Application of oil flooded compression with regeneration to a packaged heat pump system

Bin YANG[1]*, Timothy BLATCHLEY[2], Christian K. BACH[3], Jim E. BRAUN[4], W. Travis HORTON[5], and Eckhard A. GROLL[6]

[1]Purdue University, School of Mechanical Engineering,
Ray W. Herrick Laboratories, West Lafayette, IN, US

[1]yang62@purdue.edu
[2]tim.blatchley@gmail.com
[3]bachc@purdue.edu
[4]jbraun@purdue.edu
[5]wthorton@purdue.edu
[6]groll@purdue.edu

* Corresponding Author

ABSTRACT

The heating capacity and coefficient of performance (COP) of conventional air-source heat pumps decreases towards lower ambient temperatures. In addition, high discharge temperature at the compressor discharge might limit the operation of the heat pump at very low ambient temperatures.

Oil injected into the compression chamber at the beginning of the compression process can absorb part of the heat generated during the compression process, which can result in significant reduction of the compressor discharge temperature. Discharge temperature decreases with increasing injected oil mass fraction, especially at low ambient temperatures. Therefore, oil injection allows the application of air-source heat pumps in regions with very low ambient temperatures in winter. Additionally, oil injection decreases the compressor power consumption by providing better sealing and lower friction during the compression process. Furthermore, if oil injection is combined with a regenerative heat exchanger and an oil cooler, the system performance of a vapor compression system can be improved significantly.

The work presented in this paper shows the experimental results of a 5-ton (17.6 kW) R410A packaged heat pump which was retrofitted with an oil injected compressor, integrated indoor heat exchanger oil cooler and regenerator. The effect of different oil mass fractions on the system performance was investigated under standard AHRI 210/240 heating test conditions. According to the results, up to 8% system COP improvement was observed compared to the baseline system for all ambient temperatures.

1. INTRODUCTION

According to a survey conducted by U.S. Department of Energy (2009), space heating energy consumption contributes to 32% of the total residential energy consumption. Compared to electric heating, air-souce heat pump systems have much higher coefficients of performance while producing more heating capacity with the same electric power consumption. However, in regions of very low ambient temperatures, auxiliary electric heating is required. This is a result of increasing load and decreasing heating capacity of conventional single-stage air source heat pumps with decreasing ambient, which leads to a growing gap between heating and available capacity. Considering the relatively high cost of electricity and low coefficient of performance of electric heating, it is attractive to people living in regions with low winter temperatures if an air-source cold-climate heat pump can be used instead of electric heating.

The heating capacity and coefficient of performance of a conventional air-source heat pump drop significantly at low evaporating temperatures. Due to the low evaporating temperatures and relatively constant condensing temperatures, the pressure ratio across the compressor in a heat pump system operating at very low ambient temperatures is high. This leads to excessive discharge temperatures and a low compressor efficiency. A high pressure ratio may also increase wear and and reduced compressor life. In cold climate regions, the heat pump may often be shut down for system protection due to high discharge temperatures, which means that the auxiliary must meet the heating requirements. The above shortcomings reduce the seasonal efficiency of a conventional heat pump in cold climate regions.

To increase the heating capacity and coefficient of performance (COP), cascade systems, variable speed compressors and other high performance components (e.g., expanders for work recovery) can be used. However, the increased cost and compelxity of these systems has inhibited the launch of these modifications in the marketplace. Oil flooded compression might be a more cost effective alternative to address the low temperature cutout issue and to increase COP and capacity at low ambient temperatures.

By injecting liquid of high specific heat capacity into the compressor, the heat generated in the compression process can be absorbed by the liquid, leading to lower refrigerant temperatures, larger refrigerant density and decreased compressor specific work. The discharge temperature can be dramatically decreased, which is good news for heat pump compressors operating in cold climate regions.

Hugenroth (2006) studied the effect of liquid flooding in an Ericsson cycle. It was found that the thermodynamically ideal flooding liquid was water due to its high specific heat capacity. The author also experimentally investigated the effect of flooding with oil on the system performance for a heat pump. A possible COP improvement of up to 13% was found for heating mode (Hugenroth et al., 2006). Li et al. (1992) developed a detailed model for an oil injected scroll compressor focusing on the benefit of the oil on sealing and lubrication. It was found that both the specific compression work and discharge temperature were decreased with the injection of oil. Sakuda et al. (2001) concluded that the compressor efficiency increased with a decrease in the oil injection flow rate, since the effect of injecting oil on the increase of suction heating is predominant compared with its effect on reducing the leakage loss. Hiwata et al. (2002) proposed that the optimal oil flooding rate was a tradeoff between increased leakage at lower oil flow rate and increased suction gas preheat at high oil flow rate. Bell et al. (2012) developed a detailed model for liquid flooded compression process in scroll compressors. In his thesis (2011), the physical background of liquid flooding and its impact on system performance was provided, and also, using a simple cycle model, the coefficient of system performance was found to be 50% higher than for a convertional heat pump when liquid flooding and regeneration technology was employed. Ramaraj (2012) experimentally tested an oil flooded compressor and conducted a simplified system analysis. She estimated a 25% heating COP improvement at the ambient temperature of -10 ℃ for the oil flooded system combined with a regenerator relative to a single-stage baseline heat pump system.

The purpose of this paper is to present experimental results for a 5-ton (17.6 kW) R410A packaged heat pump, which was retrofitted with an oil injected compressor, integrated oil cooler and regenerator. The effect of different oil flooding mass fractions on system performance is investigated. Test data was taken at the standard AHRI 210/240 heating conditions.

3. EXPERIMENTAL FACILITY AND MEASUREMENTS

3.1 Test facility

A commercial 5-ton (17.6kW) R410A packaged heat pump system was modified to use an oil flooded scroll compressor as schematically depicted in Figure 1. After compression, the mixture of oil and refrigerant is separated into two streams in the oil separator – refrigerant vapor and liquid oil. In heating mode, the refrigerant vapor stream goes into the indoor unit where the heat is rejected to the indoor air flow through built-in fin&tube coils. The refrigerant flow at the outlet of indoor unit is further subcooled when flowing through the regenerator (internal heat exchanger, IHX). The liquid refrigerant stream is filtered before it passes through the mass flow meter, and is then expanded to evaporating pressure using two parallel piston expansion valves. The piston expansion valves are used for one flow direction only (i.e. heating mode). In the other flow direction, the piston expansion valve works as a short tube. In the outdoor heat exchanger, the refrigerant flow is evaporated and superheated. The superheated low pressure vapor flow is further superheated in the internal heat exchanger by heat exchange with the subcooled liquid refrigerant flow. The flow passes then through the reversing valve and accumulator to arrive at the compressor suction port. The oil exiting the oil separator flows through the oil cooler rejecting heat to the indoor air flow. One of the six circuits of the indoor coil is used as the oil cooler in heating mode leading to a system that has less indoor heat transfer area for the refrigerant flow than the original single-stage system. After the oil flow meter (FM2), the cooled oil flow is throttled to compressor suction pressure using an electronic expansion valve (EXV2), and then mixed with the refrigerant vapor at the suction port. The electronic expansion valve (EXV1) is closed in heating mode. Figure 2 shows pictures of the experimental setup.

In air conditioning mode, the refrigerant flow direction is reversed after the oil separator. The refrigerant vapor flows through the outdoor heat exchanger first, then through the expansion valve and afterwards, through the indoor heat exchanger. The minimum oil mass flow rate to maintain the compressor oil level is kept through the oil cooler by regulating EXV1. EXV2 is closed since no oil injection is needed in air conditioning mode.

Both the refrigerant and oil flow rates are measured with Coriolis-type mass flow meters. Surface mounted thermocouples are used to measure the temperatures at all the inlets and outlets of the components. A plate heat exchanger is used as the regenerator. Since the size of two parallel piston expansion valves are too big for the flow, an additional electronic expansion valve (EXV3) is used as a pre-expansion device for compressor superheat control. A production type unit would not use this complicated valve arrangement.

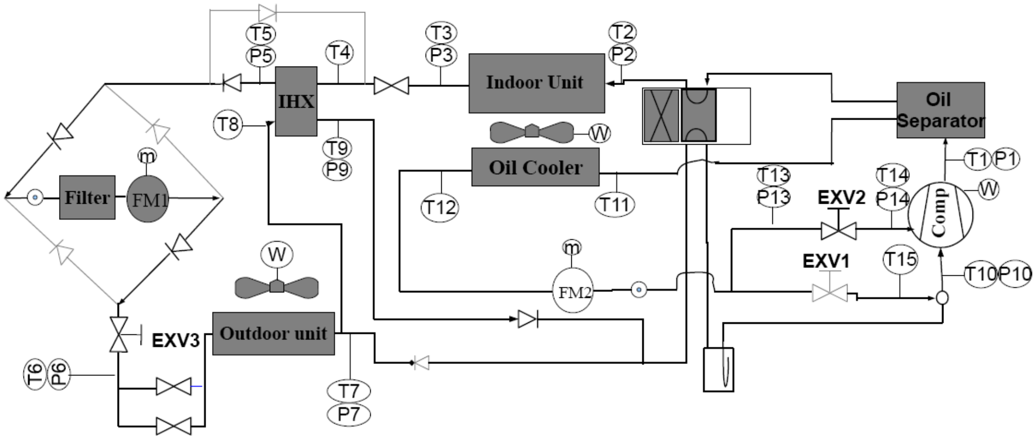
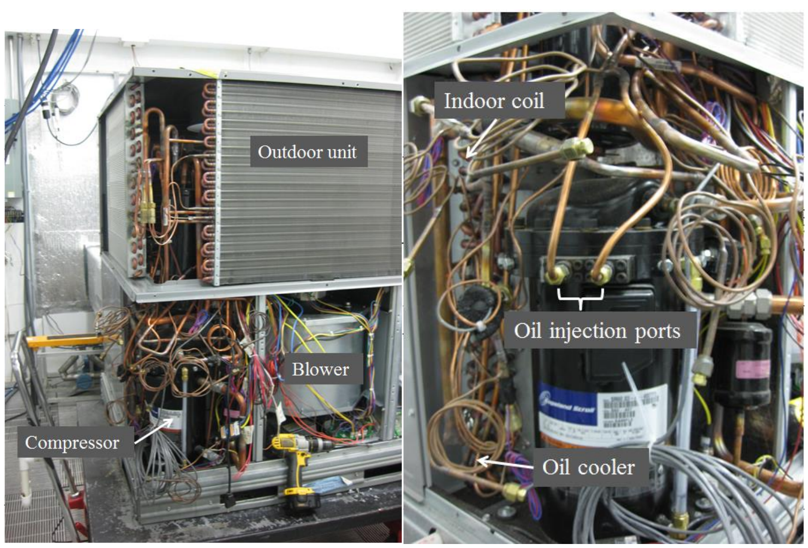


Figure 1: Schematic of a packaged heat pump using oil flooded compression with regeneration technology



(a) Setup Overview **(b) Oil flooded compressor**

Figure 2: Experimental setup

3.2 Test matrix

The modified heat pump was tested in psychrometric chambers in order to fully understand the effect of oil flooded compression with regeneration on system performance in both cooling and heating mode. The test matrix is based on the AHRI 210/240 standard cooling and heating conditions as shown in Tables 1 and 2. Since the focus of the testing was heating mode, the test conditions for cooling mode were limited. For the heating test conditions, an additional test condition (H3*) was added to make sure the heat pump was tested at low ambient temperature.

In order to investigate the impact of oil mass fraction on the system performance in heating mode, the oil mass fraction, which is the ratio of oil mass flow rate to total mixture flow rate, was regulated from 0.0 to 0.3 by regulating the electronic expansion valve (EXV2).

Table 1: Test matrix for oil flooded heat pump, air conditioning mode

Test Name	Oil Mass Fraction	Evap. EDB [°F (°C)]	Evap. EWB [°F (°C)]	Cond. Entering Air Temperature [°F (°C)]	Indoor Air Flowrate [CFM (m^3/s)]
A	0	80 (26. 7)	67 (19.4)	95 (35)	1750 (0.826)
B	0	80 (26. 7)	67 (19.4)	82 (27. 8)	

Table 2: Test matrix for oil flooded heat pump, heat pump mode

Test Name	Indoor Unit [°F (°C)]		Outdoor Unit [^{o}F (°C)]		Indoor Air Flowrate [CFM (m^3/s)]
	EDB	EWB	EDB	EWB	
H1			47 (8.33)	43 (6.11)	
H2	70 (21.11)	<=60 (15.56)	17 (-8.33)	15 (-9.44)	1750 (0.826)
H3*			0 (-17.78)	minimum	

4. RESULTS AND DISCUSSION

The COP improvement of the oil flooded system compared to the baseline system is defind as:

$$COP_{improvement} = \frac{COP_{oilflooded} - COP_{baseline}}{COP_{baseline}} \times 100\% \tag{1}$$

Figure 3 shows the COP improvement at variable oil mass fractions for different ambient temperature and compressor suction superheat. At the ambient temperature of 47°F (8.33 ℃), a larger superheat leads to a higher COP improvement showing that the negative effect of increasing compressor suction superheat on system performance can be lowered using the oil flooding technology. The test data successfully captures the optimum oil mass fraction for low ambient temperature conditions (17°F (-8.33 ℃) and 5°F (-15 ℃)), however, the optimum oil mass fraction is not captured for the ambient temperature of 47°F (8.33 ℃).

The heating capacity improvement of the oil flooded system is shown in Figure 4. It can be seen that the heat pump system modified with oil flooded compression and regeneration does not lead to a large improvement in heating capacity. The heating capacity improvement is in the range of 1.6% to 3.3%. One possible reason can be ascribed to the reduction of the heat transfer area for refrigerant flow in the condenser. In this modified system, one circuit of the indoor coil is used as the oil cooler and thus, the heat transfer area of the condenser is reduced for the refrigerant flow. Additionally, there is another potential problem. The refrigerant mass flow rate in each circuit is increased since the number of circuits is altered, which may lead to an unpredictable maldistribution in the indoor coils and an increase in pressure drop. Considering all these respects, a new package heat pump system retrofitted with oil flooded compression technology is going to be tested. In this new system, additional fin tube coils are added as the oil cooler which will eliminate the above issues.

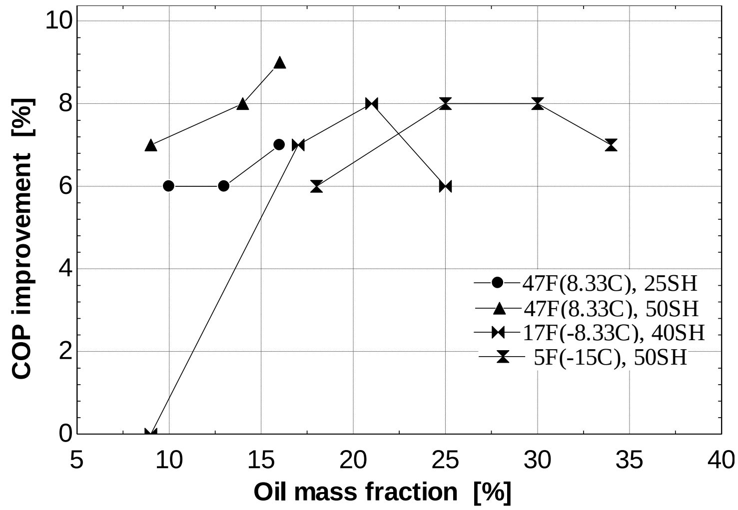


Figure 3: COP improvement for oil flooded system combined with regeneration technology

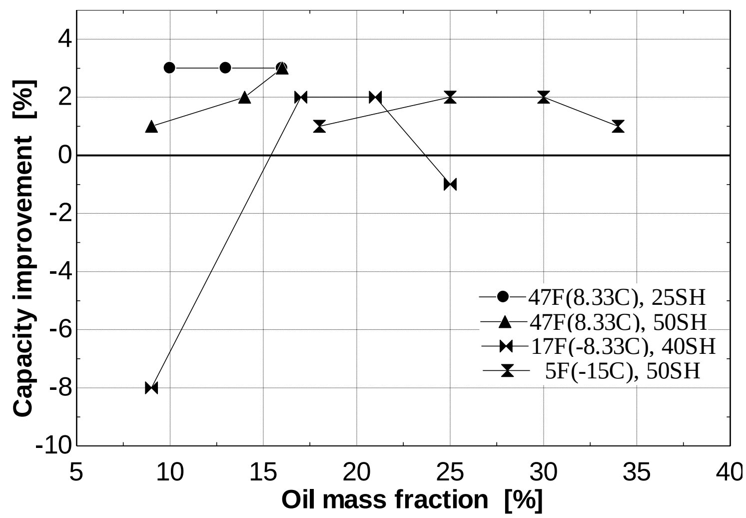


Figure 4: Heating capacity improvement for oil flooded system combined with regeneration technology

Estimates of seasonal efficiency and seasonal COP for the baseline and oil flooded systems in heating mode for climate data associated with Minneapolis, MN, is given in Figure 5. The seasonal efficiency was defined as follows:

$$\text{Heating seasonal efficiency} = \frac{Q_{tot,heating}}{W_{tot,elec}} \tag{2}$$

where $Q_{tot,heating}$ is the total spacing heating required during the spacing heating season, Btu's. $W_{tot,elec}$ is the total electrical energy consumed by the heat pump system during the same season, expressed in watt-hours. The seasonal efficiency was calculated based on a simple bin method where an ambient temperature of -3.43 ℃ was chosen as the balance temperature between heating requirement of the building and heat pump capacity (Heat pump balance point, HPBP). The heating requirement was assumed to be a linear function of ambient temperature below a building balance point, where no heating is required. This corresponds to the heating load curve of application 1 shown in Figure 6. The building heating requirement is covered entirely by the electric auxiliary heat for the standard single-stage system below ambient temperatures of –17.7 ℃. This is a result of the compressor discharge temperature cutout. In contrast, the oil flooded system can operate over the entire range of ambient conditions. Compared to the standard single-stage system, the heat pump system using oil flooded compression combined with regeneration has a 10% higher seasonal efficiency in heating mode. Considering that one circuit of the indoor coil is taken as the oil cooler, which reduces the heat transfer area used for rejecting heat from refrigerant to the indoor air flow (as mentioned before), this is an very positive result. Further improvements in heating seasonal efficiency improvement can be expected if optimized heat exchangers are used for the indoor coil.

Figure 6 shows the heating load of two applications with different building heating requirements and heating capacity provided by the oil flooded system and single stage system. The single stage system is expected be shut down when the ambient temperature is lower than -10 ℃ since the high discharge temperature. For application 1 (heating requirement of 17.6 kW at -10 ℃), the heat pump balance temperature is -3.43 ℃, below which the oil flooded system cannot provide sufficient heating capacity to the building. However, for application 2, the system can provide the necessary capacity of 10.5 kW (3 ton) at an ambient temperature of -10 ℃ without the need for auxiliary heating. Figure 7 shows the COP of the oil flooded system and single stage system at different ambient temperatures. As a result of a higher balance temperature, the system COP for application 1 is significantly lower than that for application 2 at low ambient tempertures, especially below -10 ℃. System COP approaches 1, which is the COP of auxiliary heat, as ambient temperature decreases.

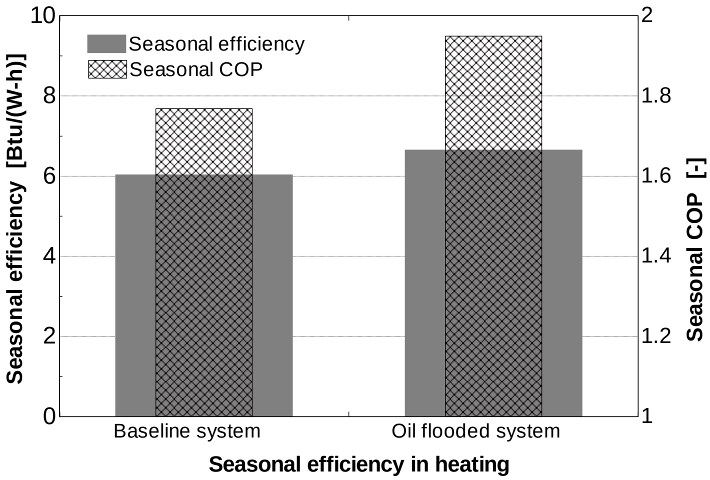


Figure 5: Heating seasonal efficiency and COP improvement of oil flooded system

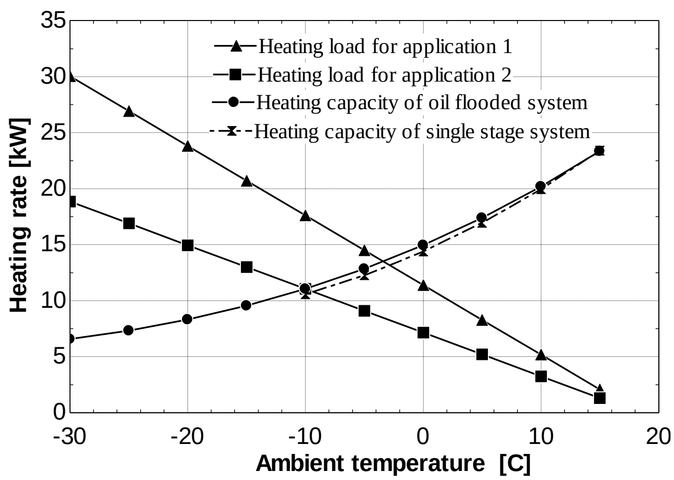
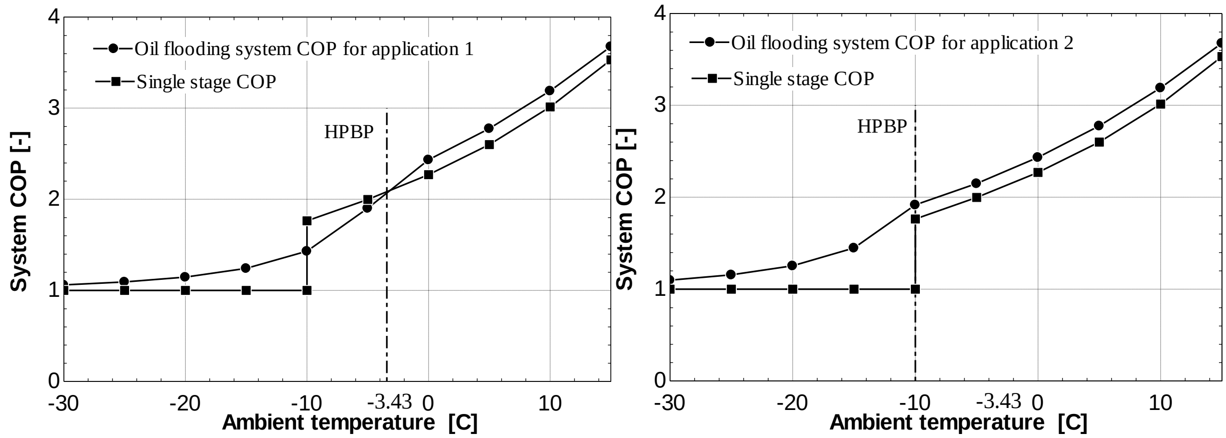


Figure 6: Heating load and heat capacity for different ambient temperatures



(a) application 1 (b) application 2

Figure 7: Oil flooded system COP at different ambient temperatures for application 1 and 2

5. CONCLUSIONS AND FUTURE WORK

The heating COP and capacity of a conventional air source heat pump system decreases towards lower ambient temperatures. A heat pump may have to be shut down due to extremely high discharge temperature when it is operating in very cold climate regions. In this paper, a prototype heat pump system retrofitted with oil flooded compression and regeneration technology was tested. Up to % COP improvement was achieved for the oil flooded system relative to the baseline system, which resulted in at 10% increase in seasonal heating performance since the baseline system was cut out at low temperature. The heating capacity was found to be slightly higher for the oil flooded system than the baseline system. In this system, the integrated oil cooler resulted in less heat transfer area and potentially maldistribution in the indoor coil. This indicates that the full potential of the oil flooded compression technology has not yet been achieved. A new prototype oil flooded system is currently being built. This system has a different baseline configuration in which the indoor coil face area is kept unchanged while an additional coil is added as the oil cooler. This system is expected to achieve a greater performance improvement for the oil flooded system with regeneration than the system presented in this paper.

6. NOMENCLATURE

EDB	Entering dry bulb temperature	**(°F (°C))**
EWB	Entering wet bulb temperature	**(°F (°C))**
COP	Coefficient of performance of the system	(-)
Q	Total spacing heating required during the spacing heating season	(**Btu**)
W	Total electrical energy consumption during the same season	(**Watt-hours**)

Subscripts

oil flooded	Oil flooded system
baseline	Baseline system
tot, heating	Total heating requirement
tot, elec	Total electrical energy consumption

REFERENCES

ANSI/AHRI 210/240, 2008, ANSI/AHRI Standard 210/240 with Addendum 1(formerly ARI Standard 210/240), 2008 Standard for Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment, available from http://www.ari.org/search+standards.aspx

2009 RECS Survey Data http://www.eia.gov/consumption/residential/data/2009/#housing-characteristics.

Bell, I., 2011, Theoretical and Experimental Analysis of Liquid Flooded Compression in Scroll Compressors, PhD thesis, Purdue University.

Bell, I., Lemort, V., Groll, E., Braun, J., King, G., and Horton, W.T., 2012, Liquid-flooded Compression and Expansion in Scroll Machines – Part I: Model Development, *International Journal of Refrigeration*, Vol. 35, No. 7, p: 1878-1889.

Hiwata, A., Iida, N., Futagami, Y., Sawai, K., and Ishii, N., 2002, Performance Investigation with Oil-injection to Compression Chambers on CO2-scroll Compressor, *Compressor Engineering Conference at Purdue University*, Paper #C18-4.

Hugenroth, J., 2006, Liquid flooded Ericsson cycle cooler, PhD thesis, Purdue University.

Hugenroth, J., Braun, J., Groll, E., and King, G., 2006, Oil Flooded Compression in Vapor Compression Heat Pump Systems, *IHR-IRHACE Conference*, p: 492-499.

Li, H., Wang, D., Wang, H., and Chen, P., 1992, Research of oil-injected scroll compressor working process, *Compressor Engineering Conference at Purdue University*, p: 118b1-118b13.

Sakuda, A., Sawai, K., Iida, N., Hiwata, A., Morimoto, T., and Ishii, N., 2001, Performance improvement of scroll compressor with new sealing-oil supply mechanism, *International Conference on Compressors and their Systems*, number C591/019.

Ramaraj, S., 2012, Vapor Compression Cycle Enhancement for Cold Climate Heat Pumps, Master thesis, Purdue University.

ACKNOWLEDGEMENT

The authors would like to thank their colleagues at the Herrick Labs for their help with the design and testing of the system. In addition, the authors greatly appreciate the support of the research sponsors, Carrier Corporation, Indianapolis, IN and Emerson Climate Technologies, Sidney, OH.