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Experimental study on energy consumption of mechanical vapor recompression evaporation method for recovery of water from oily wastewater

Xuan Zhang, Xingkai Zhang*, Ruiquan Liao, Baocheng Shi, Lijuan Wu, Kai Liu

Petroleum Engineering College, Yangtze University, Wuhan 430100, China, Tel. +86-027-6911-1061; email: zhangxingkai001@163.com (X. Zhang), Tel. +86-130-9883-8041; email: 1031354536@qq.com (X. Zhang), Tel. +86-027-6911-1061; email: liaoruiquan@263.net (R. Liao), Tel. +86-027-6911-1061; email: shibaoch@126.com (B. Shi), Tel. +86-027-6911-1061; email: wljytz@126.com (L. Wu), Tel. +86-027-6911-1061; email: 4400173@qq.com (K. Liu)

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ABSTRACT

Direct reinjection and discharge of water mixed with oil leads to environmental pollution and exhaust heat waste. Of the various methods used to treat wastewater, the mechanical vapor recompression (MVR) evaporation method has obvious advantages in energy consumption and environmental protection. In this paper, the use of MVR for water recovery from water co-produced with oil was studied and analyzed through experimentation. Based on the design calculations for the MVR system, the MVR rig was set up and a series of experimental studies were conducted. The results show that when keeping the suction pressure constant at 0.035 MPa, the compressor power increases with the increase of discharge pressure. The experimental value of the compressor power was slightly greater than the theoretical value, with a deviation of less than 5%. The amount of distilled water produced rose slightly when the compressor discharge pressure was increased. The power consumption for producing 1 m^3 of distilled water increased with the increase of discharge pressure, which is also in accordance with the design condition. The experimental value of the power consumption per unit of distilled water is larger than the theoretical value, but the deviation is within 10%.

Keywords: Water recovery from oily wastewater; MVR method; Compressor; Distilled water flow rate; Power consumption

1. Introduction

Most of China's oilfield production in its final stages [1,2], results in an oily wastewater that contains more than 90% water. The world's daily water co-produced with oil is about 250 million barrels. Wastewater generated during oilfield exploitation is oily in nature [3], which contains various organic, inorganic, bacterial and radioactive impurities [4], and has high water content, high salinity and high temperature. Therefore, the wastewater must be treated before discharge. After the wastewater is treated, not only can a large amount of usable water be recovered, but damage to the environment from untreated wastewater discharges can be avoided. Moreover, the recovered water can be used to meet the water injection demands in the oil exploitation process, resulting in huge economic benefits.

At present, the water co-produced with oil is mainly managed by three methods: reinjection, efflux and reuse. In the late stage of production and development [5], water is typically injected into the reservoir to maintain the pressure of the reservoir. The oily sewage produced in the oil well that has been filtered through the soil is the main part of the reinjected water. This practice effectively recycles the sewage. At present, the oilfield treats the offsite discharges with three types of water treatment technology: physical treatment, chemical and physical treatment, and biological treatment [6,7]. Domestic and foreign oily wastewater treatment technologies generally include ion exchange, membrane method

* Corresponding author.

[8] and evaporation. The ion exchange method is suitable for the water co-produced with oil with low salinity and low total hardness. As the number of times of mining increase, the degree of mineralization correspondingly increases, and therefore a membrane method or an evaporation method is required.

The heavy oil extraction process of steam-assisted gravity drainage consumes a large amount of steam, and at the same time, a large amount of oily bottom water is separated after the crude oil is produced. Utilization of evaporation for the heavily salinized bottom water that can then be used to supply the steam boilers is the future development trend of oilfield bottom water treatment. The evaporation method [9,10] mainly includes multi-effect distillation (MED), thermal vapor compression and mechanical vapor recompression (MVR). Thiel et al. [11] analyzed and compared several methods for treating bottom water in high-salt oil fields, and comparing the energy consumption of various methods. Among them, the efficiency of MVC is about 32%, and the efficiency of MED is about 26% [11].

The MVR method is a heat pump technology, which is a highly efficient method for secondary steam heat recovery and is widely used in other countries, such as United States. The MVR system [12,13] consists of two main types of equipment: an evaporator and a steam compressor. The use of mechanical vapor recompression instead of thermal compression eliminates the need for an external heat source and requires only a small amount of electrical energy to continuously evaporate the wastewater. Although it is a mature technology, it is mostly used for seawater desalination and liquid waste treatment. However, in oil fields, the use of multi-effect evaporation with external heat-source steam consumes a lot of energy, requires a steam heat source, and is not suitable for large-scale water co-produced with oil treatment in oil fields. This paper reports the study of the development of a small MVR evaporation system designed such that the water co-produced with oil can be evaporated continuously to obtain distilled water without an external heat source.

2. MVR system and work principle

In the design, to simplify the analysis, we made the following assumptions:

- Disregard the increase in boiling point caused by salt in the bottom water and the decrease in temperature of the steam as it passes through the pipeline;
- The specific heat of the solution is constant and does not change with concentration;
- The steam does not contain salt;
- There is no heat loss;
- Disregard heat exchange between the evaporator, preheater, compressor and the environment;

2.1. MVR system and work principle

Fig. 1 shows a sketch of the MVR system with components labeled from 1 to 7. The main components of the MVR system include the evaporator, compressor and preheater. The process flow is also shown in Fig. 1. The water co-produced with oil with temperature T_1 enters the preheater at Location 1, where it is condensed. The resulting distilled water is heated to T_2 and then flows from Location 2 to the evaporator. The pressure inside the evaporator (P_3), and the latent heat from the water co-produced with oil, results in the formation of high-pressure steam. When the steam reaches the saturation temperature T_3, it evaporates. The evaporated steam at T_3 temperature enters the compressor from Location 3. After entering the compressor, the steam is compressed, heated and pressurized, reaching Location 4, where the temperature is T_4, and the pressure is P_4. The compressed superheated steam enters the other side of the evaporator, providing heat for evaporation of the oily wastewater, condensing the steam, and producing distilled water that exits the evaporator at Location 5. Finally, the condensed distilled water enters the preheater where it provides heat for the entering oily wastewater, and then flows out at Location 6.

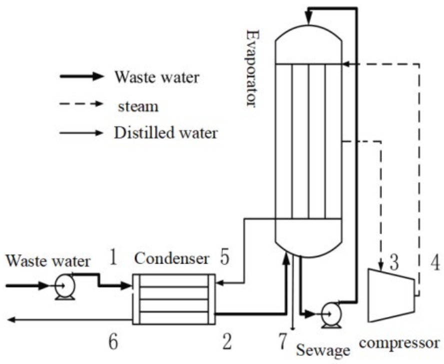

Fig. 1. MVR method to treat the water co-produced with oil design model.

2.2. Main equipment and performance parameters

2.2.1. Compressor and performance parameters

The pressure ratio ε of the compressor is the ratio of the compressor discharge pressure to the suction pressure, which is a key parameter of the system because it determines the heat exchange temperature difference of the evaporator. Different compression ratios and compressor discharge pressures determine the difference between condensing pressure and condensing temperature. Assuming that the compression process is an adiabatic process [14], the saturated steam is ideal gas, the compressor exhaust port is overheated, and the compressor discharge temperature is calculated as Eq. (1). The compression process is theoretically adiabatic definite entropy compression, but in practice, the pressure and temperature do not reach the final state of the reversible process, so it is necessary to assume the temperature deviation, assuming the deviation τ = 10 K.

$$T_4 = T_3 \times \varepsilon^{\frac{\kappa-1}{\kappa}} - \tau \quad (1)$$

Here κ, ε, τ, T_4 and T_3 represent the fixed entropy index, compression ratio, irreversible process temperature deviation/°C, compressor discharge temperature/°C, and compressor suction temperature/°C, respectively.

According to the above-mentioned parameters, the compressor power value is obtained according to Eq. (2):

$$W = P_3 \times q_v \times \frac{\kappa}{\kappa - 1}\left(\varepsilon^{\frac{\kappa-1}{\kappa}} - 1\right) \quad (2)$$

Here q_v is the compressor theoretical power/W, compressor volume flow/m^3 h^{-1}.

Considering that the function of the compressor is to increase steam temperature and pressure, the work done should be the difference in the water vapor before and after the compression, and the compressor work can be determined according to this principle.

The actual power of the compressor is calculated by Eq. (3):

$$W_0 = \frac{W}{\eta} \quad (3)$$

here W_0, η are the compressor actual power/W, compressor efficiency.

It is assumed that the superheated steam is not limited by the heat exchange capacity of the evaporation system, and the heat exchange can be sufficient, that is, the latent heat and sensible heat of the superheated steam condensate are all used to heat the bottom water to generate steam, and the heat exchange amount of the system is:

$$Q = h_3 - h_1 \quad (4)$$

Referring to the definition of the coefficient of performance in the refrigeration process [15], energy efficiency ratio is used to reflect how much revenue can be obtained by consuming a certain amount of compensatory energy, that is, the ratio of heat exchange to the actual power of the electric compressor. This is an economic indicator for evaluating the entire system. The compressor is the main energy-consuming equipment of the whole system. Under a certain compressor power consumption, the greater the heat exchange amount, the more of the water co-produced with oil is evaporated, and the more distilled water is obtained; that is, the economy is higher. The energy efficiency ratio is defined as follows:

$$\text{COP} = \frac{Q}{W_0} \quad (5)$$

2.2.2. Evaporator and performance parameters

The evaporative heat exchanger follows the mass and energy conservation calculations. The heat exchanger follows the conservation of the incoming and outgoing materials to complete the energy transfer. If heat dissipation is not considered, the overall energy is also conserved.

Compressor mass flow can be obtained from the compressor volume flow, expressed as follows in Eq. (6):

$$q_m = q_v \times \rho \quad (6)$$

here ρ, q_m are the vapor density/kg m^{-3}, compressor mass flow/kg h^{-1}.

As the water co-produced with oil evaporates, the concentration of dissolved matter will become higher. When it reaches a certain level, it needs to be discharged. As a result, the water inflow is not equal to the amount of evaporation. It is necessary to set a sewage discharge amount of approximately 10%. Then the mass flow rate of bottom water (M1), namely the mass flow rate at the inlet of the preheater is expressed as follows in Eq. (7):

$$M_1 = \frac{M_3}{90\%} \quad (7)$$

here M_1 and M_3 are the system inlet water co-produced with oil mass flow/kg h^{-1}, compressor inlet steam flow, is evaporation/kg h^{-1}.

The water inflow and outflow and the energy in and out of the evaporator are shown in Fig. 2. During the evaporation process, according to the quality of the water co-produced with oil and dissolved substances, you can get:

$$M_1 = M_3 + M_7 \quad (8)$$

$$M_1 \times X_1 = M_7 \times X_7 \quad (9)$$

here M_7, X_1, and X_7 are the sewage mass flow/kg h^{-1}, water co-produced with oil dissolved content, and dissolved content of the sewage.

The latent heat of condensation is transmitted to the water co-produced with oil which is nearly saturated from preheating through the heat exchange tube bundle, so that the water co-produced with oil evaporates and a portion of the secondary steam is generated. According to the assumption that the energy of the system is conserved, and there is no heat loss, that is, the heat absorbed by the evaporation process is equal to the heat released during the condensation process, the evaporator energy conservation equation can be obtained:

$$M_1 C_{P_1}\left(T_3 - T_2\right) + M_3\gamma_3 = M_3 C_{P_v}\left(T_4 - T_5\right) + M_3\gamma_4 \quad (10)$$

here C_{P_1}, γ_3, C_{P_v}, and γ_4 are the constant pressure-specific heat capacity at temperature at the inlet of the water co-produced with oil/kJ (kg K)$^{-1}$, latent heat of vaporization at evaporation temperature and pressure/kJ kg^{-1}, constant pressure specific heat capacity under evaporation temperature and

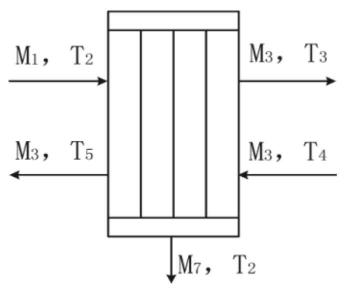

Fig. 2. Evaporator inlet and outlet model diagram.

pressure/kJ (kg K)$^{-1}$, latent heat of condensation at condensing temperature and pressure/kJ kg^{-1}.

3. Experimental rig description and measurement of parameters

This experiment mainly considers the successful operation of the system from two aspects of the device process and experimental equipment, obtains the operating parameters, compares with the design data, and verifies the feasibility of the experimental system. Fig. 3 is a schematic illustration of an experimental system.

In Fig. 3, the flow into the evaporator is at the saturation temperature, near atmospheric pressure. After the compressor is started, the pressure in the evaporator is rapidly reduced, and the experimental rig and equipment causes evaporation and generates steam, which enters the compressor for compression. The compressed superheated steam enters the heat exchanger tube in the evaporation tank, transfers heat to the experimental rig and equipment in the evaporation tank for evaporation, and the superheated steam in the tube condenses and is discharged into the distilled water tank. The regulating valve in Fig. 4 is used to adjust the compressor discharge pressure.

The function of the compressor is to suck low-temperature saturated steam, compress it into superheated steam and send it to the high-temperature condensation side of the evaporator to function as a heat pump. The compression ratio of the compressor determines the performance of the evaporation system. The pressure ratio of the compressor ranges from 1 to 4, and the compressor needs to be temperature resistant. The maximum volume flow rate needs to be greater than 5 m^3/h. According to the above requirements, a small Hitachi scroll compressor (model number SHW33TC4-U) was used in this experiment.

The main compression assembly in a Hitachi scroll compressor consists of two involute scrolls of the same shape with a 180° offset diagonally opposite. The steam enters the suction chamber from the compressor inlet and is then sucked into the crescent-shaped space that communicates with the suction chamber at the periphery of the scroll. As the crescent-shaped space of the outer ring closes, it no longer communicates with the inlet, and its crescent-shaped volume gradually moves to the center of the fixed scroll and shrinks, so that the gas is continuously compressed, and the temperature and pressure are raised. The compressor needs to adapt to the case where the current is too large and the internal temperature is too high. An overload relay is installed on the top of the scroll compressor. When a certain temperature is reached, the relay operates and the compressor stops running. The friction pair in the compressor generates heat during operation, and refrigeration oil is required to take away the heat generated by the friction, which is very important for ensuring the normal operation of the compressor. The Hitachi scroll compressor selected in this experiment is fully enclosed and uses internal fuel injection cooling.

Using horizontal-tube falling film evaporators, steam is condensed in the tube, and a liquid film is formed outside the tube. The preheating heat exchanger uses a plate heat exchanger. The preheating heat exchange amount accounts for less than 1/10 of the total heat exchange amount. In the experiment, the preheater and the evaporator are combined to simplify the experimental system. In addition, it is important to maintain the complete wetting of the outer wall of the heat exchange tube, so as to ensure that there is no dry area on the heating surface. Because the boiling heat transfer coefficient is high enough, the heat exchanger is directly placed in the bottom water, high enough to ensure that the heat exchange tube is completely wet and simplify the heat exchange device.

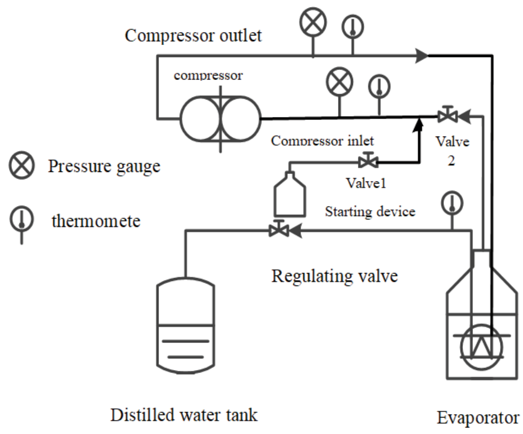

Fig. 3. Experimental system diagram.


Fig. 4. Compressor device diagram.

The evaporation device uses a heat preservation container and a heat exchange tube. The heat exchange tube is placed in the heat preservation container, according to the heat exchange area designated in the foregoing section. As shown in Fig. 5, the evaporator is composed of an inlet pipe and an outlet pipe, and a triple-heat pipe. In order to save space, the heat exchange tubes are wound into a disk shape, and the two ends of the heat exchange tubes are welded on the steam inlet pipe and the steam outlet pipe, respectively, and then placed in the heat preservation container. The three heat exchange tubes are round, pure copper tubes with an inner diameter of 0.004 m, an outer diameter of 0.006 and a length of 0.06 m.

In this experiment, a WSS-583 bimetal thermometer is used to measure the compressor suction temperature T_3, the compressor discharge temperature T_4, and the evaporation temperature T_3. The range of the thermometer is 0°C–200°C, the accuracy grade is 1.5. The pressure gauge uses a YB-150B

Fig. 5. Evaporation device physical map.

precision gauge to measure compressor suction pressure P_3 and compressor discharge pressure P_4. The range is 0–0.25 MPa, and the accuracy grade is 0.25. When distilled water flows into the distilled water tank, the amount of distilled water is measured by weight. The running time in the experiment is measured by a stopwatch, which is mainly used to calculate the amount of distilled water per unit time, thereby obtaining the electricity consumption per unit of distilled water.

The experimental measurements mainly include three energy consumption measures: power consumption, running time and distilled water volume. Measurements also included temperature and pressure values at the key points of the system, that is, temperature and pressure values at the inlet and outlet of the compressor. The evaporation temperature of the experimental rig and equipment in the evaporator is theoretically equal to the suction temperature of the compressor, but in actual operation, the system has air leakage and heat leakage, so additional measurement is required.

This paper only carried out preliminary research and performance verification experiments using the previous MVR system. Design and production of the specific pilot experimental device is underway.

4. Experimental results and deviation analysis

4.1. Verification experiment

The design of the test bench was based on the design criteria of suction pressure of 0.035 MPa and discharge pressure of 0.105 MPa, with experimental verification of design results. The evaporator heat exchange area of the experimental system is larger than the design value, which is to meet the requirement that the actual flow rate is greater than the design flow rate under variable operating conditions. The results of each test point in the experiment are presented in Table 1.

It can be seen from Table 1 that the suction pressure and the discharge pressure can be adjusted to meet the design conditions. The experimental results show that the difference between the compressor power and the design value is small, the amount of distilled water is less than the design value, and the power consumption to produce 1 m^3 of distilled water is greater than the design value. The evaporation temperature in the evaporator is almost equal to the saturation temperature at the compressor suction pressure. However, the temperature at the inlet and outlet of the compressor is much lower than the design value. This is because the vacuum in the evaporation is high, due to the suction of the compressor, which causes air to enter, so the influence on the system temperature is very large.

4.2. Variable working condition experiment

According to the above experimental process and measurement results, when the suction pressure of the compressor is 0.035 MPa, the discharge pressures are 0.105, 0.11, 0.115, 0.12, 0.125 and 0.135 MPa. The parameters such as compressor suction temperature T_3, evaporation temperature T_3, compressor discharge temperature T'_4, compressor power, distilled water production, and distilled water power consumption, vary with discharge pressure.

Fig. 6 shows experimentally measured compressor suction temperature T_3, evaporation temperature T'_3, compressor discharge temperature T'_4, and compressor discharge temperature as a function of discharge pressure. The measured evaporation temperature T'_3 is equal to the saturation temperature corresponding to the suction pressure of the compressor, indicating that the evaporation side of the evaporation system is operating normally. Since the suction pressure P_3 is constant, the evaporation temperature T'_3 is also unchanged. The temperature T_3 at the compressor inlet is lower than the evaporation temperature T'_3, and the compressor discharge temperature T'_4 is also much lower than the design value. This is due to the leakage of air into the

Table 1
Experimental results

Measuring point	Design value	Measurements
Compressor suction pressure P_3, MPa	0.035	0.035
Compressor suction temperature T_3, °C	72.68	53
Compressor discharge pressure P_4, MPa	0.105	0.105
Compressor discharge temperature T_4, °C	165.9	92
Evaporation temperature T_3, °C	72.68	72
Compressor power *W*, kW	0.07542	0.07798
Amount of distilled water *L*, kg/h	1.105	1.040
Evaporator heat exchange area *A*, m^2	0.008968	0.01260
Power consumption to produce 1 m^3 distilled water W', kWh/m^3	68.25	74.96

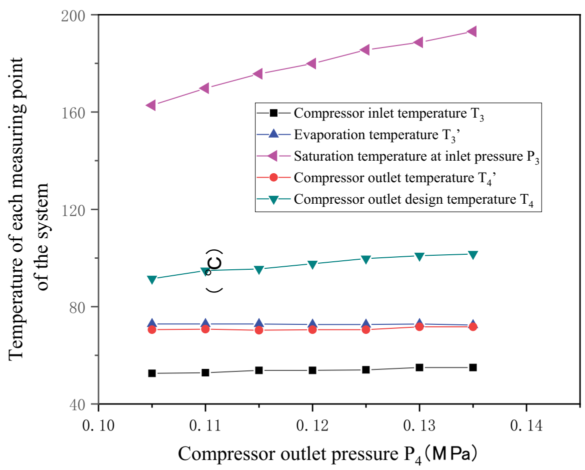

Fig. 6. Effect of discharge pressure on temperatures at various points in the system.

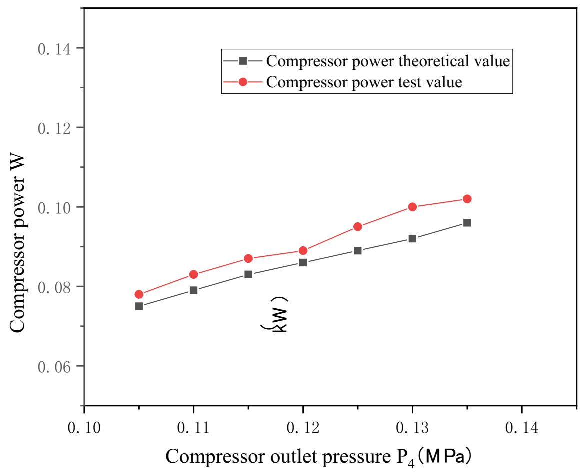

Fig. 7. Compressor power as a function of discharge pressure.

evaporator, and the low temperature of the air. The mixing into the water vapor causes the temperature at the inlet of the compressor to be lower than the evaporation temperature. The air will also exert partial pressure on the water vapor, resulting in a decrease in the relative pressure and density of water vapor in the compression chamber. The saturation temperature of water vapor is also lower than that under the measured pressure, resulting in the discharge temperature of the compressor being far below the design value. The saturation temperature corresponding to water vapor is also lower than the saturation temperature at the measured pressure, resulting in a compressor discharge temperature that is far below the design value. However, as the discharge temperature of the compressor is still higher than the evaporation temperature required by the evaporator, the compressed gas enters the evaporator tube to make the water outside the tube evaporate. The main heat exchange section is two sections, that is, the source of the main heat exchange amount is that the latent heat of condensation is transmitted to the outside of the tube for evaporation. The heat transfer temperature is the difference between the evaporation temperature and the saturation temperature under the condensing pressure. Therefore, although the superheated temperature at the outlet does not reach the design value, it has little impact on the system.

As shown in Fig. 7, when the compressor suction pressure P_3 is 0.035 MPa, the compressor power W increases as P_4 increases. This is because when the regulating valve is turned down, the pressure in the evaporation chamber is increased, and the compressor needs to do more work to increase the pressure. At the same time, it can be seen from the figure that the experimental value of the compressor power is slightly larger than the theoretical value, but the deviation is less than 5%. This part of the deviation is caused by the estimation of the efficiency when calculating the theoretical value. At the same time, it also shows that the compressor work is only related to the inlet and discharge pressure, and the mixed air in the water vapor has little effect on the compressor power.

As shown in Fig. 8, when the compressor suction pressure P_3 is 0.035 MPa, the amount of distilled water increases slightly as the compressor discharge pressure increases.

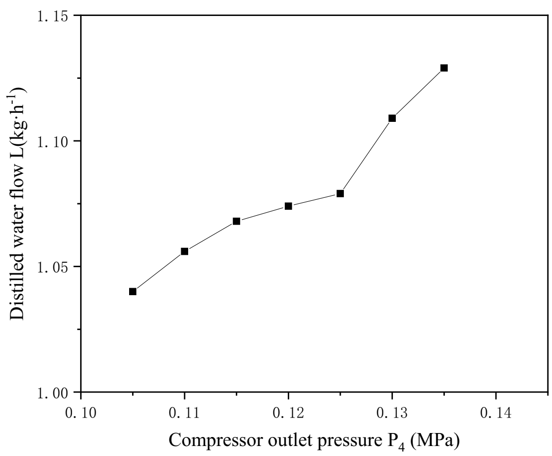

Fig. 8. Distillate flow rate as a function of discharge pressure.

Since the construction test bench refers to the working condition that the suction pressure is 0.035 MPa and the discharge pressure is 0.105 MPa, the amount of distilled water is constant under this parameter. However, the output of distilled water depends on the evaporation rate. When the discharge pressure increases, the compressor discharge temperature increases, and the heat exchange temperature difference increases. Therefore, under the same heat exchange area, the amount of heat exchange increases, that is, the more water co-produced with oil is evaporated.

As shown in Fig. 9, when the compressor suction pressure P_3 is 0.035 MPa, the power consumption to produce 1 m^3 of distilled water W' increases as the discharge pressure increases, which is consistent with the trend of the design value. The experimental value of the power consumption to produce 1 m^3 of distilled water is higher than the design value. Based on the experimental results, this is mainly because as the discharge pressure increases, the increase in compressor power consumption is relatively greater than the increase in the amount of distilled water. It can be concluded that the lower the compressor discharge pressure, the smaller the amount of electricity consumed to produce the same amount of distilled water, which is consistent with the design results.

4.3. Deviation analysis

Based on the above experimental results, the deviation between the experimental value and the theoretical value is calculated. As shown in Fig. 10, the compressor power deviation is about 5% on average, the design calculation formula is reasonable, and the theoretical value is accurate. The deviation of the power consumption for production of 1 m^3 of distilled water is within 10%, and the deviation is larger than the power deviation of the compressor. Mainly due to non-condensable gas entering the air in the evaporation chamber, water production is reduced, and the power consumption for production of 1 m^3 of distilled water is increased.

According to the above analysis of the experimental results, the experimental deviation mainly comes from the amount of air leakage, that is, due to the high degree of vacuum in the evaporation chamber, air enters the evaporation chamber, so that when the steam enters the compressor, the air temperature is low, causing temperature loss. The presence of air and superheated steam will also cause heat transfer to deteriorate, resulting in a decrease in heat exchange. At the same time, the air is not condensed, resulting in a decrease in the production of distilled water and an increase in the power consumption of the unit. In addition, although the experimental system uses thermal insulation measures for some exposed pipes, the system will dissipate heat to the surrounding environment, losing some of the heat and causing deviations.

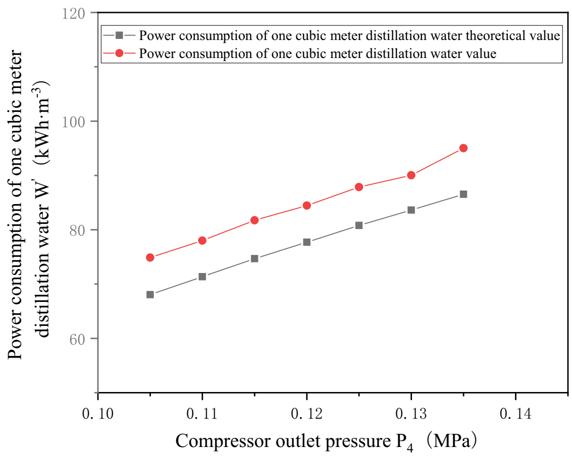

Fig. 9. Change of power consumption for production of 1 m^3 distilled water with change of discharge pressure.

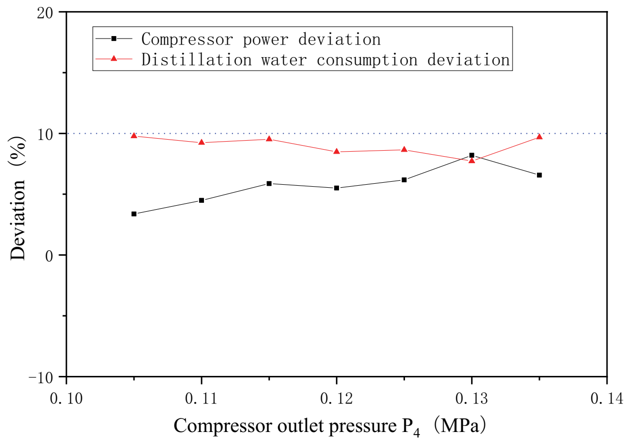

Fig. 10. Variation of experimental deviation with changes in discharge pressure.

5. Conclusion

In this paper, the feasibility of the use of the MVR method for recovering water from oily wastewater was verified and the performance of the MVR system was tested by small experiments. It was shown that the calculated results using design conditions are correct. It also provides a data foundation for future medium experiments. The evaporation temperature T'_3 is equal to the saturation temperature corresponding to the suction pressure of the compressor, although the temperature T_3 at the compressor inlet is lower than the evaporation temperature T'_3, and the compressor discharge temperature T'_4 is lower than the design value. However, since the main heat exchange amount is the latent heat of condensation, the main heat exchange temperature difference is the evaporation temperature and the saturation temperature under the condensing pressure, so the outlet superheated temperature does not reach the design value, but has little effect on the system.

When the suction pressure is kept constant at 0.035 MPa, the compressor power W increases as the discharge pressure P_4 increases, which is consistent with the trend of the design conditions. At the same time, the experimental value of the compressor power is slightly larger than the theoretical value, with a deviation of less than 5%. The power consumption to produce 1 m^3 of distilled water W' increases with the increase of the discharge pressure P_4, which is consistent with the trend of the design conditions. Moreover, the experimental value of the power consumption to produce 1 m^3 of distilled water is higher than the theoretical value, but the deviation is within 10%.

Acknowledgments

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