



Performance evaluation of a small capacity compression–absorption refrigeration system

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ABSTRACT

Simulation of a 100 kW ammonia–water compression–absorption system has been carried out for water chilling application by incorporating detailed heat and mass transfer calculations in the absorber and desorber of the system. Warner's technique has been used for the convergence of iterations. The effect of relative solution heat exchanger area on the COP, cooling capacity, absorber heat load and effectiveness of solution heat exchanger has been studied. With increase in the relative solution heat exchanger area from 17 to 50%, COP increases initially, becomes maximum at 39% and then decreases. However, capacity of the system decreases considerably as a result of increase in relative solution heat exchanger area. Due to pressure drop in the solution heat exchanger on weak solution side, pumping power increased by 30% but COP decreased by only 0.92%. The performance of this system has been compared with that of a conventional vapor compression type chiller.

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1. Introduction

The compression–absorption (CA) systems have been looked upon to be potential alternatives to conventional vapor compression and vapor absorption systems due to some of their unique features such as better or comparable performance, better capacity control and environment friendliness owing to use of ozone friendly mixtures like ammonia–water and lithium bromide–water etc.

The overall performance of the system depends upon the performance of the absorber, desorber and solution heat exchangers which are main components of the system. In the previous studies carried out by Pourezza–Djourshari and Radermacher [1], Ahlby et al. [2], Groll and Radermacher [3], Groll [4], Sveine et al. [5] and Itard [6], the performance of the compression–absorption system has been calculated by modeling the heat exchangers as black box. Also, various pressure drops in the system have not been accounted for in the analyses. In other studies carried out by Hulthen and Berntsson [7,8], performance results of a compression–absorption heat pump have been presented by computing heat and mass transfer in the absorber and

desorber. Further, most the studies published in the literature are confined to heat pump applications. Therefore, the present work has been devoted to study refrigeration application of compression–absorption system. Simulation of a 400 kW CA refrigeration system has been carried out in the previous work [9].

A 100 kW capacity ammonia–water compression–absorption refrigeration system has been designed for water chilling application and simulation of this system has been carried out by performing actual heat and mass transfer calculations in the absorber, desorber and solution heat exchanger. The effect of relative area distribution, solution heat exchanger pressure drop, tube side and shell side pressure drops in the absorber and desorber on the performance of the system have been evaluated. The performance of this system has also been compared with that of a conventional R 22 vapor compression type chiller.

2. Description of the system

The compression–absorption system has been shown in Fig. 1. An ideal mixer before the absorber allows the hot refrigerant (ammonia) vapors coming from compressor and the weak solution of ammonia coming from desorber to come into thermal equilibrium with each other. After mixer, weak liquid solution and ammonia vapors enter separately into the absorber (5a) where liquid is distributed on the inner surface of the tube as a falling film and refrigerant (ammonia) vapors are made to

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Nomenclature		x_e	equilibrium concentration of ammonia in solution, (kg kg ⁻¹)
A	area, (m ²)	y	concentration of ammonia in vapor, (kg kg ⁻¹)
c_{pw}	specific heat of water, (kJ kg ⁻¹ K ⁻¹)	<i>Greek Alphabets</i>	
CA	compression–absorption	η_i	isentropic efficiency, (–)
COP	coefficient of performance, –	ϵ	effectiveness of SHX, (–)
D	diameter, (m)	μ	dynamic viscosity, (kg m ⁻¹ s ⁻¹)
D_{shell}	shell diameter, (m)	ν	kinematic viscosity, (m ² s ⁻¹)
g	gravitational acceleration, (m s ⁻²)	Γ	linear wetting density, (kg s ⁻¹ m ⁻¹)
h	film heat transfer coefficient, (Wm ⁻² K ⁻¹)	<i>Subscripts</i>	
k	thermal conductivity, (Wm ⁻¹ K ⁻¹)	abs	absorber
m	mass flow rate, (kg s ⁻¹)	c	compressor
n	number of variables, (–)	des	desorber
Nu	Nusselt number, (–)	e	equilibrium condition
P	pressure, (Pa)	i	inner
P_t	tube pitch, (m)	o	outer
Pr	Prandtl number, (–)	p	pump
Q_{abs}	heat load of absorber, (kW)	s, sol	solution
Q_o	cooling capacity of desorber, (kW)	ss	strong solution
Re	Reynolds number, (–)	v	vapor
SHX	solution heat exchanger	wa	water, absorber
T	temperature, (°C)	wd	water, desorber
U	overall heat transfer coefficient, (Wm ⁻² K ⁻¹)	ws	weak solution
UA	overall heat transfer coefficient area product, (W K ⁻¹)		
W	power input, (kW)		
x	concentration of ammonia in solution, (kg kg ⁻¹)		

flow in the core in the shell and tube type absorber. The refrigerant vapors are absorbed in the weak solution and the heat of condensation and absorption is rejected to the cooling water (sink) therefore temperature of cooling water increases. The cooling water flows in the shell in the countercurrent direction with solution i.e. it enters from the bottom of the absorber and comes out from the top. As ammonia vapors are absorbed in the weak solution, rich solution is obtained at 6 i.e. at the outlet of the absorber. The schematic diagram of the

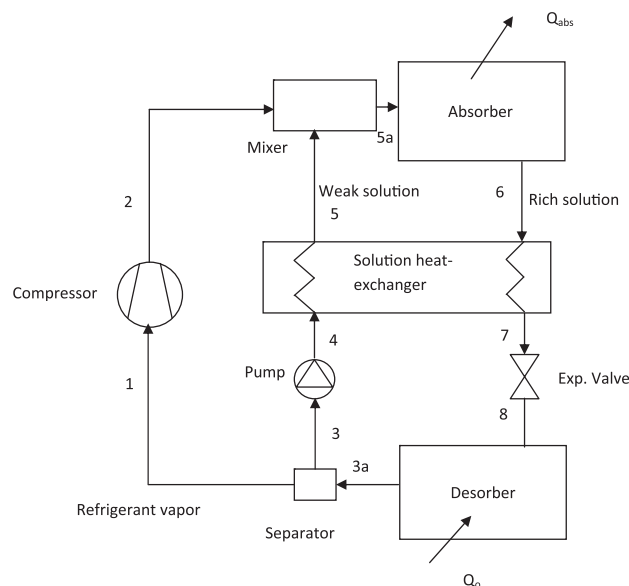


Fig. 1. Compression–absorption refrigeration system.

absorber has been shown in Fig. A.2 in the Appendix. The rich solution is passed through a solution heat exchanger where it further gets cooled by exchanging heat with subcooled weak solution. The rich solution is then throttled at 7 where its temperature and pressure decreases. This low temperature solution enters the desorber where it descends as falling film inside the tubes. The water (source) flows in the shell of desorber in the countercurrent direction with the solution. The thermal energy (heat) of the source (medium to be cooled) is used in the desorber to cause desorption of low temperature strong solution entering at state 8. Thus refrigerant vapors are generated off the solution and weak solution is obtained. In this process, temperature of water (source) decreases and chilled water is obtained. The design of desorber is same as that of absorber. The function of the separator is to facilitate the separation of weak solution and refrigerant vapors after desorption. The refrigerant vapors so generated are compressed by a compressor and discharged at state 2 and the weak solution is pumped (process 3–4) back to absorber at 5 while passing through a solution heat exchanger counter-currently with strong solution coming from absorber at 6.

3. Modeling of the components

Modeling and simulation of various components of CA system has been discussed in this section. The absorber has been modeled as vertical co-current shell and tube heat exchanger, with the solution as falling film on inside tube surface, vapor in the core co-current with solution and water on baffled shell side in countercurrent direction with solution. Jain [10] has carried out the simulation of shell and tube type of absorber for LiBr–H₂O based liquid desiccant cooling system by solving the mass, material and energy balance in the form of first order differential equations.

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