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An open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water

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Abstract

This paper presents an open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water, and proves its feasibility through performance simulation. Pinch technology is used to analyze the cooling of the wet air after compressor and the water used for cooling wet air after compressor. Its refrigeration depends mainly on the sensible heat of air and the latent heat of water vapor, its performance is more efficient than a conventional air-cycle, and the utilization of turbo-machinery makes it possible. The adoption of this cycle will make deep freeze easily and reduce initial cost because very low temperature, about −55 °C, air is obtained. The sensitivity analysis of coefficient of performance to the efficiency of compressor and the efficiency of compressor, and the results of the cycle are also given. The simulation results show that the COP of this system depends on the temperature before turbine, the efficiency of compressor and the efficiency of compressor, and varies with the wet bulb temperature of the outdoor air. Humid air is a perfect working fluid for deep freeze with no cost to the user.

Keywords: Turbo-machinery; Air cycle; Deep freeze units; Natural working fluid; Refrigeration

1. Introduction

Many aquatic products such as shrimps and abalones are produced in south China in large quantities. The income derived from these products is normally minimal due to inadequate conservation and storage facilities and also lack marketing structures. Deep freezing of aquatic products is the most widespread preservation technique and becoming more and more an alternative to marketing fresh aquatic product since the demand for high quality deep frozen aquatic products is permanently increasing all over the world. Ammonia refrigerating system or liquid nitrogen has been applied in these aquatic product freezing preservation factories recent years.

The air compression refrigeration cycle was studied long ago. Several disadvantages prevented air from being used as a working fluid in refrigeration. These included the low volumetric refrigerating effect, which may result in a large compressor, and the low COP due to the inefficiency of the compressor and expander. After the invention of chlorofluorocarbons (CFCs) in the 1930's, people paid little attention to actual air compression refrigeration.

Recently, as a result of the depletion of the ozone layer by chlorofluorocarbon (CFC) and the pressure of increased concern about environmental protection, research on the air refrigeration cycles had a renaissance [1–3]. Optimizations of air cycles are also carried out using finite-time thermodynamics (FTT) [4,5] or entropy-generation minimization (EGM) [6,7].

Chen et al. investigated the cooling-load versus COP characteristics of a simple [8], a regenerated [9,10] air refrigeration cycle with heat-transfer loss and/or other irreversibilities. Luo et al. [11] optimized the cooling-load and the COP of a simple irreversible air refrigeration cycle by searching for the optimum pressure-ratio of the compressor and the optimum distribution of heat conductance of the hot and cold-side heat exchangers for the fixed total heat-exchanger inventory.

Spence et al. reported the design, construction and testing of an air-cycle refrigeration unit for road transport [12] and perfor-

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Nomenclature

B	wet air pressure	Pa
d	humidity ratio of wet air	g/kg (da)
h	enthalpy of wet air	kJ/kg (da)
q_1	the heat rejected to outdoor air per kilogram dry air	kJ/kg (da)
q_2	the refrigerating capacity per kilogram dry air	kJ/kg (da)
P	pressure	Pa
t	temperature	°C
T	temperature	K or °C
T_{wet}	wet bulb temperature	K or °C
w_c	ideal input work of a compressor	kJ/kg (da)
w_t	ideal output work of a turbine	kJ/kg (da)
w_m	practical work consumed by the system	kJ/kg (da)
R	gas constant	kJ/kg K
n	the poly-tropic exponent	
η_c	efficiency of a compressor	
η_t	efficiency of a turbine	

Subscripts

da	dry air
vap	the water vapor in moist air
s	saturated

mance analysis of a feasible air-cycle refrigeration system for road transport [13].

Zhou et al. presented the cooling load density analysis and optimization for an endoreversible air refrigerator [14], the cooling load density characteristics of an endoreversible variable-temperature heat reservoir air refrigerator [15], the theoretical optimization of a regenerated air refrigerator [16] and the cooling-load density optimization for a regenerated air refrigerator [17]. Tu et al. investigated the optimization of cooling load and COP for a real regenerated air refrigerator [18,19].

Chen and Su gave the exergetic efficiency optimization for an irreversible Brayton refrigeration cycle [20]. Chen et al. presented a performance optimization for an irreversible variable-temperature heat reservoir air refrigerator [21]. Williamson and Bansal studied the feasibility of air cycle systems for low-temperature refrigeration applications with heat recovery [22].

Angelino et al. studied the prospects for real-gas reversed Brayton cycle heat pumps [23]. Wu et al. optimized steady flow heat pumps [24]. Chen studied the heating load vs. COP characteristics for irreversible air-heat pump cycles [25].

Ni et al. presented the performance analysis for endoreversible closed regenerated Brayton heat pump cycles [26]; Chen et al. gave the performance analysis of a closed regenerated Brayton heat pump with internal irreversibilities [27] and the performance of real regenerated air heat pumps [28].

With the development of the aeronautical industry, highly efficient axial compressors and turbines have become a reality. At present, the stagnation iso-entropic efficiencies of a single stage axial compressor and a turbine can reach 0.88–0.91 [29]. Nowadays, high-speed fans are commonly used in ordinary air conditioning systems.

However, the water vapor in the working fluid was not considered in all the above research on air compression refrigeration cycles. The equipment used was centrifugal compressor and centripetal turbine, both of which have lower efficiencies than the axial compressor and turbine. The amount of the water extracted from high pressure wet air can reach 18–30 g/kg (da). And the amount of the latent heat rejected from the condensed vapor, about 45–75 kJ/kg (da), exceeds the sensible heat from air, 30–50 kJ/kg (da).

Hou and Li [30,31] presented both an open heat pump and axial-flow air–vapor compression installation for air conditioning, in which wet air is a working fluid and the axial compressor and turbine were used, but these methods have not yet attracted widespread attention.

Braun et al. [32] gave an energy efficiency analysis of air cycle heat pump dryers where the feasibility of an air heat pump (reversed Brayton) cycle for tumbler clothes dryers was investigated. An air cycle heat pump dryer with practical components was found to be capable of significantly improved efficiency as compared with conventional dryers.

Hou et al. has successfully applied Pinch technology [33–36] and exergy analysis [37] to performance optimization of solar humidification-dehumidification desalination process. Hou et al. gave an open air–vapor compression refrigeration system for air conditioning and hot water cooled by cool water [38] and an open air–vapor compression refrigeration system for air conditioning and desalination on ship [39].

Hou and Zhang [40] presented an axial-flow air–vapor compression refrigerating system for air conditioning cooled by circulating water, in which wet air is a working fluid, an axial compressor and a turbine were used and circulating water cooled the wet air. The paper proves its feasibility through performance simulation and also indicates its advantages. These advantages include the possible simplification of air-conditioning systems, the possible reduction of the amount of the cost of an air-conditioning system and the possible maximization of comfort in air-conditioned spaces.

The aim of this paper is to present an improved system, which is an open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water. In this open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water, the air left from freeze chamber is used to cool water, and then the cooled water we got is used to cool the wet air before turbine further. So, we could get the lower wet air temperature before turbine and higher COP.

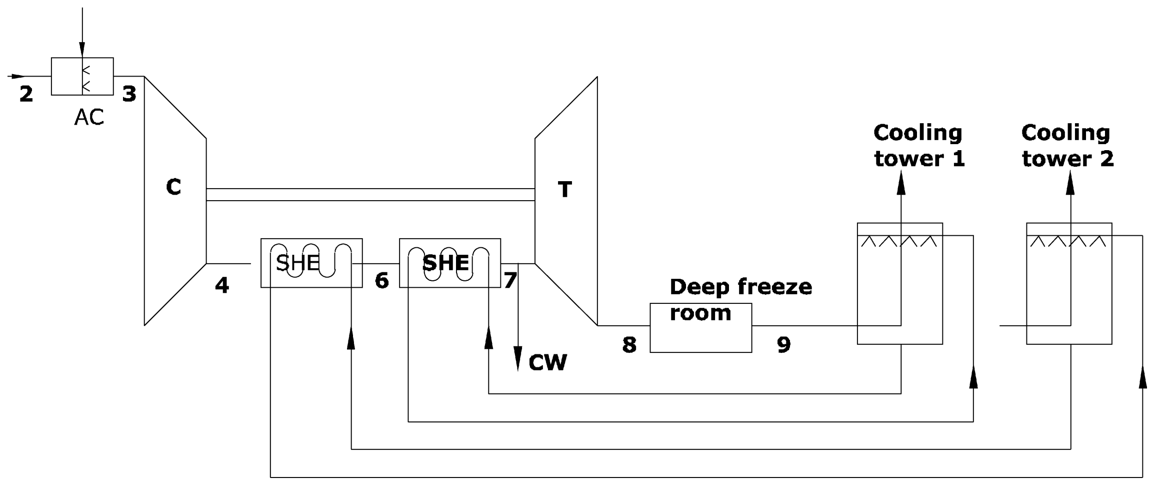

Fig. 1. Circuit diagram of an open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water. AC atomizing chamber, SHE surface heat exchanger, CW condensed water, C axial compressor, T axial turbine.

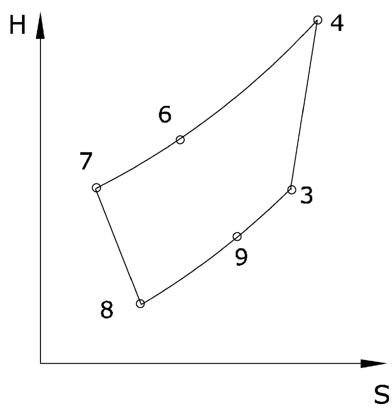

Fig. 2. Representation of an open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water on enthalpy–entropy coordinates.

2. System

The circuit diagram of an open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water is shown in Fig. 1. The representation of the open reversed Brayton cycle with regeneration using moist air on enthalpy–entropy coordinates is shown in Fig. 2.

The outdoor air at 2 is drawn into the atomizing chamber (AC), cooled into the saturated air at 3 with some fine water droplets and then compressed by an axial compressor (c). A flow of compressed air at 4 with higher temperature, T_4, and high pressure, P_4, is obtained. Then the compressed air at 4 is firstly cooled into the saturated air at 6 with a temperature of T_6 by circulating water in a surface heat exchanger (SHE) after the axial compressor outlet. The saturated air at 6 is then cooled into the saturated air at 7 with a temperature of T_7 in another surface heat exchanger (SHE) before the turbine inlet by the water that is cooled by the return air at 9, which is discharged from freeze rooms. Some water vapor is condensed and the latent heat of the vapor is discharged from 4 to 7. Then the saturated air at 7 is expanded and cooled into the cold air at 8 in the turbine (T). The cold air at 8 is then ducted to the freeze room.

The water injection before the axial compressor aims at decreasing both the temperature of the working fluid and the polytropic exponent in the compression process. Thus, we can save some compression work. This method has been used in a jet engine when a fighter plane speeds up. However, the difference is that what is injected in a jet engine is water, alcohol, etc.

The water vapor in the compressed air can easily be extracted by a surface heat exchanger. With the same temperature, the humidity ratio of the saturated wet air at high pressure P_4 is only about P_3/P_4 of that at the pressure P_3. The method of using compressed air to acquire dry air has been used in some workshop in southern China.

3. Performance simulation

3.1. Wet air

The humidity ratio of wet air, d, is obtained from [41]:

$$d = 621.98 P_{\text{vap}}/(B - P_{\text{vap}}) \tag{1}$$

The enthalpy of wet air, h, is calculated from [41]:

$$h = 1.006t + 0.001d(2501 + 1.805t) \tag{2}$$

The adopted relation for water vapor between saturation pressure and saturation temperature, $P_s = f(t_s)$, is fitted from the datum in ASHRAE Handbook—Fundamentals Volume [41].

3.2. Axial compressor

During the compression process of wet air, the fine water droplets in the air may evaporate. Because the evaporation of water intakes heat, we can regard the ideal compression process of the wet air in the compressor as a polytropic process. Therefore, we can obtain the ideal consumed work of the compressor per kilogram of dry air, w_c, from:

$$w_c = \frac{n}{n-1}(R_{\text{da}} + 0.001dR_{\text{vap}})T_3\left[1 - (p_4/p_3)^{(n-1)/n}\right] \tag{3}$$

In which, n is the poly-tropic exponent for the compression process.

The practical work consumed by the axial compressor is w_c/η_c, in which η_c is the thermal efficiency of the compressor.

3.3. Surface heat exchanger

Pinch technology is used in the analysis of heat exchange in the surface heat exchanger and the temperature difference at pinch point is 6 °C. Pinch technology is a graphical method of identifying technically and economically interesting energy efficiency measures. The minimum cooling and heating demands in the system can thereby be determined, together with the net heat for each temperature level. The concepts and methodology of pinch technology are well explained in the works of Linnhoff et al. [42], Eastop and Croft [43], Linnhoff [44] and Mubarak Ebrahim [45]. The optimum mass flow rate ratio of cool water to dry air could be obtained from hot and cold curves according to pinch technology. Two surface heat exchangers are used because the different cooling water flow rates and temperatures.

3.4. Turbine

The expansion of the saturated air in the turbine cannot be regarded as an adiabatic expansion of an ideal gas. With the decrease of the wet air pressure in the turbine, the temperature of the wet air decreases, and some heat is discharged during the condensation of some water vapor. The heat discharged may cause the increases in both the temperature of the turbine outlet and the work done in the expansion.

For this problem, we can imagine that no phase change exists and that there is some heat added to the wet air during the expansion process when we calculate the work done by the expansion process. According to the above assumption, this problem can be simplified to a problem of the polytropic expansion of an ideal gas. Consequently, we can obtain the ideal consumed work done by the expansion, Wt, through iteration, and then obtain the real work generated by the turbine and the temperature of the turbine outlet.

3.5. Performance

The refrigerating capacity per kilogram of dry air, q_2, can be determined by the enthalpy difference between the inlet of the compressor and the outlet of the turbine by using the following formula:

$$q_2 = h_9 - h_8 \tag{4}$$

The heat rejected per kilogram of dry air, q_1, can be determined by the enthalpy difference between the inlet of the compressor and the outlet of the first surface heat exchanger by using the following formula:

$$q_1 = h_4 - h_6 \tag{5}$$

The regenerated heat per kilogram of dry air, q_R, can be determined by the enthalpy difference between the outlet and inlet of the second surface heat exchanger by using the following formula:

$$q_R = h_6 - h_7 \tag{6}$$

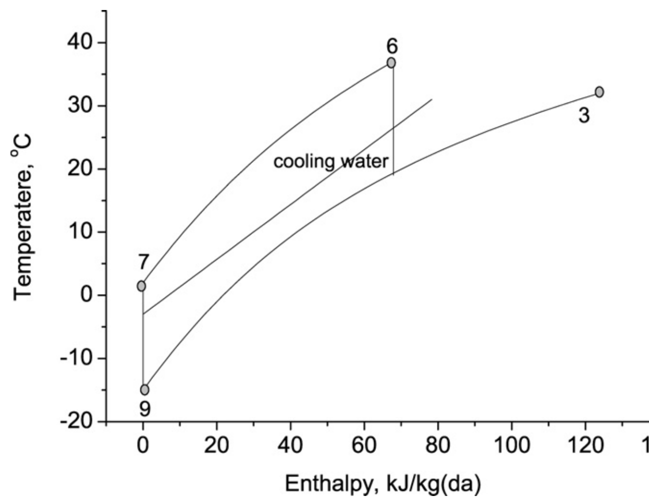

Fig. 3. Pinch chart of the compressed wet air, the circulating water and cooling wet air.

The work consumed by the refrigeration cycle is calculated by:

$$w_m = w_c/\eta_c - w_t.\eta_t \tag{7}$$

The COP of this refrigeration cycle is calculated by the following formulas. (The work consumed by circulating water system is not included in w_m.)

$$COP = q_2/w_m \tag{8}$$

4. Results

There are many factors that may affect the COP of the open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water. These include the pressure ratio of the axial compressor, P_4/P_3, the efficiencies of the axial compressor and turbine, the wet bulb temperature of the atmosphere T_{wet} (T_3) and the wet air temperature of turbine inlet, T_7.

During simulation, the pressure ratio of axial compressor varied from 2.0 to 3.5, wet bulb temperature of the outdoor air from 0 to 30 °C and the temperatures at 6 are 7–12 °C higher than wet bulb temperature of the outdoor air. The temperature of the circulating water from the cooling tower 2, which is cooled by the outdoor air, is 3–4 °C higher than wet bulb temperature of the outdoor air. The temperature of the water from the cooling tower 1, which is cooled by the rejected air from deep freeze room (about −15 °C), is about 0 °C or below and the mass flow rate of the cooling water is 0. 55 kg/s kg (da). The wet air temperature at 7 is 0–5 °C. There is 300 Pa pressure loss before the axial compressor, 300 Pa between the axial compressor and turbine, and 600 Pa after the turbine.

Pinch chart of the compressed wet air, the circulating water and cooling wet air is illustrated in Fig. 3. The circulating water is cooling by the regenerated wet air firstly, and then the compressed wet air is cooled from 6 to 7 by the circulating water. The pinch point temperature differences between the circulating water and cooling wet air and between the compressed wet air and the circulating water are 6 °C. From Fig. 3, we see the temperature at 7 can be easily cooled to 3 °C, and would be lower if

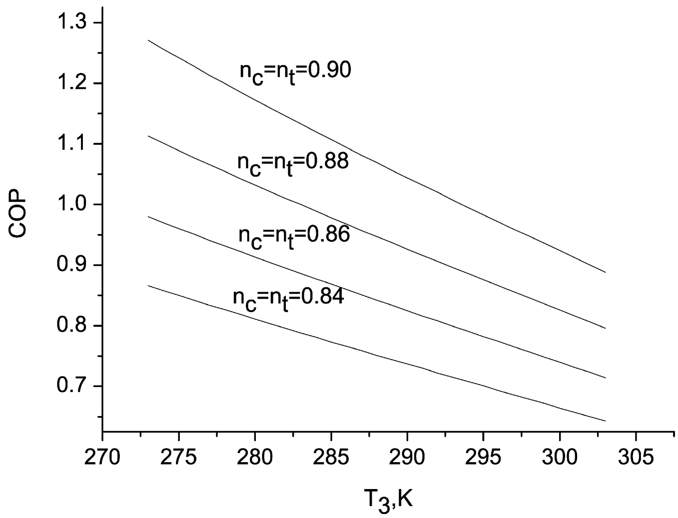

Fig. 4. The sensitivity of COP ($T_7 = 276$ K, $T_9 = 258$ K and $P_4/P_3 = 2.8$) to efficiencies of the axial compressor and turbine.

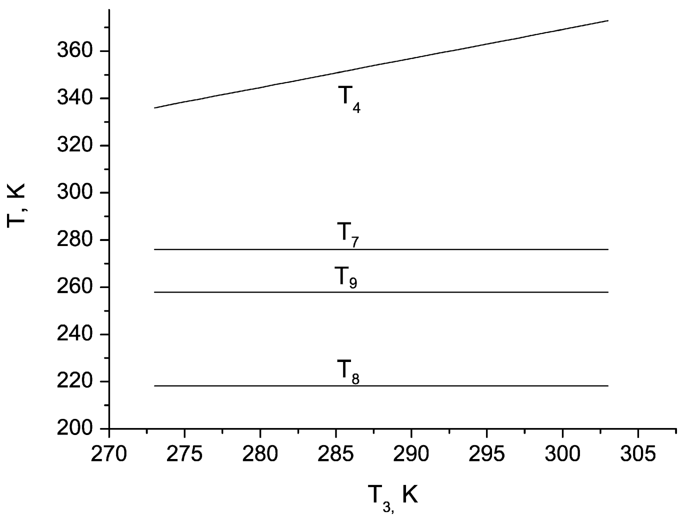

Fig. 5. Simulated temperatures when $T_7 = 276$ K, $T_9 = 258$ K, $P_4/P_3 = 2.8$ and $\eta_c = \eta_t = 0.88$.

using one more cooling tower with less circulating water. That is, the COP would be better.

The sensitivity of COP ($T_7 = 276$ K, $T_9 = 258$ K and $P_4/P_3 = 2.8$) to efficiencies of the axial compressor and turbine is illustrated in Fig. 4. The four lines are the COP lines of an open air-compression refrigeration cycle for deep freeze cooled by circulating water when efficiencies of the axial compressor and turbine are 84, 86, 88 and 90%, respectively. From Fig. 4, efficiencies of the axial compressor and turbine influence on the COP of an open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water greatly. The higher efficiencies of the axial compressor and turbine induce the higher the COP. The COP of this regenerated refrigerating system for deep freeze goes down when outdoor temperature, T_3, rises. The simulated temperatures when $T_7 = 276$ K, $T_9 = 258$ K, $P_4/P_3 = 2.8$ and $\eta_c = \eta_t = 0.88$ is illustrated in Fig. 5. The temperature after axial compressor rise with outdoor wet bulb temperature rising. The temperature of the regenerated

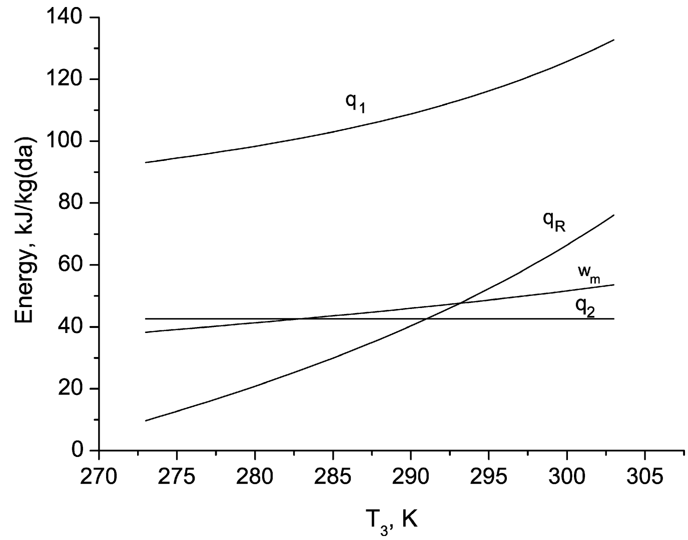

Fig. 6. Simulated heat and work when $T_7 = 276$ K, $T_9 = 258$ K, $P_4/P_3 = 2.8$ and $\eta_c = \eta_t = 0.88$.

wet air at 9 is 258 K. The temperature before turbine, T_7, is selected at 276 K according Pinch chart and the temperature after turbine, T_8, is calculated at 218 K (−55 °C). The simulated heat and work when $T_7 = 276$ K, $T_9 = 258$ K, $P_4/P_3 = 2.8$ and $\eta_c = \eta_t = 0.88$ is illustrated in Fig. 6. The rejected heat, the regenerated heat and the work consumed by system rise with the outdoor wet bulb temperature. Refrigerating capacity, q_2, keep unchanged because the temperature before turbine is selected at $T_7 = 276$ K. According to the axial compressor performance characteristics among rotate speed, mass flow rate and pressure ratio, we can adjust the rotate speed of the axial compressor to control its mass flow rate and pressure ratio and the refrigerating capacity.

5. Conclusions

This study shows the feasibility of an open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water. The simulations show:

- The open reversed Brayton cycle with regeneration for deep freeze cooled by circulating water given in this paper use the mixture of air and vapor as its working fluid and free for user. Its refrigeration depends mainly on both air and vapor, and differs from a conventional air-cycle system. The use of turbo-machinery with high efficiencies makes this possible.
- Humid air is a perfect working fluid for refrigeration for deep freeze.
- Pinch technology is a good solution to calculate and analyze the heat exchanges between hot wet air and cooling water and between water and its cooling wet air.
- The COP of the open reversed Brayton cycle with regeneration for deep freeze varies with the wet bulb temperature of the atmosphere. The higher the wet bulb temperature of the atmosphere, the lower COP of the open reversed Brayton cycle with regeneration for deep freeze.

- The COP of the open reversed Brayton cycle with regeneration for deep freeze rests mainly on η_c and η_t. The temperature of turbine inlet will also affect it heavily. Although the sensitivity of the COP to η_c and η_t, the open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water is still feasible.

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