SIMULATION OF A NEW TYPE GAS-FIRED AIR-COOLED AIR-CONDITIONER

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ABSTRACT

Energy and environment has been issues concerned all over the world. In China, the application of small size gas-fired air-cooled air-conditioner as an alternative for electric compression air conditioning systems has shown broad prospects due to occurrence of electricity peak demand, lack of water resources and environment-friendly working pairs. Especially, the new air-cooled air-conditioner has many advantages. In order to evaluate its cycle performance, the mathematical model of the new air-conditioner was presented. After algorithm of the model and code program were made, the amount of heat recovery from refrigerant was calculated, and influence of solution distribution ratio into high pressure generator, outdoor air temperature and inlet temperature of cooled water to the evaporator on the cycle performance was evaluated. Results indicated that heat recovery contributed to energy-conservation and COP boost. Solution distribution ratio had an obvious influence on the performance of the system.

KEYWORDS

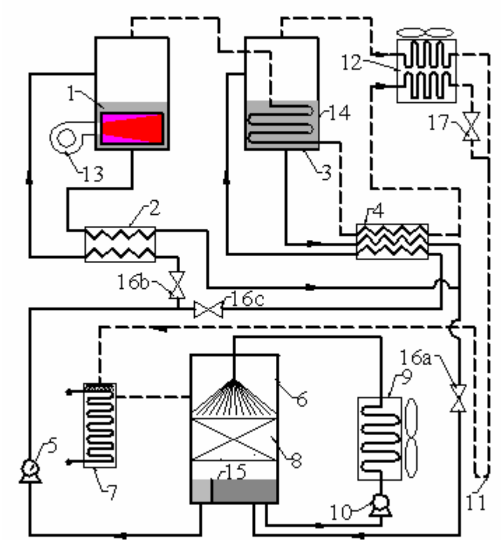
Absorption refrigeration, Heat Transfer, Mass transfer, Mathematical model, Adiabatic absorber

INTRODUCTION

The problems of power supply and potential environmental destruction have become momentums of developing air-cooled gas fired air conditioner using lithium bromide-water solutions as a working fluid. In the last ten years, rapid expansion in the use of electricity-powered air conditioning system in Chinese big cities partially has caused occurrence of electricity demand peaks in the summer and winter seasons. Chinese government has done something to change the adverse situation by encouraging customers into using heat-driven absorption refrigeration devices. It is promising in China. Firstly, it uses natural working fluids and helps to protect environment. Secondly, there are plentiful natural gas resources in China. In addition, the Project of the Western Natural Gas Transported into the East Area has been fulfilled. It is popular that water-cooled absorption heat pumps supply cooling and heat for large buildings, but it is advisable to develop air-cooled absorption refrigeration systems for small building such as villa. The air-cooled absorption refrigeration has great advantages such as low construction, installation and operation cost, outdoor installation, no special machine rooms, increased ease of handing and enhanced level of reliability(Yoon J et al. 1999). On the other hands, the advantages in human health and application in areas lack of water source are remarkable. However, the higher temperature in the condenser and absorber causes boost of pressure, temperature and concentration in the generators, increase of risk of crystallization and acceleration of corrosion (Kurosawa S et al 1989). In order to solve these problems, some theoretical or experimental studies are devoted to invention of new cycle modes, selection of new working fluids and enhancement of heat and mass transfer process in the absorber (Chen GM et al 2002, Lee H et al 2000, Summerer F et al 1996, Velazquez N et al 2002). In view of the facts that the above questions caused by the traditional air-cooled falling film absorption aren't really solved in terms of practical application, in this paper, based on previous works, an air-cooled adiabatic absorption refrigeration air conditioner are developed. To ensure the optimal control and reliability of operation in extremely hot climates, in this study, its operation performance is evaluated.

DEVELOPMENT OF AIR CONDITIONER

Fig.1 shows schematically the gas-fired air-cooled double-effect absorption refrigeration device with an adiabatic absorber. The system has two generators. The refrigerant vapor generated from the high-pressure generator (HG) supplies heat energy for the low-pressure generator (LG) to increase the efficiency of the cycle. It uses outdoor air as a cooling source for an air-cooler and a condenser. The weak solution leaving the absorber is split into two parts that flow in parallel. One part flows into the high-pressure generator (HG) after passing through the high temperature heat exchanger (HX), and the other one flows into the low-pressure generator (LG) after passing through the low temperature heat exchanger (LX). When weak solution flowing into the high-pressure generator (HG) is concentrated into the strong solution, the strong solution passes back through the high temperature heat exchanger (HX) and its heat is transferred to the weak solution stream going to the high-pressure generator (HG). Similarly, when weak solution flowing into the low-pressure generator (LG) is concentrated into the strong solution, the strong solution passes back through the low temperature heat exchanger (LX) and its heat is transferred to the weak solution stream going to the low-pressure generator (LG). Finally, the two parts of strong solutions are mixed and return to the absorber.



1-HG 2-HX 3-LG 4-LX
5-solution pump 6-adiabatic absorber
7-evaporator 8-packed bed 9-air-cooler
10-circulation pump 11- throttle tube
12-condenser 13-gas-fired heater
14-solution heat exchanger in LG
16-regulation valve 17-nozzle

Fig.1: Schematic diagram of an air-cooled air-conditioner

The system has two new features of waste heat recovery of condensed water from generator and an adiabatic absorber with an air cooler, and contributes to solving the problems occurring in the conventional air-cooled absorption refrigeration system Owing to different mechanics of heat and mass transfer, it is difficult to enhance heat and mass transfer in air-cooled falling film absorber simultaneously. However, Heat and mass transfer process is separated in adiabatic absorber with an air-cooler. In this way, heat is rejected effectively in an air-cooler while the mass transfer occurs subsequently in an adiabatic absorber. There is a packed bed to enhance the mass transfer in adiabatic absorber. The pack bed augments interface between vapor and spray solution, and provides enough space and time for mass transfer process.

SIMULATION MODEL

The simulation models involves mathematical model of governing equations for each component making up the cycle system. The governing equations are developed based on conversation of mass and conversation of energy. For the purpose of simplification of the models and steady state simulation, the following assumptions are made:

1. After the superheated vapor from the high pressure generator supplies heat for low pressure generator, the coolant leaving the low pressure generator is in the saturated water state.
2. Thermophysical properties of subcooled water are treated as that of saturated water at the same temperature conditions. Subcooled water temperature from the low temperature heat exchanger is 15 ℃ higher than weak solution temperature leaving the absorber.
3. After the superheated vapor from the low-pressure generator is condensed in the condenser, the coolant leaving the condenser is in the saturated water state.
4. The pressure drop in the absorber is considered.
5. Flow in the heat exchangers is regarded as the counterflow.
6. Heat loss to the environment is negligible, and pressure drop in the system except the absorber is neglected.
7. Thermophysical properties of air are constant.

According to conversion of mass, total mass balance equation and solution mass balance equation for each component is shown as follows respectively

$$\sum G_i - \sum G_o = 0 \qquad (1)$$

$$\sum G_i X_i - \sum G_o X_o = 0 \qquad (2)$$

where G_i, G_o is mass flow rate of solution (or refrigerant) flowing into and leaving each component respectively, kg/s; X_i, X_o is concentration of solution flowing into and leaving each component respectively, %(wt).

According to conversion of energy, energy balance equation for each component is written as follows

$$Q + \sum G_i h_i - \sum G_o h_o = 0 \qquad (3)$$

where Q is heat capacity, W, and it is equal to zero, for the other component except high pressure generator; h_i, h_o is enthalpy of solution (or refrigerant) flowing into and leaving each component respectively, J/kg.

Coefficient of preference (*COP*) is defined as ratio of the cooling capacity output from the system to the sum of heat input from the gas-fired heater and power input from pump, fan and other electric component, which is shown as follows:

$$COP = Q_{EV} / (Q_{HG} + P) \qquad (4)$$

To solve the set of governing equations, several specific model details incorporated into the model for each component must be addressed. The model details involve the empirical relations and the thermophysical properties of lithium bromide-water solutions, water and steam. The nucleate boiling heat transfer coefficient of lithium bromide-water solutions is derived from Charters' correlations (Charters WWS et al 1982). The convection heat transfer coefficient of water is evaluated according to the equation given by Dittius and Bolter (Holman JP 2002). The evaporation heat transfer coefficient outside horizontal tubes is calculated using the correlation by Lorenz et al (1979). The convective condensation heat transfer in tubes is obtained from the correlation employed by Sarma et al (2002). Thermophysical property values of lithium bromide-water solution, water and steam required by the model in solving the governing equations were evaluated from the literature (McNeely LA 1979, Lee RJ et al 1982, Hyland RW et al 1983, Malhotra A et al 2001). Based on the thermodynamic property and transport property equations in the literatures, a subroutine is made. Thermophysical properties values are calculated when the subroutine is called from the main program.

In addition, the model involves heat transfer area of each component. According to optimization design results, Solution distribution ratio (SDR) into HG is 0.483, which is defined as the ratio of the solution mass flow rate into the high pressure generator to total solution mass flow rate leaving the absorber, and power input from pump, and fan and other component is equal to 1.486 kW. Table 1 shows the optimization design results of heat transfer surface area for LG, HX, LX, air-cooler, condenser and evaporator.

Tab1: Results of heat transfer area obtained from optimization design for the normal system

Component	**Area (m^2)**
Coolant side of LG	1.189
Strong solution side of HX	8.842
Strong solution side of LX	7.939
Coolant side of LX	0.249
Air side of air cooler	118.202
Air side of condenser	48.099
Water-cooled side of evaporator	1.949

RESULTS AND DISCUSSION

The influences of SDR into HG, outdoor air temperature and cooled water inlet temperature to evaporator on cycle performances are investigated.

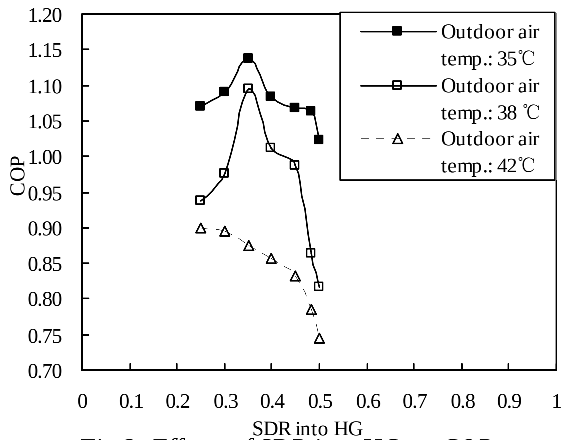


Fig.2: Effects of SDR into HG on COP

Fig.2 shows the effects of SDR into HG on COP at outdoor air temperature of 35℃, 38℃ and 42℃. It is seen that the maximum of COP appears as SDR into HG varies. So at different outdoor air temperature, optimal SDR into HG makes the cycle the most efficient.

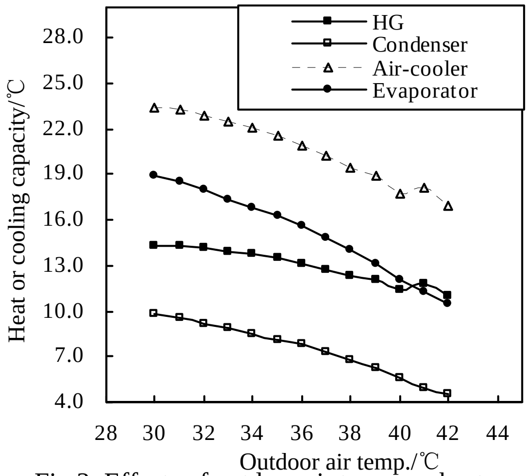


Fig.3: Effects of outdoor air temp. on heat or cooling capacity

Tab.2 Solution crystallization temp in HX and outdoor air temp with crystallization

SDR	**crystallization temp (℃)**	**Air temp. in crystallization(℃)**
0.25	55.2	39.0
0.30	51.8	36.0
0.35	47.7	32.0
0.40	43.2	28.0

Tab.2 illustrates both solution crystallization temperature in HX and outdoor air temperature with crystallization decrease with SDR into HG rise. It results from the increase of solution leaving the HG as SDR decreases.

Fig.3 shows the influences of outdoor air temperature on heat capacity of three components (HG, condenser and air-cooler) and cooing capacity of evaporator. It is seen that both heat capacity and cooling capacity decrease in steps with air temperature increase. However, the slope of heat input into HG is the smallest of all.

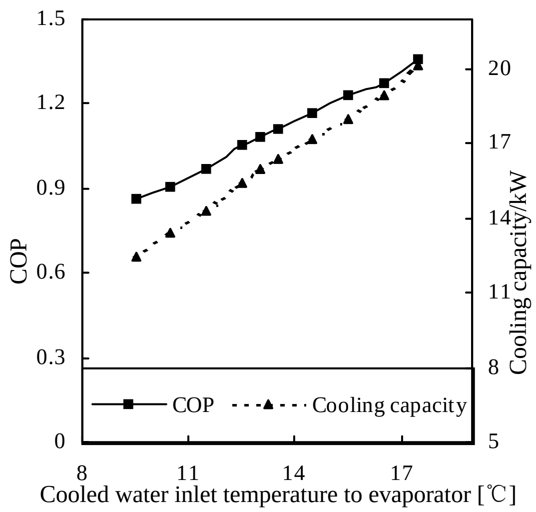


Fig.5: Effects of cooled water inlet temperature to evaporator on COP and cooling capacity

Fig.4 shows the trends of variation of COP with outdoor air temperature increase when SDR into HG is 0.35. It is seen that COP decreases as air temperature rises. Especially at extremely hot climates, when air temperature reaches 43℃, COP reduces significantly. At the same air temperature, COP with heat recovery is higher than that without heat recovery. On the average, COP with heat recovery increases 0.058.

Fig.5 shows the effects of cooled water inlet temperature to evaporator on COP and cooling capacity at SDR of 0.483. It is seen that COP and cooling capacity increases linearly with cooled water inlet temperature rise. It indicates that, at the extremely hot climates, cooled water inlet temperature to evaporator is properly boosted so as to improve COP and cooling capacity.

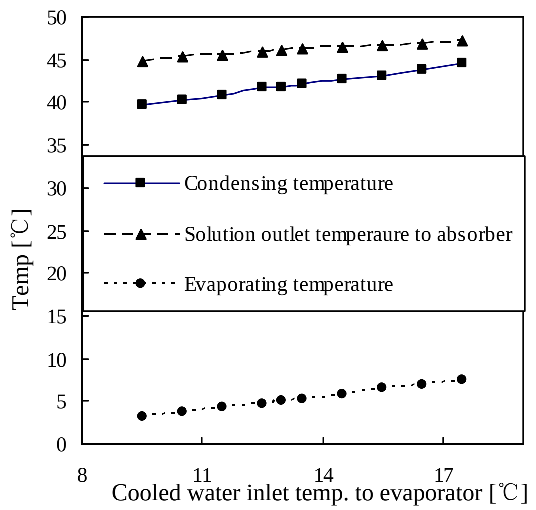


Fig.6: Effects of cooled water inlet temp to evaporator on solution outlet temp to absorber, condensing temp and evaporating temp

Fig.6 shows the effects of cooled water inlet temperature to evaporator on solution outlet temperature to absorber, condensing temperature and evaporating temperature at SDR of 0.483. As cooled water inlet temperature to evaporator rises, condensing temperature, evaporating temperature and solution outlet temperature to absorber increase in steps. However, the rate of increase of solution outlet temperature to absorber is not obvious. When cooled water inlet temperature to evaporator varies between 9.5 ℃ and 17.5℃, solution outlet temperature to absorber ranges from 44.8 ℃ to 47.2 ℃, but condensing temperature ranges from 39.7 ℃ to 44.5℃ and evaporating temperature from 3.3 ℃ to 7.4 ℃.

Fig.7 shows the effects of cooled water inlet temperature to evaporator on circulation solution flowrate, recirculation solution flowrate, subcooling of spray solution, which is defined as temperature difference between solution from absorber and spray solution from air cooler, at SDR of 0.483. It is seen that, recirculation solution flowrate linearly rises as cooled water inlet temperature to evaporator, but circulation solution flowrate decreases in steps. Recirculation solution flowrate has a strong effect on power input from circulation pump, but circulation solution flowrate has a small effect on power input from solution pump because circulation solution flowrate is too small compared with recirculation solution flowrate. The figure illustrates that subcooling temperature has small variations when cooled water inlet temperature to evaporator varies. Although Subcooling is almost kept constant, absorption effects of solution are still enhanced owing to increase of evaporating pressure.

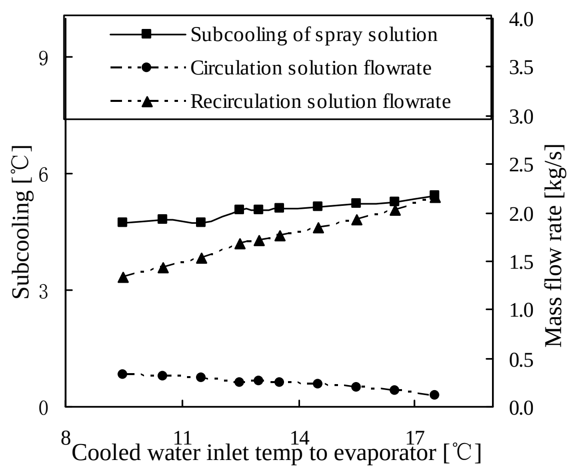


Fig.7 Effects of cooled water inlet temp to evaporator on subcooling of spray solution, circulation flowrate and recirculation flowrate

CONCLUSIONS

Small size gas-fired air-cooled adiabatic absorption air conditioner is developed. Heat is recovered from coolant-saturated water in the low temperature heat exchanger. It contributes to energy-conservation and COP boost. Adiabatic absorber with air-cooler separates mass transfer process with heat transfer process. It is feasible to enhance mass transfer in adiabatic absorber and heat transfer in air cooler effectively. Based on performance simulation and characteristic analysis, the optimal control and mode of reliable operation in extremely hot climates are obtained. It indicated that the device resolved some problems occurring in air-cooled falling film absorption refrigeration systems and helped save energy. At extremely hot climates, boost of cooled water inlet temperature to evaporator contributes to increase of cooling capacity and COP, but it causes increase of condensing temperature. Solution outlet temperature to absorber is insensitive to cooled water inlet temperature to evaporator.

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