Purdue University Purdue e-Pubs

International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2010

Evaluation of a Virtual Refrigerant Charge Sensor

Woohyun Kim Purdue University

James E. Braun *Purdue University*

Follow this and additional works at: http://docs.lib.purdue.edu/iracc

Kim, Woohyun and Braun, James E., "Evaluation of a Virtual Refrigerant Charge Sensor" (2010). *International Refrigeration and Air Conditioning Conference*. Paper 1121. http://docs.lib.purdue.edu/iracc/1121

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html

Evaluation of a Virtual Refrigerant Charge Sensor

Woohyun Kim¹, James E. Braun^{*1}

¹Purdue University, Mechanical Engineering, West Lafayette, IN, USA

*Corresponding Author: jbraun@purdue.edu

ABSTRACT

The primary goal of work described in this paper is to evaluate and enhance a virtual refrigerant charge sensor, developed in a previous study. The virtual refrigerant charge sensor algorithm employs low-cost and non-invasive measurements (i.e. surface mounted temperature measurements) to estimate refrigerant charge level for packaged air conditioning systems. It can be embedded within a portable device (i.e. a PDA) for a technician's use in the field or permanently installed on units. Based on the evaluations for a wide range of systems and conditions, the virtual charge sensor was found to work well in estimating refrigerant charge for systems that do not utilize accumulators when using the original default parameters. For systems with accumulators, however, the parameters needed to be improved. A new method for determining default parameters was developed that depends on three elements: liquid line length, rated subcooling, and rated charge. The liquid line length is particularly important because a substantial amount of refrigerant is stored as liquid. The parameters decreased the errors between the actual and predicted charge. Even better performance was achieved for the virtual refrigerant charge sensor when the improved parameters were tuned, minimizing the errors by using test data and linear regression. Overall, the enhanced method provided estimates of refrigerant charge that were within 10 percent of the actual charge over a wide range of operating conditions for a number of different systems.

1. INTRODUCTION

Studies conducted by various investigators (Proctor and Downey, 1995; Cowan, 2004; Li and Braun, 2006b) have shown that more than 50 percent of packaged air conditioning systems are improperly charged. Improper refrigerant charge can increase energy usage, reduce capacity, and decrease equipment lifespan. Furthermore, refrigerant charge leakage can contribute to global warming in the long term. The Montreal protocol restricted the manufacture of some refrigerants that impact of the ozone layer, whereas the Kyoto's protocol addresses refrigerants that contribute to the greenhouse effect. The laws governing chlorofluorocarbon now do not allow HVAC&R contractors to add Freon to a leaky system. They are first required to find and fix the leak, or they may lose their license.

Based on other research covering more than 4,000 residential cooling systems in California, it is clear that many systems have incorrect refrigerant charge levels (Proctor 2000). Data from these tests indicate that about 34 percent are undercharged, 28 percent are overcharged, and only 38 percent have correct charge. Additional data for residential cooling systems in the field from Blasnik et al. (1996) and Proctor (1997, 1998) indicated that an undercharge of 15 percent is common.

Despite the fact that there are slight differences between manufacturers, the typical approach currently used to verify refrigerant charge in the field involves the use of either superheat at the compressor inlet or subcooling at the condenser outlet. These approaches can only determine whether the charge is high or low, not the level of charge. In order to find a charge level, a technician needs to evacuate the system and weigh the removed charge. The correct amount of charge is then added to the system using a scale. This is time-consuming and costly. In addition, the current charge verification protocols utilize compressor suction and discharge pressure to determine refrigerant saturation temperatures that are used in calculating superheat and subcooling. However, the measurement of pressures requires the installation of gauges or transducers that can lead to refrigerant leakage. As a result of these limitations, a virtual refrigerant charge sensor was developed (Li and Braun: 2006a, 2009) that uses a correlation in

terms of superheat and subcooling that are determined using surface mounted temperature sensors. The method can obtain refrigerant charge levels using low-cost surface mounted temperature sensors without disturbing the system, can use readily available manufacturers' data to estimate empirical parameters for the algorithm, and is relatively insensitive to the existence of other system faults.

Based on the previous research, the virtual charge sensor was found to work well in estimating the refrigerant charge for systems that do not utilize accumulators. This paper presents evaluations of virtual refrigerant charge sensor performance based on testing data for different system types, including systems with accumulators, under a wider range of testing conditions, including heating mode. It also presents a simulation method to estimate empirical parameters of the charge algorithm and presents results associated with tuning of empirical parameters.

2. VIRTUAL REFRIGERANT CHARGE SENSOR ALGORITHM

The algorithm of the virtual refrigerant charge sensor utilizes evaporating, condensing suction line, and liquid line temperatures as inputs, as shown in Figure 1. The data acquisition device provides input channels for the four temperature sensors (e.g., thermocouples) and provides calibrated measurements as inputs to the steady-state detector and virtual sensor algorithm. The virtual refrigerant charge sensor is based on the steady-state operating conditions. Therefore, the state detection algorithm filters out the transient data. This algorithm uses a fixed-length sliding window of recent measurements to compute the slope of the best-fit line and standard deviation about the mean values. The virtual refrigerant charge algorithm uses steady state measurements and empirical parameters. A refrigerant charge display interface shows the refrigerant charge gauge readings to users.



Figure 1. Measurements and Scheme of the Virtual Refrigerant Gauge

Li and Braun (2007) developed a virtual refrigerant charge algorithm for correlating the refrigerant charge level in terms of superheat and succooling. Deviations from nominal charge are related to superheat and subcooling using four empirical parameters according to

$$\frac{\left(m_{total} - m_{total,rated}\right)}{m_{total,rated}} = \frac{1}{K_{ch}} \left\{ \left(T_{sc} - T_{sc,rated}\right) - K_{sh/sc} \left(T_{sh} - T_{sh,rated}\right) \right\}$$
(1)

where *m* is the actual total charge, m_{rated} is the nominal total refrigerant charge, $K_{sh/sc}$ and K_{ch} are two constant characteristics of a given system, and $T_{sc,rated}$ and $T_{sh,rated}$ are liquid line subcooling and suction line superheat at rated conditions with the nominal charge, respectively.

The two constants $T_{sc,rated}$ and $T_{sh,rated}$ can be readily obtained from technical data provided by manufacturers. As presented by Li and Braun (2009a), $K_{sh/sc}$ and K_{ch} can be estimated using the following equations.

$$K_{ch} = \frac{m_{total,rated}}{K_{sc}} = \frac{1_{sc,rated}}{(1 - \alpha_o) \cdot X_{hs,rated}}$$

$$(2)$$

$$K_{sh/sc} = \frac{K_{sh}}{K_{sc}} = \frac{(I_{sc} - I_{sc,rated})}{(T_{sh} - T_{sh,rated})}$$
(3)

where $X_{hs,rated}$ is the ratio of high-side charge to the total refrigerant charge at the rated condition and α_o is the ratio of refrigerant charge necessary to have saturated liquid at the exit of the condenser to the rated refrigerant charge.

In this paper, three different approaches were considered for determining the empirical parameters within the refrigerant charge algorithm: default parameters, simulation parameters, and tuned parameters determined with regression applied to the measurements to improve the accuracy of the virtual refrigerant charge sensor.

Based on data available from Harms (2002), a reasonable estimate value for $X_{hs,rated}$ was found to be 0.73 whereas a value of 0.75 was determined for α_o as default parameters. A reasonable estimate for $K_{sh/sc}$ for systems using a TXV or FXO as the expansion device is 1/2.5 based on test results. For a system using an EEV as the expansion device, superheat remains constant regardless of refrigerant charge, and the refrigerant inventory in the evaporator is relatively constant. In this case, a reasonable estimate for $K_{sh/sc}$ value is 0. According to Li and Braun (2009a), the virtual refrigerant charge sensor worked well with these values, unless the system was extremely over or undercharged. Also, the original default parameters did not include the effect of variations in liquid line length.

To overcome these limitations, an improved method for estimating K_{ch} was developed that is based on a simulation approach. K_{ch} should depend on three elements of each system: the liquid line length, the rated subcooling, and the rated charge. Different split and packaged systems can have very different liquid line lengths. The rated subcooling and the rated charge also vary as well, depending on each unit. K_{ch} can be calculated from the refrigerant mass distribution in the system. The total charge in a refrigerant system is given by

$$M_{total} = M_{vapor, pipe} + M_{liquid, pipe} + M_{evaporator} + M_{compressor} + M_{condenser}$$
(4)

where $M_{compressor}$ is the mass in the compressor, $M_{vapor pipe}$ is the mass in the vapor piping, $M_{liquid pipe}$ is mass in the liquid piping between the evaporator and condenser, and $M_{evaporator}$ and $M_{condenser}$ denote the mass within the evaporator and condenser, respectively.

 $M_{condenser}$ is the summation of the mass within the superheated, two-phase (= vapor + liquid refrigerant), and subcooled regions of the condenser. Similarly, $M_{evaporator}$ is mass within the two-phase and superheated regions. $M_{compressor}$ does not need to be considered since it is constant regardless of the refrigerant amount. The refrigerant mass within the single-phase region can be calculated using geometries of the system and properties of the refrigerant. The refrigerant mass of the two-phase sections requires use of void fraction models. The void fraction is generally presented as a function of mass quality, x, and combinations of various properties which remain constant for a given average evaporator or condenser saturation temperature. The quality was assumed to vary linearly with tube length in the two-phase sections which corresponds to the assumption of a uniform heat flux. The void fraction correlations based on the homogeneous equation from Rice (1987) and the mass flux dependent method developed by Tandon (1985) and Zivi (1964) were considered. The refrigerant mass in the heat exchanger is estimated by adding up the refrigerant mass of the single-phase and two-phase regions.

Then, α_o and $X_{hs, rated}$ are estimated from the refrigerant mass distribution in each component.

$$X_{hs,rated} = \frac{M_{condenser} + M_{liquid,pipe}}{M_{tot} + M_{tot} + M_{tot} + M_{tot}}$$
(5)

$$\alpha_o = \frac{M_{cond,sc=0}}{M_{condenser} + M_{liquid,pipe}} \tag{6}$$

$$M_{cond,sc=0} = M_{cond,wapor} + M_{cond,wo-phase}$$
⁽⁷⁾

where $M_{cond,sc=0}$ is the refrigerant mass necessary to have saturated liquid exiting the condenser at the rating conditions.

Table 1 shows comparisons of parameters determined from this calculation approach with parameters determined directly from the measurements of Harms (2002) for three different. For these calculations, void fraction was determined by Zivi model.

Parameters	from measurement	s of Harms	Parameters from simulation approach			
K_{ch}	α_o $X_{hs,rated}$		K_{ch}	$lpha_o$	$X_{hs,rated}$	
56.76	0.73	0.73	60.81	0.72	0.71	
23.97	0.7	0.78	32.26	0.77	0.76	
59.29	0.82	0.68	57.81	0.78	0.56	

Table 1 Comparison between parameters based on measurements and calculations

Alternatively, the empirical parameters within the virtual refrigerant charge sensor algorithm can be tuned to improve accuracy if data are available over a range of refrigerant charge levels and operating conditions. The parameter tuning method minimizes the errors between predicted and known refrigerant charge by using linear regression techniques. The linear regression techniques are applied to all of the available data points for each system: which can include variations in charge level, outdoor flow rate, indoor flow rate, ambient temperature, and indoor dry bulb temperature. More data leads to more accurate parameter values in the tuning process but it requires more time and therefore has higher cost.

As an alternative approach, linear regression was processed with three data points and the outcomes were compared with the tuned parameters obtained from using all data points. Three data points, selected based on different refrigerant charge levels and ambient temperatures: low charge in high ambient temperature, rated normal charge in moderate ambient temperature, and high charge in low ambient temperature. By considering these six conditions to determine three data points, the data points can fairly represent the initial values for overall data. The combinations of conditions were found to be work well in determining parameters of the virtual charge algorithm.

3. EVALUATION OF VIRTUAL REFRIGERANT CHARGE SENSOR

3.1 Existing Laboratory Test

Existing laboratory data were obtained for different systems which were operated in cooling mode over an average range of refrigerant charge levels from 70 to 120 percent and at only one indoor temperature of 27 C. The systems include a window unit, residential split systems, and light commercial packaged systems with different types of compressors. The systems used either a TXV or FXO as an expansion device and R-22, R-407c or R-410a as a refrigerant. Some of the units included low-side accumulators. The data was used to perform initial evaluations of virtual refrigerant charge indicator. Most of the tests (10 out of 14 systems) were performed at the nominal condenser and evaporator airflow rates and at one ambient temperature. Table 2 shows the range of refrigerant charge and other conditions considered for each unit.

The accuracy of the virtual refrigerant charge sensor was evaluated for all of test data in terms of RMS deviation from the actual charge levels presented on a percentage basis. The performance of the virtual refrigerant charge sensor was evaluated based on default, simulation, and tuned parameters with all points and three points and results are shown in figures 2to 5.

When the default parameters were applied, the virtual refrigerant charge sensor worked well for the systems without an accumulator but showed very large RMS errors for systems with accumulators. For the systems without an accumulator, the performance of the virtual refrigerant charge sensor was within 5 percent over a large variation of refrigerant charge amount. However, the use of the default parameters led to some significant errors greater than 10 percent in refrigerant charge estimates for the systems with accumulators. For the system with EEV and with tandem compressor, the errors were over 30 percent.

System	Capacity (ton)	Refrigerant	Refrigerant Charge Level [%]	Expansion Device	Accumulator	Assembly Type
Ι	2.5	R-407c	86~144	TXV	Х	Split
II	5.0	R-22	$78 \sim 127$	TXV	Х	Packaged
III	7.5	R-22	$80 \sim 148$	TXV	Х	Split
IV	3.0	R-410a	86 ~ 122	FXO	Х	Packaged
V	3.0	R-410a	$58 \sim 130$	FXO	0	Split
VI	3.0	R-410a	57~113	TXV	Х	Split
VII	0.45	R-22	61 ~ 141	FXO	Х	Window
VIII	3.0	R-22	$75 \sim 125$	TXV	0	Split
IX	4.0	R-22	$80 \sim 100$	EEV	1500 [cc]	Split
Х	4.3	R-22	60 ~ 110	FXO	No / 1000 [cc]	Split
XI	4.3	R-22	$75 \sim 100$	FXO	No / 1000 [cc]	Split
XII	4.0	R-22	60 ~ 100	FXO	Yes	Split

Table 2 System description of existing refrigerant charge level test data





Fig. 2 Virtual refrigerant sensor performance for existing data based on default parameters



Fig. 3 Virtual refrigerant sensor performance for existing data based on simulation parameters



Fig. 4 Virtual refrigerant sensor performance for existing data based on tuned parameters using three data points

Fig. 5 Virtual refrigerant sensor performance for existing data based on tuned parameters using all data points

When the parameters were estimated using the simulation approach, there was a significant improvement compared to using the original default parameters. The virtual charge predictions were within 5 percent for all systems with an accumulator. In particular, results for systems with an EEV and tandem compressor were noticeably improved with errors of less than 7 percent. When the simulation parameters were employed then the RMS overall error was 4.2 percent while the default parameters yielded an overall RMS error of 11.2 percent.

To increase the accuracy of charge determination, the parameters were tuned for each specific system based on measurements obtained at different refrigerant charge levels, using three data points and all available data points. Overall, the virtual refrigerant charge sensor algorithm predicted the actual charge levels (relative to nominal charge) within 3.4 percent based on three data points and within 2.4 percent based on all data points. The results verified that the tuned parameters were more accurate than the parameters determined from simulation. Tuning the parameters can also lead to very significant improvements in cases where the simulation parameters do not work well, such as at extremely low outdoor temperatures and very high charge level. Compared to using three data points, use of all data points for training led to some improvements.

3.2 New Laboratory Test Data

The need for additional testing was verified through analysis of the existing data. The existing data were limited to 1) cooling mode only, 2) 27 C as the lowest outdoor temperature condition, 3) one indoor temperature conditions, 4) 57 percent as the lowest refrigerant charge level, and 5) systems that do not incorporate multi-speed fans. To better assess the accuracy and broaden the application of the virtual refrigerant charge sensor, new test plans were established considering the following key issues: 1) heating mode operation, 2) operation under lower outdoor temperature than 27 C, 3) various indoor temperature conditions, 4) lower levels of refrigerant charge, and 5) a system with multi-speed fans. In particular, it was vital to have data for heat pumps under heating mode operation in order to evaluate the algorithm for the operation during winter. The data for lower outdoor temperatures in cooling condition were necessary to test the validity of the algorithm during off-season when regular maintenance procedures are often performed. Furthermore, indoor temperatures and refrigerant charge levels vary in the field. According to a diagnostic company, refrigerant charge levels in the field can be as low as 40 percent which can lead to compressor failure.

Two heat pump units were selected for testing and installed within the psychrometric chambers at Herrick Laboratory, Purdue University. One unit employed R-22 as the refrigerant, whereas the other used R-410a. Both units incorporated low-side accumulators and multi-speed fans. The laboratory test plans included 1) heating mode conditions, 2) ambient temperatures ranging from 5 to 45 C for cooling and -10 to 15 C for heating mode, 3) indoor temperature ranges from 20 to 32 C for cooling and 16 to 20 C for heating mode, 4) refrigerant charge level ranging from 40 to 130 percent. The specification of the unit is given in Table 3 and the testing conditions in cooling and heating mode are given in Table 4.

System	Size (ton)	Refrigerant Type	Expansion Device	Accumulator	Assembly Type	
XIII	3.0	R-22	TXV (Cooling / Heating)	0	Split	
XIV	3.0	R-410a	TXV (Cooling)	0	Split	
	5.0		FXO (Heating)	0		
V V	3.0	R _22	TYV	0	Split	

Table 3 System description for laboratory test units

Tuble + Testing conditions for habitation (costs)									
	Mode	Indoor / Outdoor Temperature						Indoor Unit	Refrigerant
System		٨	D	C	р	Б	F	Air Flow Rate	Charge
		A	D	U	D	E			Level
		(C)					(CFM)	(%)	
XIII	Cooling	20/10	20/35	20/45	20/10	20/35	20/45	800(a,b,c), 1600(d,e,f)	$70 \sim 130$
	Heating	21/-8	21/1	21/8	21/-8	21/1	21/8	900(a,b,c), 1500(d,e,f)	$70 \sim 130$
XVI	Cooling	21/4	21/35	21/51	27/4	27/4		1000	$40 \sim 130$
	Heating	15/-8	15/8	15/16	21/-8	28/-8		1000	$40 \sim 130$
XV	Cooling	20~32 / 5~52					Auto	100	

Table 4 Testing conditions for laboratory tests

Figure 6 presents performance of the virtual refrigerant sensor based on default parameters. The RMS errors are about 22 percent. The test results showed relatively large errors in predicted refrigerant charges for both cooling and heating mode. As the refrigerant charge level decreased, there was bigger difference between predicted and real charge amount. The errors were also large at low ambient temperature. For example, the virtual sensor predicted 20 percent of nominal charge when the system was charged at 40 percent of nominal charge in cooling mode.

Figure 7 shows results based on the use of parameters determined from simulation. The RMS errors was reduced to 17 percent. When the simulation parameters were applied in cooling mode, the virtual refrigerant charge sensor showed better performance than when the default parameters were applied. The use of simulation parameters led to very significant improvements in cases where the default parameters did not work well. However, the use of simulation parameters led to some significant errors in refrigerant charge estimates at low charge level. Also, when the outdoor temperature was low refrigerant charge error was large compared to other test conditions. This is because there were cases at low outdoor temperature and low charge when subcooling was zero. In heating mode, there was improvement in the charge predictions but the errors were still large at high charge levels.









Fig. 7 Virtual refrigerant sensor performance for new lab data based on simulation parameters



Fig. 8 Virtual refrigerant sensor performance for new lab data based on tuned parameters using three data points

Fig. 9 Virtual refrigerant sensor performance for new lab data based on tuned parameters using all data points

Figure 8 shows performance of the virtual refrigerant charge sensor based on tuned parameters determined using three data points. The RMS errors were reduced to 12 percentCharge predictions were improved at low charge level in cooling mode but not at high charge in either cooling or heating mode. Refrigerant charge prediction errors were also large at low charge levels in heating mode.

Figure 9 shows performance based on parameters tuned using all the data. The RMS errors were reduced to 8.2 percent. The RMS errors were reduced in cooling mode but were relatively high (over 5 percent) at low charge and low ambient when subcooling was zero. For heating, it was possible to make accurate charge evaluations when refrigerant charge was less than 100 percent. However, when refrigerant charge was over 100 percent, the additional refrigerant charge was stored within the accumulator with little effect on subcooling and superheat. As a result, the charge sensor did not work well for these cases. Overall, there may not be a significant advantage in detecting refrigerant charge under these circumstances. It also was not possible to determine the refrigerant charge in heating mode at a low ambient temperature of -8 C. At this condition, superheat, subcooling, and system performance (heating capacity and energy consumption) were relatively insensitive to charge. Additional work is necessary to

accurately and robustly determine charge level for heat pumps with accumulators when operating in heating mode at high charge or at low ambient temperatures.

6. CONCLUSIONS

The performance of the virtual refrigerant charge sensor with the original approach for estimating default parameters worked well for the systems with no accumulators at moderate to high outdoor temperatures. However, the performance was significantly worse for units with accumulators and at low outdoor temperatures in both cooling and heating mode. An improved method for estimating default parameters was developed to overcome the limitations and provided improved performance in many cases. Even better performance was achieved when parameters were tuned. When the algorithm was tuned for each system using all available data, then the overall RMS error for the virtual charge sensors was 3.77 percent, compared to 5.63 percent when only three data points were used. The only cases where the virtual refrigerant charge sensor with tuned parameters had difficulty were for heat pumps with accumulators when refrigerant was overcharged or at low ambient temperatures. This is due to the overcharged refrigerant being stored in an accumulator. When the improved default parameters were employed then the RMS overall error was 7.37 percent while the original default parameters yielded an overall RMS error of 13.87 percent. The only cases where the virtual refrigerant charge sensor had difficulty were for heat pumps with accumulators when refrigerant charge sensor had difficulty were for heat pumps with accumulators when refrigerant charge sensor had difficulty were for heat pumps with accumulators when refrigerant charge sensor had difficulty were for heat pumps with accumulators when refrigerant charge sensor had difficulty were for heat pumps with accumulators when refrigerant was overcharged or at low ambient temperatures. It is due to the overcharged refrigerant being stored in an accumulator which interrupts the accurate detection.

The virtual refrigerant charge sensor is an improvement over existing charge checking methods because it indicates the charging amount and not just whether the charge is high or low. It is very robust against both variations in operating conditions and impacts of other faults and can be easily implemented at low costs in terms of both hardware and software. The virtual refrigerant charge sensor is also generic for different types of systems. The virtual refrigerant charge sensor could be used as part of a permanently installed control or monitoring system to indicate charge level and/or to automatically detect and diagnose low or high levels of refrigerant charge. It could also be used as a standalone tool by technicians in order to determine existing charge and during the process of adjusting the refrigerant charge.

ACKNOWLEGEMENTS

This work was co-supported by the California Energy Commission (CEC), U.S. Department of Energy (DOE), and the Purdue Research Foundation.

NOMENCLATURE

EEV	Electronic expansion valve			Subscripts
FXO	Fixed orifice		compressor	Compressor
k _{ch}	empirical constant		condenser	Condensing
k _{sc}	condenser geometry constant	evaporator	Evaporating	
k_{sh}	evaporator geometry constant	hs	high side	
$k_{sh/sc}$	empirical constant		hs,o	high side for zero-subcooling
k_{th}	threshold for k		li	Liquid Line
М	Refrigerant mass	(Ibm)	ls	Low side
т	actual total charge	(Ibm)	ls,0	low side for zero-superheat
TXV	Thermostatic expansion valve		SC	subcooling
$X_{hs,rated}$	Ratio of high side charge to the total refrigerant charge at rated condition		sc,rated	rated subcooling
Т	temperature	(C)	sh	superheat
X_{tt}	Lockhart-Martinelli correlating parameter		sh,rated	rated superheat

	Greek	suction	suction
α _o	Ratioof refrigerant charge necessary to have saturated liquid existing the condenser at rating conditions to the rated refrigerant charge	rated	nominal total
		total	total

REFERENCES

Blasnik, M., T. Downey, J. Proctor, and G. Peterson. 1996, "Assessment of HVAC Installations in New Homes in APS Service Territory", Proctor Engineering Group Report for Arizona Public Service Company

Cowan, A., 2004, Review of recent commercial rooftop unit field studies in the Pacific Northwest and California. Northwest Power and Conservation Council and Regional Technical Forum, Portland, Oregon, October 8

Harms, T.M., 2002, "Charge Inventory System Modeling and Validation for Unitary Air Conditioners", Ph.D. Thesis, Herrick Labs 2002-13, Report No. 5288-2, Purdue University, West Lafayette, IN.

Li, H. and Braun, J.E., 2006a, Virtual Refrigerant Charge Level Gauge. U.S. Disclosure 64440-P1-US, filed January 18, 2006.

Li, H. and Braun, J.E., 2006b, "Evaluation of a Decoupling-Based Fault Detection and Diagnostic Technique – Part II: Field Evaluation and Application," Journal of Harbin Institute of Technology, Vol. 13, Supplemental Issue, Pages 164-171.

Li, H. and Braun, J.E., 2007, "Evaluation of a Virtual Refrigerant Charge Level Gauge for Vapor Compression Equipment," IIR Congress of Refrigeration, Beijing, China.

Li, H. and Braun, J.E., 2009, "Virtual Refrigerant Pressure Sensors for Use in Monitoring and Fault Diagnosis of Vapor Compression Equipment", In Press, HVAC&R Research.

Proctor, J. and Downey, T., 1995, Heat pump and air conditioner performance. Handout from oral presentation. Affordable Comfort Conference, Pittsburgh, PA, March 26-31.

Temple, K.A. and Hanson, O.W., 2003, Method of determining refrigerant charge level in a space temperature conditioning system. U.S. Patent Number 6571566.

Proctor, J., 1997, "Field Measurements of New Residential Air Conditioners in Phoenix, Arizona", ASHRAE Transactions ,103(1): 406-415. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Proctor, J. 1998, "Monitored in-situ Performance of Residential Air-Conditioning Systems", ASHRAE \ \ Transactions, 104(1): 1833-1840. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Rice, C.K., 1987. "The effect of void fraction correlation and heat flux assumption on refrigerant charge inventory predictions", ASHARE Transactions 93, pp.341-367.

Tandon, T.N., Varma, H.K., and Gupta, C. P., 1985, "A void fraction model for annular two-phase flow," International Journal of Refrigerant, Vol 16, pp. 373 – 389.

Zivi, S.M., 1964, "Estimation of steady-state steam void-fraction by means of the principle of minimum entropy production." Journal of Heat Transfer, Vol. 86, pp. 247-252.