An open Reversed Brayton cycle with Regeneration Using Moist Air for Air Conditioning Cooled by Circulating Water

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Abstract

This paper presents an open reversed Brayton cycle with regeneration (ORBCR) using moist air for air conditioning cooled by circulating water, and proves its feasibility through performance simulation. 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 air-conditioned room more comfortable and reduce initial cost because of the very low temperature air obtained. The sensitivity analyses of coefficient of performance (COP) of the cycle without regeneration and the equivalent coefficient of performance (COPE) of the cycle with regeneration to the efficiency of compressor (ηc) and the efficiency of compressor (ηt), and the results of both cycles are also given. The simulation results show that the COPE of this system depends mainly on the temperature before turbine (T7), ηc and ηt, and varies with the wet bulb temperature of the outdoor air (Twet). Humid air is a perfect working fluid for central air conditioning and no cost to the user. The ORBCR is more efficient because of the use of the return air to cool water, and then to cool the air before turbine.

Keywords: turbo-machinery; air cycle; air conditioning units; natural working fluid; simulation; refrigeration

1. Introduction

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].

Hou and Li (1992a and 1992b)[4, 5] 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. (2002)[6] 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 (2005)[7-10] and exergy analysis (2007) [11] 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 (2007)[12] , an open air-vapor compression refrigeration system for air conditioning and desalination on ship (2008)[13] and an open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water[14].

Hou and Zhang (2004) [15] 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 temperature and relative humidity of air-conditioned rooms (ACR) are usually 23-25°C and 50%, respectively. It is wasteful if the air is rejected to outdoor directly. The air can be used to chill the circulative water from the cooling tower further. So, the temperature of the turbine inlet will decrease and the equivalent COPE will rise. The aim of this paper is to present an open reversed Brayton cycle with regeneration(ORBCR) using moist air for air conditioning cooled by circulating water, and give its performance and compare with the cycle with regeneration (without using the cool from the air rejected to outdoor)

2. System

The circuit diagram of an ORBCR using moist air for air conditioning cooled by circulating water is shown in Figure 1. The representation of the ORBCR on enthalpy-entropy coordinates is shown in Figure 2.

![Figure 1 Circuit diagram of an ORBCR using moist air for air conditioning cooled by circulating water](image-url)
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 a higher temperature, T4, and a high pressure, P4, is obtained. Then the compressed air at 4 is firstly cooled into the saturated air at 6 with a temperature of T6 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 T7 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 air-conditioned 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 air-conditioned rooms (ACR).

![Figure 2 Representation of an ORBCR using moist air for air conditioning cooled by circulating water on enthalpy-entropy coordinates](image)

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 P4 is only about P3/ P4 of that at the pressure P3. The method of using compressed air to acquire dry air has been used in some workshop in southern China.

The mount of the water extracted from high pressure wet air (4 to 7) can reach 18-30 g/kg(da). And the mount of the latent heat discharged from vapor condensed, about 45-75kJ/kg(da), exceeds the sensible heat from air, 30-50 kJ/kg(da). For this reason, refrigeration load in this air-vapor regenerated refrigeration cycle depends on a combination of the sensible heat of air and the latent heat of vapor. The regenerated cold comes from the rejected air to outdoor. Usually, the cold cannot be used at other system. Use of it could increase the equivalent COPE of this cycle.

3. Performance Simulation

3.1 Wet Air

The humidity ratio of wet air, d, is obtained from [16]:

\[ d = 621.98 \frac{P_{\text{vap}}}{B - P_{\text{vap}}} \]  

(1)
The enthalpy of wet air, $h$, is calculated from [16]:

$$h = 1.006t + 0.001d(2501 + 1.805t)$$

(2)

The adopted relation for water vapor between saturation pressure and saturation temperature, $P_{S} = f(t_{S})$, is fitted from the experimental data in reference[16].

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:

$$w_{c} = \frac{n}{n-1} \left( R_{g} + 0.001d R_{w} \right) T_{a} \left[ 1 - \left( \frac{p_{a}}{p_{i}} \right)^{\frac{n-1}{n}} \right]$$

(3)

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

The practical work consumed by the axial compressor is $w_{c} / \eta_{c}$, in which $\eta_{c}$ is the thermal efficiency of the compressor.

3.3 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 gradually, and some heat is discharged during the condensation of some water vapor gradually. The heat rejected may cause the increases in both the temperature of the turbine outlet and the work done in the expansion.

When the pressure of the saturated air decreased in nozzle of turbine, the temperature of the saturated air declined, some water vapor condensed, the latent heat discharged, and the velocity of wet air increased. For the whole working fluid, it is an adiabatic expansion process. However, for the air only (without liquid water), it is polytropic expansion process with adding heat. The poly-tropic exponent could be determined by the latent heat discharged by condensed water.

So, 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 work done by the expansion, $w_{t}$, through iteration, and then obtain the real work generated by the turbine and the temperature of the turbine outlet. Because the poly-tropic exponent and the temperature of the turbine outlet are determined by the latent heat discharged by condensed water through iteration, the above assumption approach the reality and the results according to the assumption will come close to real situation.

4. Performance

The refrigerating capacity per kilogram of dry air, $q_{2}$, can be determined by the enthalpy difference between the outlet of the air-conditioned room and the outlet of the turbine by using the following formula:

$$q_{2} = h_{8} - h_{8}$$

(4)
However, considering that the cooling load of outdoor air exists at regular system and the heat from the discharged air from air-conditioned room can be used in this cycle. For the convenience to compare with regular system, the equivalent refrigerating capacity per kilogram of dry air, $q_{2E}$, of this cycle is used, which is calculated by the enthalpy difference between the inlet of the compressor and the outlet of the turbine by using the following formula:

$$ q_{2E} = h_3 - h_8 $$

(5)

The work consumed by the refrigeration cycle is calculated by:

$$ W_n = \frac{W_c}{\eta_c} - W_c^{0} $$

(6)

The equivalent COPE and the COP[15] of this refrigeration cycle are calculated by the following formulas, respectively. (The work consumed by circulating water system is not included in $W_n$.)

$$ COP_e = \frac{q_{2E}}{W_n} $$

(7)

$$ COP = \frac{q_3}{W_n} $$

(8)

5. Results

There are many factors that may affect the equivalent COPE of this ORBCR using moist air for air conditioning 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 $T_{wet}$ and the wet air temperature of turbine inlet, $T_7$.

During simulation, the pressure ratio of the axial compressor varied from 1.6 to 2.5, the wet bulb temperature of the outdoor air from 20 to $30^\circ C$ and the temperature at 6 is 7-12 $^\circ C$ higher than the wet bulb temperature of the outdoor air. The temperature of the air-conditioned room is $24^\circ C$ at 50% relative humidity. The temperature of the circulating water from the cooling tower, which is cooled by the outdoor air, is 3-4 $^\circ C$ higher than wet bulb temperature of the outdoor air. The temperature of the water from the cooling tower, which is cooled by the air discharged from air conditioned room, is about 20-22 $^\circ C$ and the mass flow rate of the cooling water is 0.75kg/s.kg(da). The wet air temperature at 7 is 20-30 $^\circ C$.

![Figure 3 The sensitivity analyses of COP (without regeneration) and COPE to efficiencies of the axial compressor and turbine](image)
There is 300 Pa pressure loss before the axial compressor, 300 Pa between the axial compressor and turbine, and 600 Pa after the turbine.

Some encouraging results are illustrated in Figures 3, 4 and 5. The results in Figures 3, 4 and 5 are calculated under the same pressure ratio of the axial compressor.

The sensitivity analyses of COP (without regeneration) and COPE (T7=298 K) to efficiencies of the axial compressor and turbine are illustrated in Figure 3. Lines B, C, D, E and F in Figure 3 are the COP lines of an open air-compression refrigeration cycle for air-conditioning cooled by circulating water when T7=T3+10 and efficiencies of the axial compressor and turbine are 90%, 88%, 85%, 80% and 75%, respectively [15]. Lines G, H, I and J in Figure 3 are the COPE lines of the ORBCR using moist air for air conditioning cooled by circulating water when T7=298 K and efficiencies of the axial compressor and turbine are 90%, 88%, 85% and 80%, respectively. From Fig. 3, one can see that the efficiencies of the axial compressor and the turbine influence the COP (without regeneration) and COPE heavily. Also, one can see that the COPE of the ORBCR using moist air for air conditioning cooled by circulating water is higher than COP of the refrigeration circle without regeneration at the same outdoor condition. That is, The COPE could get much improved when we use the ORBCR. Although the sensitivity analyses of COP
(without regeneration) and COPE to efficiencies of the axial compressor and turbine, these cycles are feasible. Firstly, this new turbo-machinery works near the design point, and efficiencies of axial compressor and turbine are very high at the design point, about 0.89-0.91. Secondly, the intake air is clean and without dust, therefore efficiencies of axial compressor and turbine will not drop greatly while working. Thirdly, there is no very complex combustion chamber and high-temperature turbine in the turbo-machinery. Consequently, it is much easier to accomplish than many people imagined. Lastly, efficiencies of axial compressor and turbine have room for improvement with additional design measures.

The simulations of an ORBCR for air-conditioning cooled by circulating water when $\eta_c=0.90$ and $\eta_t=0.90$ and $T_7=T_3+10$ are illustrated in Figures 4 and 5. Figure 4 gives the relations of the temperature after compressor, $T_4$, the temperature before turbine, $T_7$, and the temperature after turbine, $T_8$ to the temperature before compressor, $T_3$ (Twet). Figure 5 gives the relations of the refrigerating capacity per kilogram of dry air, $q_2$, the discharging heat per kilogram of dry air, $q_1$, and the work consumed by the refrigeration cycle, $w_m$, to the temperature before compressor, $T_3$ (Twet). The relation of COP to the temperature before compressor, $T_3$ (Twet) can be located in Figure 3.

6. Conclusions

This study shows the feasibility of an ORBCR using moist air for air conditioning cooled by circulating water. The calculation results show:

The ORBCR given in this paper is a new concept and its working fluid is the mixture of air and vapor. Its refrigeration depends mainly on both the sensible heat of air and the latent heat of water vapor, and differs from a conventional air-cycle system. The use of turbo-machinery with high efficiencies makes this possible.

Comparing with air-vapor compression refrigeration system for air conditioning cooled by circulating-water[15], this ORBCR is more efficient for air conditioning.

The COP and equivalent COPE of this ORBCR rests mainly on $\eta_c$ and $\eta_t$. The temperature of turbine inlet will also affect it heavily. although the sensitivity of the COP and equivalent COPE to $\eta_c$ and $\eta_t$, the ORBCR using moist air for air conditioning cooled by circulating water is still feasible.

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