DEVELOPMENT OF A HIGH PERFORMANCE AIR SOURCE HEAT PUMP FOR THE US MARKET

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Abstract: Heat pumps present a significant advantage over conventional residential heating technologies due to higher energy efficiencies and less dependence on imported oil. The US development of heat pumps dates back to the 1930's with pilot units being commercially available in the 1950's. Reliable and cost competitive units were available in the US market by the 1960's. The 1973 oil embargo led to increased interest in heat pumps prompting significant research to improve performance, particularly for cold climate locations. Recent increasing concerns on building energy efficiency and environmental emissions have prompted a new wave of research in heat pump technology with special emphasis on reducing performance degradation at colder outdoor air temperatures. A summary of the advantages and limitations of several performance improvement options sought for the development of high performance air source heat pump systems for cold climate applications is the primary focus of this paper. Some recommendations for a high performance cold climate heat pump system design most suitable for the US market are presented.

Key Words: heat pumps, cold climate, multi-stage

1 INTRODUCTION

Air-source heat pumps (ASHPs) provide efficient heating by augmenting their energy consumption with heat collected from the ambient air and "pumped" to the required supply temperature. Reversed cycle air-conditioners were presented in the 1930's as a means to efficiently provide heating in buildings (Kerr Jr. *et al.* 1934, Neeson 1938, Brace and Crawford 1938, Labberton 1939). However, these systems where not introduced to the market before the 1950's and started to be reliable and economically feasible in the 1960's as described by Hiller 1976. This industry received an increasing interest following the 1973 oil embargo prompting significant research to improve performance, particularly for cold climate locations.

In late 1975 Carrier Corporation initiated an extensive heat pump research effort (Groff and Reedy 1978 and Groff *et al.* 1979). Four residential split-system ASHPs (based on the vapor-compression refrigeration cycle) located in Seattle, Minneapolis, Syracuse, and Boston were instrumented and monitored for a full year. The field tests illustrated that these heat pumps achieved significant energy saving compared to electric resistance heating systems. Additional studies pointed out the benefits of increasing heat pump capacity for colder

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climate locations despite negative impacts on cooling season performance (Groff *et al.* 1978 and Bullock *et al*. 1980). The average efficiency of residential heat pumps sold in USA increased 2.5% per year in 1980s (Calm 1987). In 1995, EPA introduced Energy Star specifications for residential heating and cooling products, including ASHPs. Today's Energy Star label is only awarded to ASHPs with a Seasonal Energy Efficiency Ratio (SEER) of 14-14.5 (cooling SPF 4.10-4.25) or higher and Heating Season Performance Factor (HSPF) of 8-8.2 (heating SPF 2.34-2.4) or higher[1]. The current most efficient air-source heat pumps have SEERs (cooling SPFs) of 20 (5.86) and higher while the SEER of a heat pump in 1979 was just 7. Thus the energy efficiency of modern air-source heat pumps is almost three times higher than those available 30 years ago. This great efficiency achievement has resulted from technical advances in vapor compression systems and components (e.g. compressors, heat exchangers, and flow control devices, etc.) as well as microprocessor-based control, variable-speed motors, etc., all achieved while making the switch from ozone depleting refrigerants to HFCs (Karen and Herold 1993). However, several issues still negatively impact ASHP heating performance under cold ambient conditions (Roth *et al.* 2009). First, ASHP heating capacity and COP significantly decrease as ambient temperature decreases. Second, ASHPs have the drawback of accumulating frost on outdoor coils, which deteriorates energy efficiency and lowers thermal comfort.

Over the last several decades, a number of technologies and design modifications have been proposed to improve the COP and heating capacity of cold climate heat pumps. Homes and buildings in cold climates usually require higher space-heating design loads than space-cooling design loads. In an effort to increase ASHP energy efficiency in cold weather, US manufacturers are gradually introducing new products specifically designed for better cold weather performance - Hallowell International (Acadia 2010) and Nyle Special Products (Hadely *et al.* 2006) are two examples. These new products use a combination of innovative technologies coordinated by the control systems to enhance their performance. For example, the Acadia™ cold climate heat pump uses a two-cylinder compressor to accomplish efficient multi-stage compression process.

The strategy of the multi-stage vapor injection compression cycle (with multiple compression stages) is becoming attractive to improve the COP and heating capacity of heat pumps at cold operating conditions. Theoretical and experimental results presented by US national laboratories and Universities reported that multi-stage vapor injection compression cycles achieved higher COP and capacity than single-stage cycles (Domanski 1996, Bertsh and Groll 2008, Mathison *et al.* 2011, and Wang *et al.* 2009) at cold ambient conditions. Domanski (1996) evaluated the thermodynamic performance for the ideal two-stage cycles, and conclude that the two-stage cycle improves the COP for every fluid, but the degree of COP improvement is larger for working fluids with large heat capacity. Bertsch and Groll (2008) simulated, designed and tested a two-stage heat pump using R410A at ambient temperature as low as -30°C and supply temperature of up to 50°C. Their heat pump was equipped with two compressors to operate low- and high-stage compression processes. The results show that a COP of 2.1 was achieved at -30°C ambient temperature with double the heating capacity of a conventional heat pump system running at the same conditions. To reduce the cost and system complexity, Wang *et al.* (2009) replaced the two compressors by a single multi-stage scroll compressor with refrigerant vapor injection ports. Vapor injected cycles showed 20% COP gain at -17.8°C ambient temperature versus that of the same cycle without injection. Mathison *et al.* (2011) from Purdue University theoretically analyzed the performance limit for multi-stage vapor compression cycle with continuous refrigerant injection. The results illustrated a COP increase varying from 18% to 51% depending on the refrigeration application, with larger temperature lift cycles benefiting most significantly. At least one residential air-source heat pump using multi-stage compression is already commercially available in US (Acadia 2010, Hadely *et al.* 2006, and Trane 2010).

[1] http://www.energystar.gov/index.cfm?c=airsrc_heat.pr_crit_as_heat_pumps

Use of CO_2 as refrigerant in a vapor compression cycle has also been investigated to improve the performance of cold climate heat pumps. A CO_2 refrigerant cycle can provide 35% greater capacity at low ambient temperature, which can decrease the use of electric resistance heating. CO_2 is considerably different from conventional refrigerants; its critical pressure and temperature are fairly low (7.38 MPa and 31.1°C respectively). Hence, CO_2 vapor compression cycles usually result in a transcritical cycle with subcritical low-side and supercritical high-side pressure (Kim *et al.* 2004). Prototype residential CO_2 heat pump systems have demonstrated higher capacity and comparable COP compared to R410A or R22 heat pump systems at lower outdoor temperatures (Kim *et al.* 2004). The higher capacity of the CO_2 system at lower outdoor temperatures has considerable impact when accounting for the overall seasonal system efficiency for an application, as the dependence on supplementary heating is reduced (Richter *et al.* 2003). Comparison between a R22 unit and a CO_2 prototype system show the CO_2 system achieves 20% better energy efficiency due to a lower need for supplementary heat (Richter *et al.* 2003 and Nekså 2002). The CO_2 technologies are still under development and have great potential for further improvements. A two-stage CO_2 compressor has shown a potential for 20% COP improvement (Kim *et al.* 2004), as was confirmed by Groll and Kim (2007). This new compressor technology is intended for lower-temperature refrigeration applications, but also is of interest to energy saving in air conditioning and heat pumping. Further CO_2 system developments include: using of microchannel heat exchanger, increasing the isentropic efficiency of the compressor, and using an ejector or expander for expansion work recovery. There has been extensive research in improved heat pump cycles and designs over the world; the above review was limited to the current efforts in the US.

2 AIR SOURCE HEAT PUMPS: THEORY AND ADVANCEMENT

The typical configuration of an ASHP consists of a compressor, indoor and outdoor coils (air-to-refrigerant heat exchangers), two expansion valves (one for cooling and one for heating), an accumulator, and a reversing valve. In cooling mode, the indoor coil is the evaporator and the outdoor coil is the condenser, and vice versa in heating mode. This kind of system is widely used in the southern part of the United States where the winter weather is mild. At the AHRI standard heating condition, i.e. 21°C dry bulb indoor and 8.3°C dry bulb/6.1°C wet bulb outdoor (AHRI 2008), a 7.7 HSPF (2.25 heating SPF) ASHP can operate at around 3.5 COP. For high efficiency systems, the COP can be increased to the 4.5 range if larger heat exchanger coils and electronically commutated motors (ECM) are used. Compared to the electric resistance heater, where the COP is always 1, the application of the heat pump has gained significant interest from end users.

However, the COP of ASHPs decreases quickly at low ambient conditions. At the AHRI low temperature heating condition, i.e. 21°C dry bulb indoor and -8.3°C dry bulb/-9.4°C wet bulb outdoor, the COP drops by 30 to 35%. The COP drop for high efficiency systems is less - about 25 to 30%. Even though heat pumps can still operate above 1 COP at low ambient conditions, the heating capacity provided is generally not enough for comfort. Therefore, in colder climate regions, ASHPs generally must use a secondary heat source (usually electric resistance heat, etc.). When the heating load demand is high and the heating capacity from the heat pump alone is not enough, the secondary heat source turns on to supplement the heat output and ensure a comfortable living space.

There are other system configurations that can extend the heat pump application range to lower ambient conditions. Here are few examples:

1. Booster system (Shaw 2007): In this kind of system, a booster compressor is connected in series with the primary compressor. The booster normally has less capacity than the primary compressor. Figure 1 compares the booster cycle to the conventional vapor compression cycle on a P-H diagram. The booster is off during

normal operation. When the ambient temperature is low enough and the primary compressor alone cannot satisfy the load demand, the booster turns on to provide extra heating capacity and boost up the overall system COP.

2. Vapor injection technology (Lifson 2005, Siddharth *et al.* 2004, and Wang *et al.* 2009, Heoa *et al.* 2010): The vapor injection technology creates a multi-stage compression. Figure 2 (a) shows a two-stage vapor injection cycle working principle on a P-H diagram. A phase separator such as flash tank is installed after the expansion valve. The liquid portion goes through a second expansion and circulates to the evaporator, while the vapor portion is injected back to the compressor and creates a second compression effect. The compression process used in this cycle can be accomplished with multiple single-stage compressors or one multi-stage compressor. Scroll compressors can be equipped with vapor injection ports resulting in a single multi-stage compressor. Using a single multi-stage compressor is more cost effective and results in simpler system configuration. The vapor injection cycle is a proven technology that can improve heating capacity and COP at low ambient conditions.
3. Ejector technology (Elbel and Hrnjak 2008): The ejector is used to recover some of the throttling loss at the expansion valve. An ejector takes the high pressure refrigerant from the condenser to be the motive fluid and the low pressure refrigerant from the evaporator to be the suction fluid. This is illustrated as the ejection line on Figure 2 (b). The ejector mixes both fluids and ejects the mixture to a phase separator. The liquid portion goes through an expansion valve and circulates to the evaporator, while the vapor portion goes to the compressor suction. Since the ejector mixed the high and low pressure refrigerants, the suction pressure to the compressor is increased. As a result the system performance is increased.

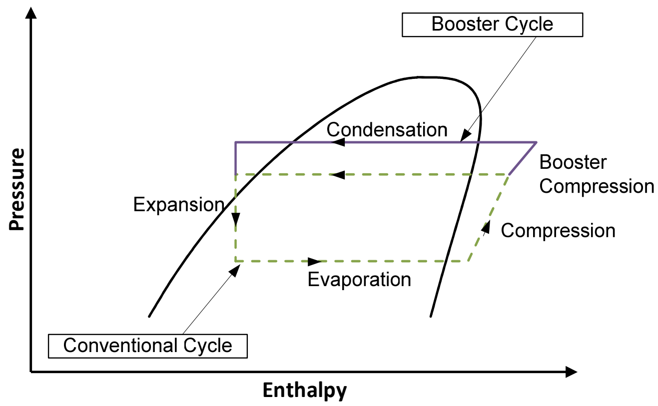

Figure 1: Comparison between conventional and booster cycles.

All of these technologies have a common effect that provides a temperature lift to the vapor compressor cycle; as a result, the heating performance can be improved. However, besides the booster system, the other two technologies have yet to be widely commercialized in the residential market. The recent development of these technologies creates an opportunity to develop a non-conventional, yet more efficient and wider application range ASHP.

Among these three technologies, the vapor injection technology is the most cost effective way to implement and can provide significant performance gain. In this paper we will focus on 3 multi-stage vapor injection cycle configurations: flash tank cycle, economizer heat exchanger (HX) cycle, and flash injection circuit cycle. We developed an in-house modeling tool and used that with an off-the-shelf multi-stage compressor map along with conventional indoor and outdoor heat exchanger designs to study the impact of operating and design parameters on the performance of these cycles. Following is a review on the different multi-stage cycles, simulation results, and discussion.

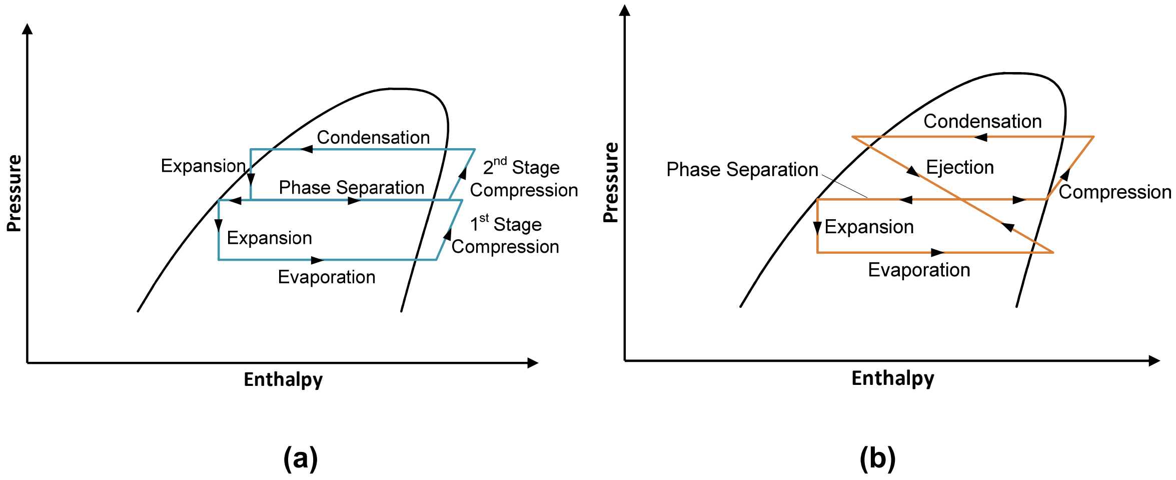

Figure 2: Advanced vapor compression cycles: (a) Vapor injection cycle, (b) Ejector cycle.

3 VARIATION OF VAPOR INJECTION CYCLES

Multi-stage vapor injection compression cycle can be classified into two fundamental configurations: (a) Flash tank cycle and (b) Economizing heat exchanger cycle. Figure 3 shows the schematics of a 2-stage cycle for each configuration.

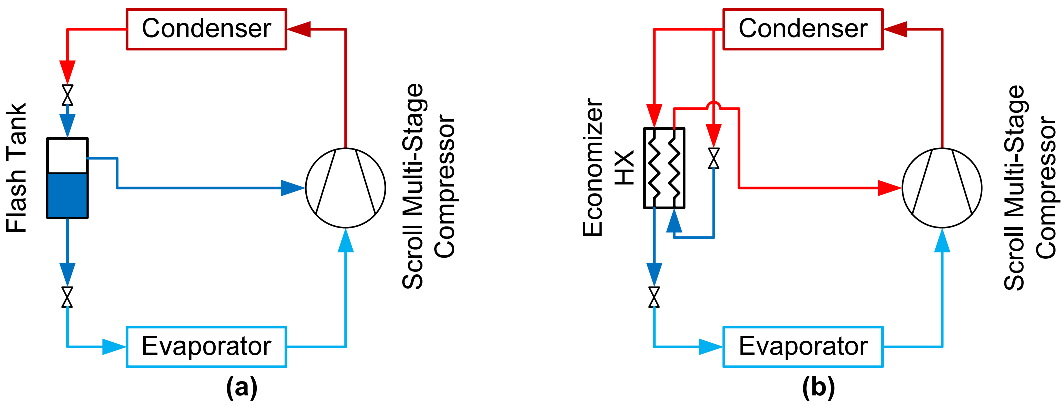

Figure 3: Schematics of two-stage cycles with flash tank (a) and two-stage cycle with economizing heat exchanger (b).

In a two-stage cycle with flash tank (Figure 3 (a)), the two-phase refrigerant after the first expansion is separated into saturated liquid and vapor by a flash tank. It has the advantage of feeding 100% of saturated vapor to the compressor injection port. However, the amount of refrigerant going to the injection port is difficult to control and is solely determined by the high side pressure.

The two-stage cycle with economizing heat exchanger (Figure 3 (b)) allows part of the liquid refrigerant at the condenser outlet to pass through an expansion valve before entering the economizer HX to further subcool the main-stream refrigerant coming from the condenser. The superheated intermediate pressure refrigerant leaving the economizer HX enters the intermediate compressor port. As a result, the separation with economizer HX is not always 100% as compared to the flash tank separation. In the mean time, the subcooled main-stream refrigerant is expanded by a second expansion valve, and then enters the evaporator.

Hence, refrigerant flow rate and pressure entering the intermediate compressor port can be easily controlled using thermostatic expansion valves. As such, this two-stage cycle has been widely investigated. Wang *et al.* (2009) demonstrated that two-stage cycle with economizer HX achieves performance improvement comparable to that of two-stage cycles with flash tank. The former has a wider operating range of injection pressure due to its freedom of setting the injection refrigerant superheat at the injection port. A few commercial heat pump products based on the concept of two-stage cycle with economizer HX have been available.

Besides the basic injection cycle configurations, Takahashi 2010 discussed the benefits of using a flash injection circuit cycle that comprises 3 electronic expansion valves. The cycle configuration of the proposed injection circuit is shown in Figure 4 below. In this design, the subcooled refrigerant leaving the condenser is first slightly expanded into a receiver containing a suction line HX (power receiver). The expanded refrigerant is then further subcooled in an economizer HX similar to that used in the cycle shown in Figure 3 (b). Using electronic expansion valves allows for the intermediate pressure refrigerant to leave the economizer HX at near saturated conditions before it enters the compressor. This results in improved heating performance. The system controller is devised such that the flash injection circuit maintains refrigerant circulation even at lower ambient conditions.

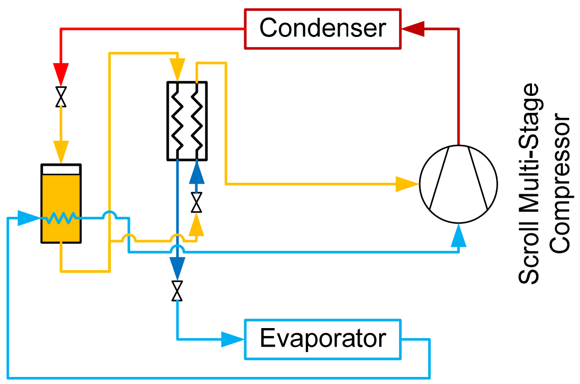

Figure 4: Two-stage cycle with flash injection circuit (Takahashi 2010)

4 RESULTS AND DISCUSSIONS

A system simulation tool was developed in-house. The tool is a component based simulation tool with a Newton-Raphson non-linear solver. Different component models are being used: segmented HX model, modified compressor map to model the multi-stage compressor, overall UA/effectiveness HX model for refrigerant-to-refrigerant economizer HX, and a flash tank model. Component connections are described in a system configuration file; hence any system configuration can be simulated. We have developed 3 system configuration files; one for the flash tank cycle described in Figure 4 (a), one for the economizer cycle described in Figure 4 (b), and the other for the flash injection circuit described in Figure 5. System components have been sized based on existing heat pumps that are available on the market. The compressor is a prototype 5 hp R410A scroll compressor with vapor injection ports.

A numerical experiment was designed to evaluate system performance under varying ambient conditions with different design parameters such as outdoor coil (OD) airflow, OD superheat (SH), indoor coil (ID) airflow, ID subcooling (SC), and economizer(s) effectiveness. The results of this numerical experiment are described in the following subsections.

4.1 Flash Tank Cycle

The flash tank cycle of Figure 3 (a) was modeled. In this cycle, the intermediate pressure was predetermined from the compressor map. A parametric study was constructed to vary

OD airflow (75% to 200% the design value), OD SH (0.56 to 11.11°C), ID airflow (75% to 200% the design value), ID SC (0.56 to 11.11°C). This study has shown that the only parameter that has noticeable impact on the performance at low ambient conditions is the superheat. The optimal OD SH is 2.78°C at design conditions and 0.56°C for the rest of the operating ambient conditions as shown in Figure 5. Furthermore, Figure 5 shows that the performance is almost constant below OD SH of 8°C for design ambient conditions (8.3°C) and below OD SH of 2.78°C for the lowest ambient conditions (-26°C).

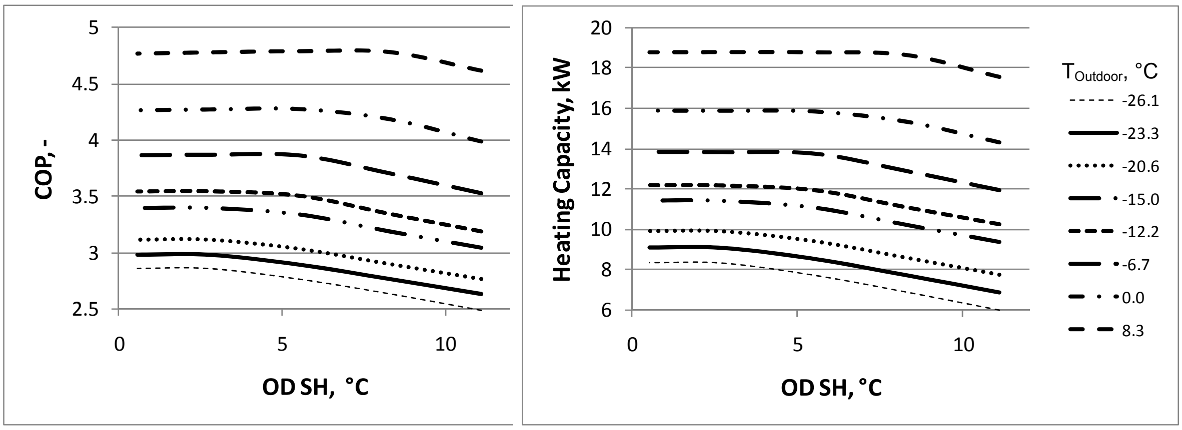

Figure 5: the impact of OD SH on the performance of the flash tank cycle.

The impacts of the other design parameters on the performance of the cold climate heat pump are summarized in Figure 6. In Figure 6, the x-axis represents the value of the different design parameters as a percentage of the design value. The cycle COP, excluding any fan power consumption, was plotted as the dependent value on the y-axis for the lowest ambient conditions of -26°C on the left and the design ambient conditions of 8.3°C on the right. The OD airflow rate had no impact on the performance at low ambient conditions while doubling the OD airflow resulted in only 3.4% performance improvement at the design conditions. Doubling the ID airflow rate resulted in 20% improvement in COP at the design conditions and only 4.5% improvement at low ambient conditions. Finally, the design ID SC of 5.6°C was found to be the optimum at design ambient conditions. At low ambient conditions, the optimal ID SC was found to be 0.6°C resulting in only 1.8% performance improvement.

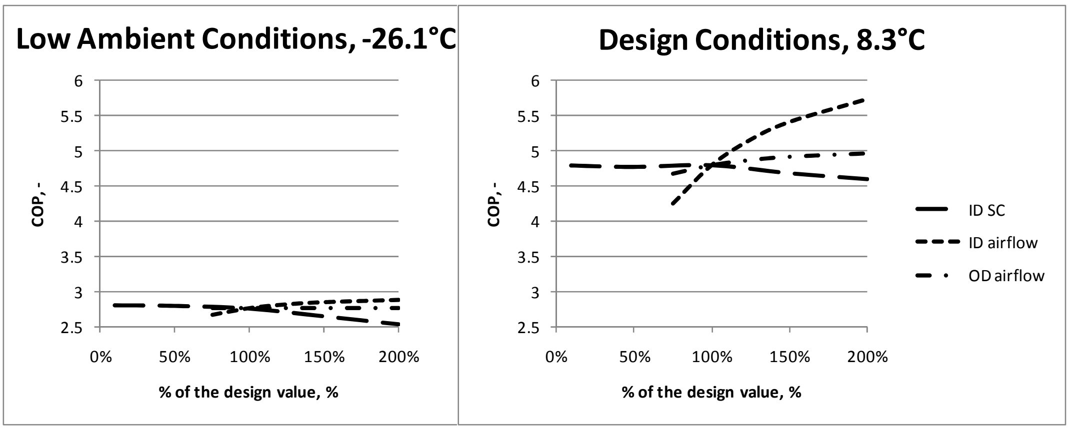

Figure 6: the impact of OD and ID airflow, ID SC on the performance of the flash tank cycle.

4.2 Economizer Cycle

Varying the outdoor airflow rate in the multi-stage economizer cycle of Figure 3 (b) showed strong impact for high ambient conditions; however the performance seemed to be less sensitive to the outdoor airflow rate as the ambient temperature falls below -5°C. On the other hand the refrigerant superheat leaving the outdoor coil had a bigger impact on the system performance. Figure 7 summarizes the impact of OD SH on the performance of the economizer cycle. The results are similar to that of the flash tank cycle. The optimal SH is 5.6°C for the design ambient conditions and 0.6°C for the low ambient conditions. Similar to the flash tank cycle, the performance was insensitive to OD SH below 8°C for the design ambient conditions and 2.8°C for the low ambient conditions. Overall, the economizer cycle showed slight performance degradation varying between 2.6% capacity and 1% COP at design conditions and less than 0.5% at low ambient conditions. This loss in capacity is largely due to having economizer effectiveness of less than 100%.

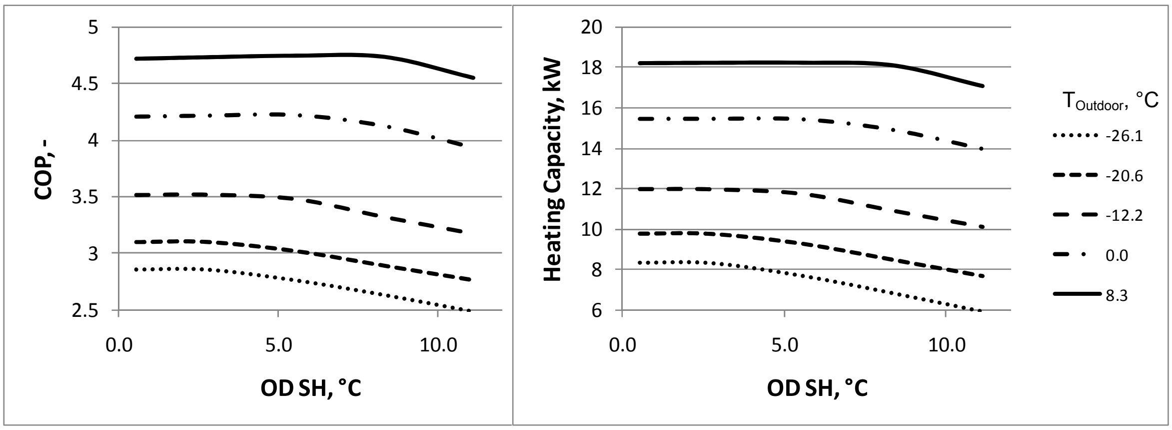

Figure 7: The impact of OD SH on the performance of the economizer cycle.

Based on the previous observation, a parametric study was devised to study the impact of economizer effectiveness on the performance of the economizer cycle. In this study, the economizer overall UA was varied between 75% to 200% of the design value, which is equivalent to 94% to 99.9% economizer effectiveness. This study showed that at low ambient conditions, an economizer with double the overall heat transfer coefficient would result in a heat exchanger effectiveness of 99.9% and would have similar performance to the flash tank cycle. Hence doubling the heat exchanger size resulted in less than 0.4% performance improvement at low ambient conditions. At design ambient conditions, the larger economizers did not improve the cycle performance. This is mainly due to the increased superheat of the injected vapor to the second stage compression chamber. The results of this study are summarized in Figure 8 showing the impact of economizer effectiveness on the cycle COP and heating capacity. These results suggest that smaller economizer can be used at minimal performance degradation as suggested by the tradeoff between added cost of larger economizer and cycle performance improvement.

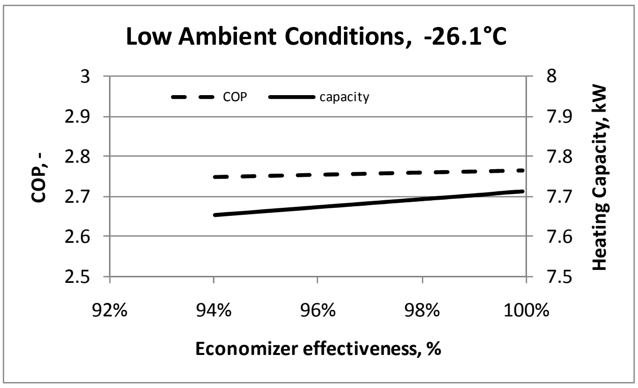

Figure 8: Effect of economizer effectiveness on the system performance.

4.3 Flash Injection Circuit Cycle

The performance of the multi-stage flash injection circuit cycle of Figure 4 was investigated. The power receiver was modeled as a refrigerant-to-refrigerant heat exchanger with constant effectiveness of 0.3 whereas the economizer was modeled as a refrigerant-to-refrigerant heat exchanger with a constant effectiveness of 0.7. The parametric study revealed similar performance sensitivity to that of the flash tank cycle and the economizer cycle. The OD SH was shown to have the most impact on the system performance at low ambient conditions.

The main difference was that 2 refrigerant-to-refrigerant heat exchangers and the required expansion valves were used instead of a single flash tank. Figure 9 summarizes the impact of OD SH on the performance of the economizer cycle. At the design ambient conditions, the multi-stage cycle with flash injection circuit showed better COP than the economizer cycle but at the cost of lower heating capacity. However, at low ambient conditions, the flash injection circuit configuration resulted in lower COP and heat capacity than the economizer cycle. This is largely due to the model assumptions that resulted in economizer effectiveness of 97.6% for the economizer cycle and only 70% for the flash injection circuit.

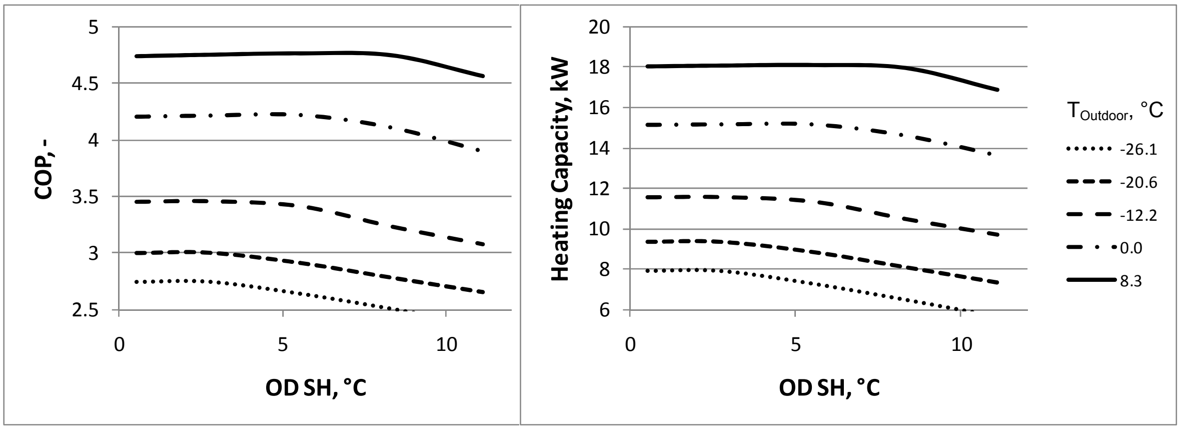

Figure 9: The impact of OD SH on the performance of the economizer cycle.

The impact of the heat exchangers effectiveness on the flash injection circuit cycle performance was further investigated. The power receiver effectiveness was varied between 20% and 70% while the economizer effectiveness was varied between 60% and 95%. The results of this parametric study are summarized in Figure 10. The results indicate that increasing the power receiver effectiveness would improve the cycle COP but would result in lower heating capacity. This is largely due to the decrease in refrigerant mass flow rate associated with the increase in superheat at the compressor inlet. The results shows that changing the power receiver effectiveness from 20% to 70% (about 5 fold increase in heat exchanger size) increased the COP by 0.4% while reducing the capacity by 1.7%. On the other hand, increasing the economizer effectiveness from 60% to 70% (about 3 fold increase in heat exchanger size) increased the COP by 0.4% and increased the heating capacity by 1%. A flash injection circuit cycle with economizer effectiveness better than 75% would surpass the COP of the flash tank cycle but would still suffer from 3% lower heating capacity.

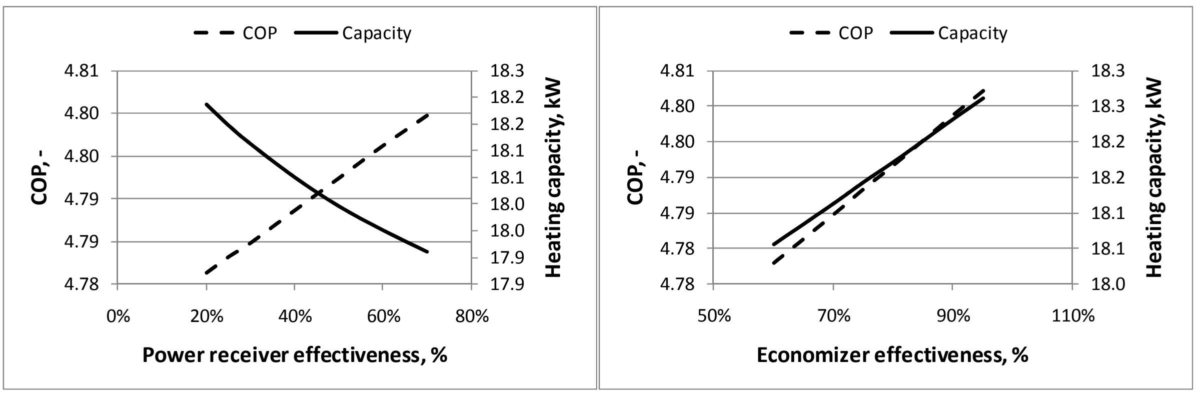

Figure 10: The impact of heat exchangers effectiveness on the performance of the flash injection circuit cycle performance.

5 Discussion

The results shown in section 4 indicated that the flash tank cycle, which is the simplest cycle to model, has the best performance. This is mainly due to the fact that a flash tank would have an effectiveness of 100% and that in our system simulations we relied on the compressor performance map to determine the injection pressure. Flash injection cycles, are simple in analysis but are not simple to implement in a commercial product due to the difficulty of controlling the intermediate pressure and the need for larger system charge and improved refrigerant charge management techniques.

The flash injection circuit cycle provided better COP than the economizer cycle. This cycle can be further optimized by varying parameters such as the expansion prior to the power receiver and the intermediate pressure. This would lead to a new optimized scroll compressor injection ports location. The tradeoff between cost and performance improvements needs to be further investigated in order to have optimal use of materials.

Finally, the compressor performance map used in the current study was developed for fixed injection fluid superheat of 5.6°C and using an economizer with a 5.5°C approach temperature controlled using TXVs (Beeton and Pham 2003). However, in our simulations, the superheat at the injection port was not controlled and the injection ratio was dictated by the system solver. There was no means to provide any corrections to the compressor performance based on the injection ratio and the injection superheat. We believe that there is a need to develop an advanced compressor map for multi-stage injection type compressor that incorporates critical operating parameters such as evaporating, condensing, and intermediate saturation temperatures, injection flow rates, and injection superheat.

6 CONCLUSIONS

Cold climate heat pumps present an opportunity for improved heating efficiency. Multi-stage injection cycles can be used to maintain acceptable system performance at low ambient conditions. A new flexible system simulation tool has been developed and presented in this paper. The new simulation capability was applied to 3 multi-stage injection cycle configurations: flash tank cycle, economizer cycle and flash injection circuit cycle. The results indicated that the simple flash tank cycle showed superior performance. However economizer cycle has negligible performance degradation and is easier to control and design. The flash injection circuit cycle offers additional system flexibility and allows for further performance improvement. These could be designed with higher COP than a flash tank cycle but at the cost of capacity reduction.

7 REFERENCES

AcadiaTM Heat Pump, 2010. “Product & Technology Review#19”, http://tinyurl.com/prodtech, accessed December 30th, 2010.

AHRI. 2008. “ARI Standard 210/240. Performance rating of unitary air-conditioning & air-source heat pump equipment,” Air-Conditioning, Heating and Refrigeration Institute, Arlington, VA.

Beeton, W.L., Pham, H., 2003. “Vapor-Injected Scroll Compressors,” ASHRAE Journal, April 2003, pp. 22-27.

Bertsch, S.S. and Groll, E.A. 2008. “Two-stage air-source heat pump for residential heating and cooling application in northern U.S. climates,” International Journal of Refrigeration, Vol. 31, pp. 1282-1292.

Brace, K. P. and Crawford, R. B. P. 1938. “Reversed Cycle Heating System,” United States Patent Office, Patent number: 2135742, Filing date: Apr 27, 1935, Issue date: Nov 1938.

Bullock, C. E., Groff, G. C., and Reedy, W. R. 1980. “Sizing of Air-to-Air Heat Pumps for Northern Climate Heating Applications,” in Proceeding of International HVAC Congress, Berlin, Germany, April 17-18.

Calm, J.M. 1987. “Heat pumps in the USA,” International Journal of Refrigeration, Vol. 10, pp. 190-196.

Domanski, P.A. 1996.”Theoretical Evaluation of the Vapor Compression Cycle with a Liquid-Line/Suction-Line Heat Exchanger, Economizer, and Ejector,” NIST, NISTIT 5606.

Elbel, S. and Hrnjak, P. 2008. “Experimental validation of a prototype ejector designed to reduce throttling losses encountered in transcritical R744 system operation,” International Journal of Refrigeration, Vol. 31, pp. 411-422.

Groff, G.C., Bullock, C.E., and Reedy, W.R. 1978. “Heat Pump Performance Improvements for Northern Climate Applications,” pp 838-846 in Proceedings of 13th Intersociety Energy Conversion Engineering Conference, San Diego, CA, September.

Groff, G.C. and Reedy, W.R. 1978. “Investigation of heat pump performance in the northern climate through field monitoring and computer simulation,” ASHRAE Transaction, Vol. 84, pp. 767-785.

Groff, G.C., Reedy, W.R., and Bullock, C.E. 1979. “Recent investigations of air-source heat pump performance in cold climates,” XV International congress of refrigeration, Venezia.

Groll, E.A. and Kim, J.-H. 2007. “Review of recent advances toward transcritical CO2 cycle technology, HVAC&R Research, Vol. 13, pp. 499-520.

Hadley, A., Callahan, J. Strok, R. 2006. “Without strip heat: In-situ monitoring of a multi-stage air source heat pump in the Pacific Northwest,” ACEEE Summer Study on Energy Efficiency in Building, 2006.

Heoa, J., Jeonga, M. W. and Kim, Y. 2010. “Effects of flash tank vapor injection on the heating performance of an inverter-driven heat pump for cold regions,” International Journal of Refrigeration, Vol. 33, pp. 848-855.

Hiller, C., C. 1976. “Improving Heat Pump Performance via Compressor Capacity Control Analysis and Test”, Ph.D. Dissertation, Massachusetts Institute of Technology, Department of Mechanical Engineering, January 21st, 1976, pp. 29.

Karen, D.B. and Herold, K. 1993 “Improving heat pumps and air conditioning”, Mechanical engineering, Vol. 115, pp. 98-102.

Kerr Jr., C., Stotz, E., and Stotz, J. K. 1934. “Reversible-Cycle Heating and Cooling Systems,” United States Patent Office, Patent number: 1942295, Filing date: Mar 1, 1933, Issue date: Jan 1934.

Kim, M.-H., Pettersen, J. and Bullard, C.K. 2004. “Fundamental process and system design issues in CO2 vapor compression systems,” Progress in Energy and Combustion Science, Vol. 30, pp.119-174.

Labberton, J. M. 1939. “Reversible Refrigerating Apparatus for Heating and Cooling,” United States Patent Office, Patent number: 2148415, Filing date: Jul 13, 1933, Issue date: Feb 1939.

Lifson. A. 2005. “Novel vapor injection method for scroll compressors,” International Conference on Compressors and their systems, pp. 381-390.

Mathison, M.M., Braun, J.E. and Groll, E.A. 2011. “Performance limit for economized cycles with continuous refrigerant Injection,” International Journal of Refrigeration, Vol. 34, pp. 234-242.

Neeson, C.R. 1938. “Refrigerating System for Cooling and Heating,” United States Patent Office, Patent number: 2131355, Filing date: Jan 25, 1933, Issue date: Sep 27, 1938.

Nekså, P. 2002. “CO2 heat pump systems,” International journal of refrigeration, Vol. 25, pp. 421-427.

Richter, M.R., Song, S.M., Yin, J.M., Kim, M.H., Bullard, C.W. and Hrnjak, P.S. 2003. "Experimental results of transcritical CO2 heat pump for residential application, Energy, 28(2003), 1005-1019.

Roth, K., Dieckmann, J. and Brodrick, J. 2009. "Heat pump for cold climate," ASHRAE Journal, February, 2009, pp. 69-72.

Shaw, D.N. 2007. "USRE39625B Patent, Boosted air source heat pump". Hallowell International, Bangor, ME.

Siddharth, J., Gauary, J. and Clark, B. 2004. "Vapor injection in scroll compressors," Proceedings of International compressor engineering Conference, 2004, Lafayette, IN, USA.

Takahashi, Y. 2010. "Lineup and Features of Air-to-Water Products for Europe," Mitsubishi Electric Advance, June 2010, pp. 13-17

Trane. 2010. Trane XL16c Packaged Heat Pumps, http://www.trane.com/residential/products/packaged-units/XL16c-Heat-Pump-Packaged#, accessed December 16th, 2010.

Wang, B., Li, X., Shi, W. and Yan, Q. 2006. "Effect of refrigerant injection on the scroll compressor," Proceedings of International Compressor Engineering Conference, 2006, West Lafayette, IN, USA.

Wang, X., Hwang, Y. and Radermacher, R. 2009. "Two-stage heat pump system with vapor-injected scroll compressor using R410A as a refrigerant," International Journal of Refrigeration, Vol. 32, pp. 1442-1451.