of stages should be kept minimal, especially near inlet temperature equal to 333 K. This value actually is inlet steam temperature for the plant designed before, so direct replacement of the mechanical compressor to thermomechanical compressor gives high power consumption. Finally, power consumption dependency on inlet temperature and number of stages was determined and presented on figure 6. To compare optimized plant with thermomechanical compressor it is necessary to compare the values of their power consumptions. In ideal case power consumption of the mechanical compressor can be calculated as compression work in adiabatic process:

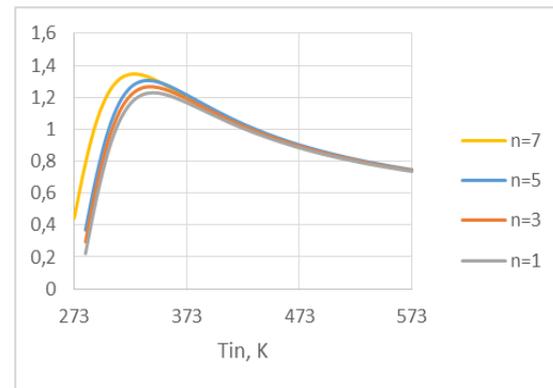
$$N_{comp} = G \frac{k}{k-1} RT_{evap} (p^I) \left[ \left( \frac{p^{II}}{p^I} \right)^{\frac{k-1}{k}} - 1 \right], \tag{6}$$

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

Plant with 5 stages and initial temperature equal to 333 K was considered as reference plant and all other plants were compared with it. Power consumption calculated by equation (5) was divided by power consumption, calculated by equation (6), resulted dependencies were plotted and presented at figure 7. It should be noted that inlet temperatures higher than 373 K require pressure higher than atmospheric one, so additional work must be spend on pressure increase of the water. The value of this work is relatively small but must be taken into account in further study. Figure 7 shows that it is possible to obtain better performance of the MVC plant with thermomechanical compressor than traditional MVC plant if appropriate optimization will be performed. However, several factors must be taken into account during careful investigation, such as deviation of real thermodynamic parameters of the steam from the equation (4), work spend on water pressurizing.



**Figure 6.** Surface of the power consumption dependence on temperature and stage number



**Figure 7.** Power consumption ratio dependency on inlet temperature and stages

#### 4. Conclusions

Parametric optimization of the MVC desalination plant with thermomechanical compressor shows that it is possible to achieve better plant performance in comparison with traditional MVC plant, however it is necessary to decrease number of stage to 1 and increase initial pressure. Decrease of the initial pressure to a certain point also helps to improve plant performance but low steam density at such pressure causes higher dimensions of the thermomechanical compressor and, thus, should be avoided.

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