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ABSTRACT

This paper presents a thorough evaluation of a method for determining refrigerant charge that employs low-cost, non-invasive measurements (i.e., surface mounted temperature measurements). The method could be used as part of a protocol for verified service providers (VSPs) in AC diagnostic tune-up or refrigerant charge, air flow (RCA) verification programs. Ultimately, the method could be embedded within a portable virtual refrigerant charge gauge for a technician's use or permanently installed on the AC unit. The accuracy of the virtual refrigerant charge sensor method is evaluated in this paper using laboratory data for a number of different systems and over a wide range of operating conditions. The systems include residential split systems and light commercial packaged systems employing either fixed capillary tubes, thermostatic expansion valves or electronics expansion valve, and R-22 as the refrigerant. The virtual refrigerant charge sensor is shown to have good performance in terms of accuracy and robustness and has the potential to be easily implemented and installed.

1. INTRODUCTION

A number of studies conducted by various investigators (Proctor and Downey, 1995; Cowan, 2004; Li and Braun, 2006b) concluded that approximately 50 percent of all air conditioners do not have the correct refrigerant charge due to improper service or leakage. Furthermore, these problems cause the equipment to operate 10 to 20 percent less efficiently than expected for normal operation. Utilities in California have developed incentive programs to encourage HVAC service contractors to tune up residential and small commercial air conditioners. A significant aspect of these programs involves refrigerant charge verification. In addition, Title 24 of the California code requires refrigerant charge verification for new installations and retrofits.

Another study sponsored by the American Council for an Energy-Efficient Economy (ACEEE) concluded that improper charging of air conditioners and poor maintenance could increase energy use in homes by 20% and waste 17,600 Terawatt-hour of energy nationwide every year (Neme, 1999). There is a direct linkage between CO_2 production (global warming) and energy efficiency, and between refrigerant leakage and ozone depletion.

In order to determine charge level with current practice, a technician generally evacuates the system and weighs the removed charge. The correct amount of charge would then be added to the system using a scale. This method is very time-consuming and costly. In addition, the current charge verification protocols utilize compressor suction and discharge pressures to determine refrigerant saturation temperatures that are used in determining superheat and condenser subcooling. However, the measurement of pressures requires the installation of gauges or transducers that can lead to refrigerant leakage (Li and Braun, 2006b).

As a result of these limitations, Li and Braun (2006a) proposed a method of obtaining refrigerant charge level that uses a correlation in terms of superheat and subcooling determined with surface mounted temperature sensors. The method could be used within a standalone tool by technicians in order to determine existing charge and during the

process of adjusting the refrigerant charge. Li and Braun (2006b) presented a performance evaluation and prototype demonstration for five different types of systems and the method was shown to have very good performance in terms of accuracy and robustness.

There are other patents related to refrigerant charge diagnosis. Most of the methods (e.g., Temple and Hanson, 2003) are able to determine system charge quantitatively but require relatively expensive pressure measurements and significant training data to fit a model and have not been tested extensively. The method evaluated in the current paper can obtain refrigerant charge levels using low-cost surface-mounted temperature sensors without disturbing the system, can use readily available manufacturers' data for training, and is relatively insensitive to the existence of other system faults. The current paper presents an extensive performance evaluation of this method based on experimental results. Four different types of systems are used for evaluation of the effects of variations in air-conditioner operating conditions and refrigerant charge on the performance of the virtual refrigerant charge sensor. The systems include three commercial split air-conditioners incorporating low-side accumulators with different types of compressors and expansion valves and a residential heat pump.

2. BRIEF DESCRIPTION OF THE VIRTUAL REFRIGERANT CHARGE GAUGE

Figure 1 depicts a typical vapor compression refrigeration cycle and the inputs and output for the virtual charge sensor method of Li and Braun (2006a). The inputs include four temperature measurements (circled symbols). As illustrated in Figure 1, the virtual refrigerant charge sensor consists of four basic components: a data acquisition device, a steady-state detector, a charge derivation algorithm, and a display interface. Each of these components is described in the paragraphs below.

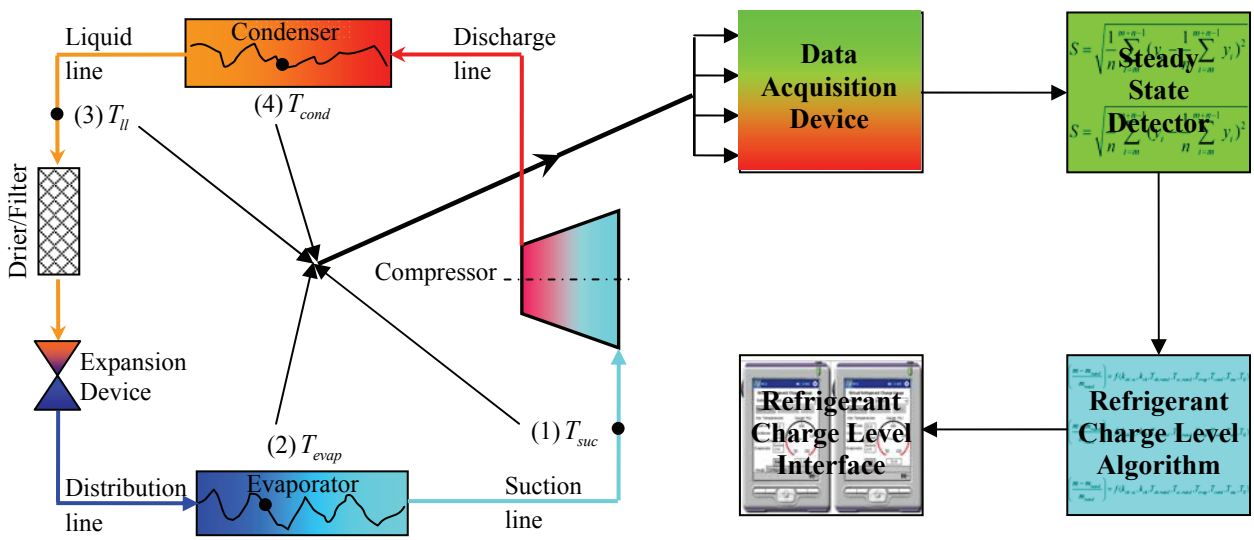

Figure 1. Measurements and Scheme of the Virtual Refrigerant Gauge

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. It could be a standalone device or integrated within a larger device that incorporates the algorithms and interface.

The steady-state detector is used to filter out the transient data, since the charge sensor algorithm is based on steady-state operating conditions. A combined slope and variance steady-state detection algorithm (Li and Braun, 2003) is used. This algorithm uses a fixed-length sliding window of recent measurements to compute the slope (k) of the best-fit line (Equation (1)) and standard deviation (S) about the mean (Equation (2)). If both of the slope and standard deviation for the sliding window are smaller than corresponding thresholds (K_{th} and S_{th}), the system is assumed to be in a quasi-steady condition. The sliding window is specified by the number (n) of data points (y_m, y_{m+1} …,y_{m+n-1}) and sampling time (τ).

$$y_i = a + k(i-m)\tau, \qquad i = m, m+1, ..., m+n-1 \tag{1}$$

$$S = \sqrt{\frac{1}{n}\sum_{i=m}^{m+n-1}(y_i - \frac{1}{n}\sum_{i=m}^{m+n-1} y_i)^2} \tag{2}$$

The refrigerant charge level algorithm is the core of the virtual refrigerant charge gauge. Most of the refrigerant exists as liquid (saturated or subcooled) for normally charged units. In addition, the majority of the refrigerant charge (up to 70%) typically accumulates as liquid in the high pressure side of the system, including the condenser and liquid line (piping and filter/drier). If there is condenser subcooling, then the liquid line is completely filled with liquid and its liquid volume doesn't change with refrigerant charge. In addition, the volume of liquid in the two-phase section is relatively constant because the refrigerant quality varies between 0 and 1 in a nearly linear manner. Therefore, changes in high-side refrigerant charge mostly occur in the subcooled section of the condenser. Furthermore, the volume of liquid within the subcooled section of the condenser is nearly proportional to the amount of subcooling at the exit, T_{sc}, and the liquid density is nearly constant throughout the high-side of the system. With this discussion in mind, the high-side refrigerant charge is related to subcooling using

$$m_{hs} = k_{sc}T_{sc} + m_{hs,0} \tag{3}$$

where $m_{hs,0}$ is the high-side refrigerant mass for the case of zero subcooling and zero refrigerant quality and K_{sc} is a constant that depends on the condenser geometry. $m_{hs,o}$ is assumed to be a constant, independent of operating conditions and total charge.

Much less of the total refrigerant charge resides in the low-side and therefore it is less important to have an accurate characterization. Within the low side, most of the refrigerant exists as liquid in the two-phase section of the evaporator and in the sump of the compressor, absorbed within the oil. The volume of liquid within the two-phase section is assumed to be proportional to the total volume of the two-phase section. Furthermore, the volume of the superheat section is nearly proportional to the exit superheat, T_{sh} and the liquid density is nearly constant throughout the low-side of the system. From these considerations, the low-side charge is related to the superheat according to

$$m_{ls} = m_{ls,0} - k_{sh}T_{sh} \tag{4}$$

where $m_{ls,o}$ is low-side refrigerant charge for the case of zero superheat and refrigerant quality of one, and k_{sh} is a constant that depends on the evaporator geometry. $m_{ls,o}$ is assumed to be a constant, independent of operating conditions and total charge.

Using equation (3) and (4) as a starting point, Li and Braun (2006a) developed an algorithm that relates the four measurements and five constants to the system charge level in terms of deviations from nominal charge.

$$\left(\frac{m - m_{rated}}{m_{rated}}\right) = f(k_{sh/sc}, k_{ch}, T_{sh,rated}, T_{sc,rated}, T_{evap}, T_{cond}, T_{suc}, T_{ll}) \tag{5}$$

where m is the actual total charge, m_{rated} is the nominal total refrigerant charge, $k_{sh/sc}$ and k_{ch} are two constants characteristic 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 last four inputs on the right side are measurements from the data acquisition device and steady-state detector. The output from the algorithm is relatively independent of other faults and operating conditions and applicable to systems having either fixed or variable-area expansion devices.

Finally, the refrigerant charge display interface is a means by which the virtual refrigerant charge gauge provides readings to users. It could be a stand-alone tool for use by service technician and implemented on a PDA or pocket PC or a unique hardware platform having an LCD or a pointer-dial display. Alternatively, it could be implemented within a larger control system having a PC interface.

Three of the constants in Equation 5, m_{rated}, $T_{sc,rated}$ and $T_{sh,rated}$, can be readily obtained from technical data provided by manufacturers at a rating condition. Li and Braun (2006a) suggested reasonable default values for the other two constants, $k_{sh/sc}$ and k_{ch}. Furthermore, they related these constants to other system parameters that could be estimated from detailed information regarding charge distribution between different components within a system. Alternatively, the $T_{sc,rated}$, $T_{sh,rated}$, $k_{sh/sc,}$ and k_{ch} parameters can be tuned for a particular system using measurements

obtained at different levels of refrigerant charge. In this case, regression techniques are applied to the data to minimize errors between prediction and known values of charge.

3. ALGORITHM EVALUATION USING DATA OBTAINED EXTERNALLY

Data was provided by an equipment manufacturer that includes the effects of refrigerant charge. There was not sufficient information to directly determine refrigerant charge distribution. Due to the lack of information about refrigerant charge distribution, the virtual charge sensor algorithm was evaluated based on the use of default parameters and parameter tuning methods. To determine refrigerant charge distribution directly, different test conditions and precise geometry data would be needed.

Specifications for the three systems are given in Table 1 and testing conditions are listed in Table 2. All of the data were obtained through laboratory testing. Systems with R-22 as the refrigerant and with an electronic expansion valve (EEV) and fixed expansion orifices (FXO) were considered. All tests were performed at nominal evaporator and condenser airflow rates and ambient temperatures for cooling conditions. The data obtained from the tests were used to evaluate the virtual charge sensor.

Table 1 System descriptions for manufacturer's data

	Capacity [KW]	Compressor	Refrigerant	Evaporator Tube Dia. (mm)	Condenser Tube Dia. (mm)	Expansion device	Accumulator	Assembly type
I	14.5	Tandem	R22	9.52	9.52	EEV	1500 [cc]	Split
II	15.2	Rotary	R22	9.52	7.0	FXO	None	Split
				9.52	9.52		1000 [cc]	
III	14.5	Reciprocating	R22	9.52	7.0	FXO	None	Split
				9.52	9.52		1000 [cc]	

Table 2 Testing conditions for three air conditioners

System	Air outdoor temperature		Air indoor temperature		Indoor unit air flow rate	Refrigerant charge level
	Dry Bulb(C)	Wet Bulb (C)	Dry Bulb (C)	Wet Bulb (C)		% Nominal Charge
I	27	19	35	24	Nominal Air Flow Rate	80 ~ 100
II	27	19	35	24	Nominal Air Flow Rate	60 ~ 110
						75 ~ 100
III	27	19	35	24	Nominal Air Flow Rate	60 ~ 100
						80 ~ 100

The first system incorporates an electronic expansion valve and an accumulator. The refrigerant charge levels were varied between about 80% and 100% of nominal with ambient temperatures for cooling conditions. Only nominal airflow rates were considered for the evaporator and condenser. Evaporator superheat and condenser subcooling were determined using refrigerant temperature measurements. The second system has versions both with and without an accumulator. R22 was used as the refrigerant and a rotary type compressor was employed. The system without an accumulator used a condenser with 7.0 diameter tubes and was tested with refrigerant charges between 60~100[%]. The system with an accumulator used a condenser with 9.0 diameter tubes and was tested with refrigerant charges between 75~100[%]. Evaporator superheat and condenser subcooling were determined using refrigerant temperature measurements. The third system was tested under the same condition as the second system but with refrigerant charges between 60~100[%] for the version without an accumulator and 80~100[%] with an accumulator.

Figures 2 and 3 show performance of the virtual refrigerant charge gauge for the three systems. Figure 2 gives performance based on the default parameters, whereas Figure 3 shows results obtained after parameter tuning. For the systems not having an accumulator (II and III without accumulator), the performance of the virtual sensors was good (within ±5%) over a large variation of refrigerant charge even without parameter tuning. However, the use of

default parameters led to some significant errors (greater than ±10%) in refrigerant charge estimates for the systems with accumulators (system I and II and III with an accumulator). As shown in Figure 3, there is a significant improvement in accuracy when the parameters are tuned using measured charge data for systems having accumulators. With parameter tuning, the virtual charge predictions are generally within about ±5% of the actual charge for these systems.

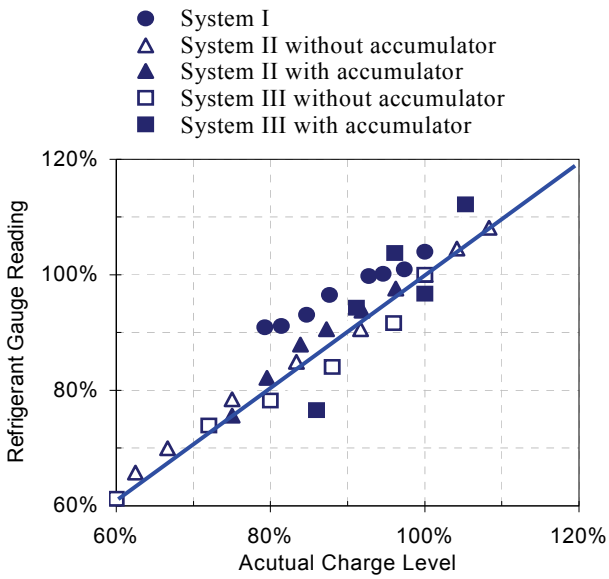

Fig. 2 Virtual refrigerant gauge performance for systems based on default parameter values

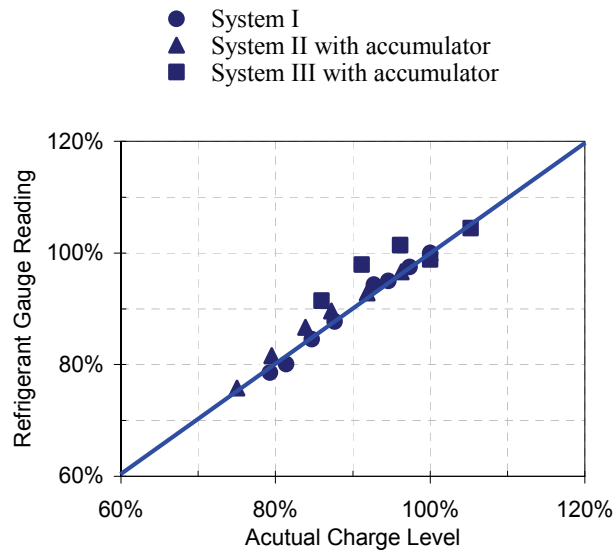

Fig. 3 Virtual refrigerant gauge performance for systems based on tuned parameter values

4. ALGORITHM EVALUATION USING DATA OBTAINED INTERNALLY

Laboratory tests were performed on a heat pump system in cooling mode. There was not sufficient information to directly determine refrigerant charge distribution so the virtual charge sensor algorithm was evaluated based on both default parameters and parameter tuning methods.. Specifications for the system are given in Table. 3 and testing conditions are listed in Table 4. The system has a nominal cooling capacity of 3 tons, uses R-22 as the refrigerant and a TXV as the expansion device and has an accumulator. Different ambient conditions and air flows were considered for a range of refrigerant charges as described in Table 4. Refrigerant charge was varied between about 70% and 130% of normal charge. The effects of reduced air flow for the condenser and evaporator were considered. Only surface temperature measurements were used to determine evaporator superheat and condenser subcooling.

Figure 4 to 7 show the gauge performance based on both default parameter values and tuned parameter values for the unitary air conditioner. Figure 4 and 5 present comparisons between predicted and measured refrigerant charges for the default parameters. Without parameter tuning, there are relatively large errors in predicted refrigerant charges for this system. The RMS errors are about 16.3%. Results obtained after parameter tuning are presented in Figures 6 and 7. The tuning leads to much better predictions of refrigerant charge in comparison to use of the default parameters. The RMS errors are reduced to 6.12% for a large variation of ambient driving conditions and in the presence of other severe fault conditions such as low outdoor air flow rates. However, the refrigerant charge errors increased at high ambient temperatures and low indoor wet bulb temperatures. Additional testing is needed at a wider range of operating conditions to further validate the method.

Table.3 System descriptions for R22 heat pump

System	Size (ton)	Refrigerant type	Expansion device	Accumulator	Assembling type
IV	3.0	R22	TXV	yes	Split

Table.4 Testing conditions for cooling mode

Test Conditions	Air entering indoor temperature	Air entering outdoor temperature	Indoor unit air flow rate	Refrigerant charge level
	Dry Bulb (C)	Dry Bulb (C)	(CFM)	% Nominal Charge
A	20.0	10.0	800, 1600	70 ~ 130
B	20.0	35.0	800, 1600	70 ~ 130
C	20.0	45.0	800, 1600	70 ~ 130

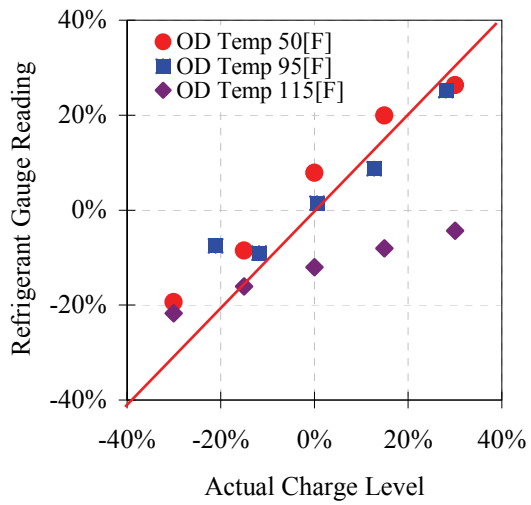

Fig. 4 Virtual refrigerant gauge performance for system IV with air flow rate 800 [CFM] based on default parameter values

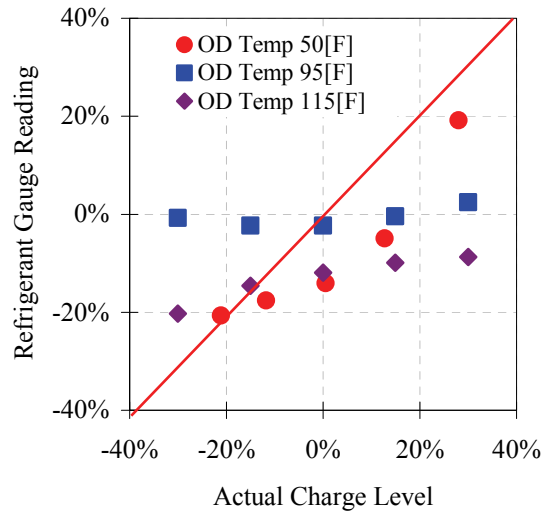

Fig. 5 Virtual refrigerant gauge performance for system IV with air flow rate 1600 [CFM] based on default parameter values

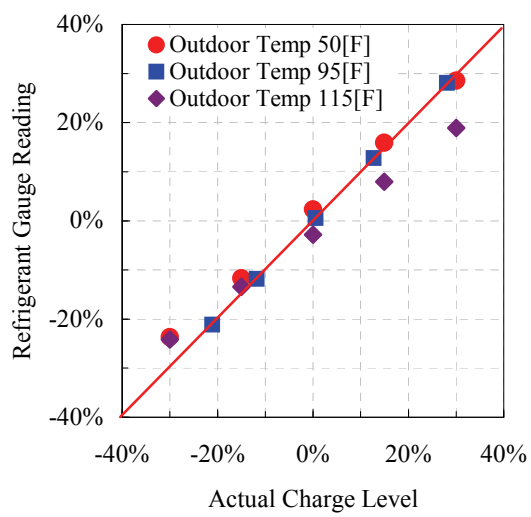

Fig. 6 Virtual refrigerant gauge performance for system IV with air flow rate 800 [CFM] based on tuned parameter values

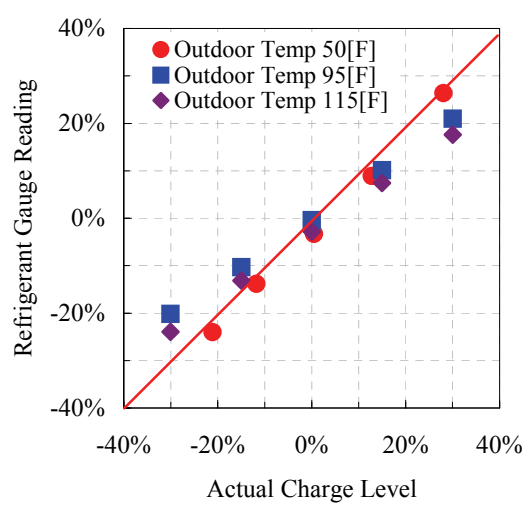

Fig. 7 Virtual refrigerant gauge performance for system IV with air flow rate 1600 [CFM] based on tuned parameter values

5. OVERALL ACCURACY AND IMPACTS ON SYSTEM PERFORMANCE

Table 5 summarizes the overall accuracy of the virtual charge gauge based on default and tuned parameters determined with the data described in the previous sections. The performance of the virtual charge sensor is measured in terms of RMS deviation from the actual charge levels presented on a percentage basis. Overall, the virtual charge sensor works very well as compared with experimental data using default parameters for systems without accumulators. However, it is important to tune parameters for systems having accumulators. The uncertainty in refrigerant charge determination appears to be less than about 5% for systems without accumulators and less than about 10% for systems with an accumulator. For system IV, the virtual sensor based on the tuned parameters worked well in most cases but did result in significant errors at high ambient temperatures.

Table 5 Gauge performance based on both default empirical and tuned parameters

System	Error	
	Default parameters	Tuned parameters
1	4 .0~ 12.0 [%]	0.0 ~ 1.0 [%]
2	-3.3 ~ 0.2 [%]$^{1)}$	-
	- 5.3 ~ 0.0 [%]$^{2)}$	-2.8 ~ 0.0 [%]$^{2)}$
3	-1.9 ~ 4.4 [%]$^{1)}$	-
	-6.9 ~ 9.4 [%]$^{2)}$	-5.5 ~ 0.8 [%]$^{2)}$
4	-10.7 ~ 10.9 [%]$^{3)}$	-6.3 ~ 3.7 [%]$^{3)}$
	-22.5 ~ 27.6 [%]$^{4)}$	-8.9 ~ 9.0 [%]$^{4)}$
	-8.3 ~ 38.7 [%]$^{5)}$	-5.9 ~ 12.4[%]$^{5)}$

** Results for system 2, 3 are divided according to 1) no accumulator and 2) accumulator.*
** Results for system 4 are divided according to 3) low, 4) moderate, and 5) high temperature.*

Important performance indices for an air conditioner are cooling capacity and energy efficiency (COP). Undercharging refrigerant can reduce air conditioner life, capacity, and efficiency. Table 6 shows the impact of low refrigerant charge on cooling capacity and COP for systems II and III at the lowest charge levels tested. Based on these results, it can be concluded that low refrigerant charge levels can cause significant reductions in both cooling capacity and energy efficiency. The reduction of energy efficiency is a worldwide issue and very much related to energy resource depletion. Also, running equipment with low or high refrigerant charge levels may shorten its lifespan.

Table 6 Capacity degradation of air-conditioner versus refrigerant charge level

	Accumulator	Refrigerant charge level reduction	Cooling capacity degradation	Energy Efficiency degradation
System II	yes	↓ 38 %	↓ 50 %	↓ 25 %
	no	↓ 25 %	↓ 25 %	↓ 26 %
System III	yes	↓ 40 %	↓ 46 %	↓ 30 %
	no	↓ 12 %	↓ 12 %	↓ 7 %

6. CONCLUSIONS

An extensive performance evaluation of the virtual refrigerant charge sensor was performed. Four different types of systems were used to consider the effects of variations in charge level, ambient conditions and air flow on the accuracy of the refrigerant charge sensor. The systems included both commercial and residential equipment with different types of compressors, with and without low-side accumulators, and using fixed orifice expansion devices and electronic expansion valves. It can be concluded that the virtual refrigerant charge level gauge has very good performance in terms of accuracy for both low and high charge when parameters are tuned using data. For systems not incorporating accumulators, default parameters lead to good performance, whereas tuning should be utilized for systems having accumulators (or better defaults should be determined for systems having accumulators). The virtual charge sensor could be easily implemented at low costs in terms of both hardware and software. The gauge 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

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NOMENCLATURE

k	slope of the best-fit line	
k_{ch}	condenser geometry constant	
k_{sc}	evaporator geometry constant	
k_{sh}	threshold for k	
$k_{sh/sc}$	empirical constant	
k_{th}	threshold for k	
m	actual total charge	(Ibm)
n	number of data points	
S	standard deviation	
T	temperature	(C)
τ	sampling time	(S)
$y_{m,m+1,\ldots,m+n-1}$	data points	

Subscripts

cond	condensing
evap	evaporating
hs	high side
hs,o	high side for zero-subcooling
li	Liquid Line
ls	Low side
ls,o	low side for zero-superheat
sc	subcooling
sc,rated	rated subcooling
sh	superheat
sh,rated	rated superheat
suc	suction line temperature
rated	nominal total

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