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EXPERIMENTAL INVESTIGATION OF A TWO STAGE VAPOR COMPRESSION HEAT PUMP WITH SOLUTION CIRCUITS

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ABSTRACT

A new scheme to significantly improve the performance of the two stage vapor compression cycle by eliminating the rectifier was first investigated with the help of computer simulation, and then incorporated in the experimental setup. Simulation results show that the cycle with a bleed line (modified cycle without the rectifier) has 20 to 30% higher cooling COP as compared to the cycle with the rectifier. It is important to note that this improvement in COP is accompanied by 10 to 15% increase in cooling load. Initial experimental results along with operating experience and description of the data acquisition program are presented here. Results show that heat can be pumped from an average temperature of 0°C to an average temperature of 100°C with a pressure ratio as low as 7.1. Cooling COPs up to 1.0 were obtained for cooling loads of about 4.17 kW.

KEYWORDS

Compression Heat Pump, Solution Circuits, Two-stage System, Hybrid Heat Pump

INTRODUCTION

The pressure ratios encountered in the two stage VCHSC can be 40 to 65% lower compared to that in a single fluid heat pump operating between the same source and sink temperatures. A two stage VCHSC can be looked upon as two single stage VCHSCs coupled together which share one compressor. This is very similar to conventional cascade vapor compression cycles which, however, require two different compressors. The option of using one large compressor rather than two small ones reduces the cost and increases the reliability of the system.

The first stage of the two stage VCHSC, as shown in Figure 1, pumps heat from the lower temperature in desorber A to an intermediate temperature in absorber B. Heat is exchanged internally between the absorber B and the desorber C. The second stage pumps heat from the intermediate temperature in desorber C to the higher temperature in absorber D. Thus, heat is pumped from the lower temperature in desorber A to the higher temperature in absorber D. As heat is pumped through a smaller temperature difference in each stage the pressure ratio in the two stage VCHSC can be 40 to 65% lower compared to that in a single fluid Rankine cycle heat pump.

The reduced pressure ratio improves the volumetric and overall efficiency of

the compressor, which in turn improves the COP and offers the possibility of using a centrifugal compressor, where otherwise a reciprocating compressor would be required. The highest temperature and pressure in the compressor of a two stage VCHSC can be significantly lower than that in a single fluid heat pump operating between the same source and sink temperatures. However, as required by thermodynamics, the increase in COP is accompanied by reduced capacity. The capacity of the two stage system is about 45% less than the single stage system, this is because the compressor supplies vapor to both the stages.

Using only one compressor to compress the vapor for both the stages leads to the need for rectification. This is because the vapor concentrations at the outlet of the low and the high temperature desorbers are not the same. The vapor at the outlet of the high temperature desorber has more water vapor than that at the outlet of the low temperature desorber. If this excess water vapor is not removed, it ends up in the low temperature solution loop. This reduces the difference between the average concentrations in the high and the low temperature solution loops, which adversely affects the performance of the heat pump by increasing the pressure ratio required to pump heat through the required temperature lift.

The VCHSC can be operated over a wide range of temperatures, between -10 and 160°C , using ammonia-water as the working fluid and with pressures not exceeding 20 bar (Stokar and Trepp 1986). At higher pressures, temperatures up to 200°C can be achieved.

PERFORMANCE ENHANCEMENT WITH A BLEED LINE

A new scheme to significantly improve the performance of the two stage cycle by eliminating the rectifier was first investigated with the help of computer simulation and then incorporated in the experimental setup. The need for a rectifier is eliminated by incorporating a bleed line. Water balance in the high and the low temperature circuits is now maintained by bleeding 2 to 5% of the weak solution flow from the low temperature solution circuit to the high temperature solution circuit.

The two stage VCHSC was simulated using a computer program comparing the performance of the cycle with rectifier with a cycle that employs a bleed line instead. The simulation results show that the cycle with the bleed line has 20 to 30% higher cooling COP as compared to the corresponding cycle with a rectifier.

DESCRIPTION OF THE TEST SETUP

The test setup is designed in such a way that the performance of both single and two stage VCHSC can be studied with no changes in the piping connections. After completion of the single stage tests, some valves were turned to obtain the two stage heat pump configuration.

Two stage VCHSC

Figure 2 shows the layout of the experimental setup without the rectifier and with a desuperheater. Five heat exchangers are required for the two stage

system. They are, the low temperature desorber (1.895 m^2), the heat exchanger which constitutes the low temperature absorber and the high temperature desorber (2.57 m^2), the high temperature absorber (2.57 m^2) and the low and the high temperature solution heat exchangers (4.367 m^2). All the heat exchangers are shell and tube type heat exchanges.

Strong ammonia solution enters at the lower end of the vertical low temperature desorber on the tube side, while 40% ethylene glycol-water solution enters from the top on the shell side. The ammonia vapor and the weak solution coming out of the desorber is piped into the separator. Ammonia vapor is drawn out of the top of the separator tank while the weak solution is pumped to the solution heat exchanger (SHX) using a reciprocating pump (RP). The solution flow rate can be changed by adjusting the pump speed.

The weak solution leaving the separator is saturated. The solution is subcooled before it enters the pump in order to prevent cavitation in the suction tube. The required degree of subcooling depends on the rate at which the desorber pressure is falling during start-up. When the system stabilizes, the desorber pressure is constant and the degree of subcooling can be reduced to less than 0.5°C . In a full scale application the need for subcooling can be avoided by providing a high liquid head at the pump inlet.

The ammonia vapor is compressed in a two-stage reciprocating, oil-free compressor. A two stage compressor is selected only because a single-stage compressor is not available in a laboratory scale capacity range. The compressed vapor and the weak solution enter at the bottom of the low temperature absorber on the shell side. Before entering the absorber, the weak solution exchanges heat with the strong solution in the solution heat exchanger.

In the absorber/desorber heat exchanger, absorption occurs on the shell side and desorption on the tube side. Weak solution from the low temperature stage and the compressed vapor enter the shell side at the lower end and the strong solution emerges at the top. In order to maintain counter current flow in this heat exchanger the strong solution enters the high temperature desorber at the top and the weak solution along with the vapor leaves from the bottom.

The high temperature stage is set up in the same way as the low temperature stage. The vapor from the low and the high temperature stages are compressed together in the compressor. The 40% ethylene glycol-water mixture which picks up the heat from the high temperature absorber flows through the tubes counter-current to the ammonia solution.

EXPERIMENTAL RESULTS

Operating Conditions

The heat pump is operated with an average glycol-water temperature in the desorber of 0°C , and that in the absorber of 100°C . The weak solution mass flow rate varied from 10.2 to 22.7 g/s in the low temperature stage and from 25.7 to 32.7 g/s in the high temperature stage. Data are collected at different weak solution concentrations in the low temperature stage ranging from 43.0 to 55.1 wt% ammonia. The mass flow rate of the glycol-water mixture in the desorber and the absorber is fixed at about 125 g/s.

Performance Calculations

The following procedure was used to calculate the parameters needed for studying the performance of the two stage VCHSC.

1. The instantaneous weak solution concentrations are calculated using the Ziegler (1986) subroutines, based on the measured values of temperatures and pressures of the ammonia solution streams exiting the desorbers. The instantaneous strong solution concentrations are calculated based on the assumption that the throttling process is adiabatic and then using the enthalpy, the mass and the concentration balance equations. The assumption that the solution and vapor exiting the desorbers are in equilibrium and that of adiabatic throttling were verified by taking samples of solution and titrating them with 1N HCl solution. The difference in the calculated and the actual measured value was less than 0.1 wt% ammonia, which is well within the errors involved in the titration method.
2. Temperatures, pressures, flow rates and concentrations are averaged over the period of an experimental run. These average values are used for all further calculations. The fluctuations in the instantaneous values for temperatures were within ± 0.5 °C. The fluctuations in solution flow rates were within $\pm 3\%$ of the average and that for glycol-water flow rates were within $\pm 0.5\%$ of the average.
3. The mass flow rate of the strong solution at the outlet of the solution heat exchanger (SHX) is calculated using the volume flow rate (which is measured using the turbine meter), and the solution density.
4. The mass flow rates of weak solution and ammonia vapor are calculated using the weak and the strong solution concentrations, the calculated vapor concentration (at the outlet of the desorber) and the mass and the concentration balance equations.
5. The desorber and the absorber heat capacities are calculated using the data for the temperature differences and the volume flow rates for the 40% ethylene glycol-water mixtures. The density and the specific heat for the mixture is evaluated at the average mixture temperature in the desorber and the absorber, based on the values reported in a Dow Chemicals product catalog (1985).
6. The desorber and the absorber capacities are also calculated using the ammonia solution concentrations, the flow rates, the temperatures at the inlet and the outlet of the desorber and the absorber, and the vapor temperatures at the inlet and the outlet of the compressor. These capacities are within ± 5 percent of the capacities calculated using the glycol-water mixture. The cooling capacities reported here are based on the calculations made using the enthalpy balance on the glycol-water mixture.
7. The cooling COP of the heat pump is based on the measured values for the power consumption of the compressor (after backing out the losses due to the electrical efficiency of the motor (85%) and the transmission efficiency of the belts (95%)).

Results

Initial test results of the two stage heat pump are very encouraging. Tables 1a and 1b shows that the cooling COP in the range of 0.69 to 1.00 were obtained for a temperature lift of 100°C. The cooling capacities were in the range of 2.02 to 4.17 kW. The pressure ratios encountered at various

operating conditions range between 7.1 and 11.0. High temperature lifts are achieved at pressure ratios which are less than half as large as those for conventional systems (pressure ratio for a conventional vapor compression heat pump using pure ammonia as the working fluid would be about 20 and the cooling COP would be about 0.6 for the same temperature lift). Higher performance may be achieved after the flow rates of the two solution pumps are optimized.

The effect of variation of the pump flow rates in the low and the high temperature stages can be understood by comparing the three data sets. Case number 3 has the highest cooling COP among the three cases listed here. This is primarily due to the low pressure ratio for this case which result from good match in the temperature glides in the different heat exchangers (the low temperature desorber Table 1b columns 2 and 3, the intermediate temperature absorber/desorber heat exchanger columns 4 and 5, the high temperature absorber columns 6 and 7).

Increasing the weak solution concentration in the low temperature loop from 43.0 to 55.1 wt% ammonia resulted in more than two fold increase in the cooling capacity. The cooling capacity increased from 2.02 to 4.17 kW (Table 1a). Increased weak solution concentration lead to increase in the desorber pressure from 0.73 to 1.49 bar (Table 1b). This increased the mass flow rate of the vapor compressed by the constant displacement compressor, which in turn increased the cooling capacity of the heat pump. Thus, the cooling capacity of the two stage VCHSC can be continuously varied by changing the weak solution concentration.

One other group (Ziegler and Hammer 1990) has reported experimental results for a two stage VCHSC (serving three temperature levels). They have reported cooling COPs of about 0.6 while pumping heat from -20°C to 80°C (for cases where no heat was delivered at the intermediate temperature level).

OPERATING EXPERIENCE

During initial operation of the two stage vapor compression heat pump two major problems were encountered.

1. Difficulty in maintaining almost same pressures in the high and the low temperature absorbers using manually operated throttle valves.
2. Setting the right bleed flow rate to compensate the excess water deposited in the low temperature solution circuit.

The following measures were taken to overcome these problems. The manually operated throttle valves were replaced with back-pressure regulating valves. These valves offer the advantage of regulating the pressure in the absorbers very close to the set value, irrespective of the solution flow rates. This was a big step ahead, because this eliminated the need for continuously adjusting the settings of the manually operated throttle valves, especially during start up.

The two problems listed above will not be faced in larger industrial size heat pumps because larger heat pumps will use centrifugal pumps instead of reciprocating, thereby eliminating the piston ring problem. Also, turbine meters are being used in the experimental set up to acquire test data and would not be present in the final design.

CONCLUSIONS

The performance potential of a two stage VCHSC is being experimentally investigated. The experiment was designed and built. In the course of the work, a new scheme to eliminate the need for the rectification in the two stage VCHSC was investigated. In the new scheme the water balance in the high and low temperature solution circuits is attained by using a bleed line instead of a rectifier. Computer simulation results show that the new scheme has the potential to increase the cooling COP by 20 to 30% and the cooling load by 10 to 15%. These results encouraged us to adopt our experimental setup to test the cycle with the bleed line. The simulation results will be sent for publication in the near future.

Initial experimental results for the two stage VCHSC demonstrate that

- * High temperature lifts (100°C) can be achieved at pressure ratios (as low as 7.1) which are less than half as large as for conventional systems (about 20).
- * The heat pump can be operated under stable conditions using the control mechanisms incorporated in the experimental setup and strategies developed.
- * Cooling COP as high as 1.00 can be achieved while pumping heat through a lift of 100°C .

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**Etude Expérimentale d'une pompe à chaleur à deux étages
de compression de vapeur avec circuits de solution**
Milind V. Rane and Reinhard Radermacher

Un nouvel arrangement augmentant la performance du cycle de compression de vapeur à deux étages par élimination du redresseur à été tout d'abord étudié à l'aide d'une simulation par ordinateur puis incorporé au montage expérimental. Les résultats de la simulation montrent que le cycle avec ligne d'écoulement (cycle modifié sans redresseur) a un coefficient de performance (COP) du refroidissement 20 à 30% plus fort que celui du cycle avec redresseur. Il est important de noter que cette amélioration en COP est accompagnée d'une augmentation de 10 à 15% en refroidissement utile. Les résultats expérimentaux ici présentés montrent que de la chaleur peut être pompée à partir d'une température moyenne de 0° jusqu'à une température moyenne de 100° avec des rapports de pression aussi faibles que 7.1. Des COPs de refroidissement allant jusqu'à 1.0 ont été obtenus avec un refroidissement utile d'environ 4.17 kW.

Figure 2: Layout of test setup in the two stage VCHSC mode:
Modified cycle with a bleed line.

Figure 1: Two-stage vapor compression heat pump with ammonia water solution cycle: Modified cycle with a bleed line and a desuperheater.

Data Set #) Name	Solution Flow Rate (g/s)				Solution (wt%)				Conc.		SIIX Eff. (%)		PR	Cooling Load(kW)	COPc
	swl	swh	ssl	ssh	dol	aol	doh	aoh	xl	xh					
1) 111390.2	22.7	25.7	26.2	28.6	52.6	58.9	21.2	28.6	86.9	82.2	7.8	3.51	0.95		
2) 121390.1	21.3	32.7	23.4	34.7	43.0	48.2	20.1	24.5	82.7	83.7	11.0	2.02	0.69		
3) 010391.2	10.2	30.7	13.9	34.3	55.1	67.0	24.1	31.7	92.9	81.8	7.1	4.17	1.00		

Table 1a: Summary of results

Data Set #) Name	Absorber and Desorber Temperature Glides (°C)							MSB (g/s)	T ₅ (°C)	Eff. (%)		P 1 (bar)	Cooling Load(kW)	COPc
	wdl	sdl	swl	sal	sdh	sah	wah			bleed	vco			
1) 111390.2	7.6	3.1	10.5	8.6	8.8	7.0	.305	101.7	76.4	75.4	1.21	3.51	0.95	
2) 121390.1	4.3	1.5	10.7	3.2	1.4	3.1	.153	98.8	80.9	69.6	0.73	2.02		
3) 010391.2	8.7	7.2	13.5	7.7	8.4	7.7	.273	102.9	70.2	70.2	1.49	4.17	1.00	

Table 1b: Summary of results

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