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ABSTRACT

The primary goal of the work described in this paper was to evaluate and extend a virtual refrigerant charge sensor (VRC) for determining refrigerant charge for equipment having variable-speed compressors and fans. To evaluate the accuracy of the VRC, data were first collected from previous laboratory tests for different systems and over a wide range of operating conditions. In addition, new laboratory tests were performed to consider conditions not available within the existing data set. The systems for the new laboratory tests were two residential ductless split heat pump systems that employ a variable-speed compressor and R-410a as the refrigerant. Based on the evaluations, the original virtual charge sensor (termed Model I) was found to work well in estimating the refrigerant charge for systems with a variable-speed compressor under many operating conditions. However, for extreme test conditions such as low outdoor temperatures and low compressor speed, the VRC needed to be improved. To overcome the limitations, the model associated with the VRC sensor was modified to include a term involving the inlet quality to the evaporator (termed model II). Both the model I and II showed good performance in terms of predicting charge levels for systems with a constant speed compressor, but model II gave better performance for systems with a variable-speed compressor. However, when the superheat of the compressor was zero, neither model I nor II could accurately predict charge level. Therefore, a third approach (Model III) was developed that includes the discharge superheat of the compressor. This model improved performance for a laboratory-tested system that included a number of points with no superheat entering the compressor.

1. INTRODUCTION

There have been laboratory studies that have documented the impact of refrigerant charge on the performance of air conditioning equipment, including work by Rice (1987), Moshen (1990), Breuker et al. (1998), and Goswami (2002). Recently, Kim and Braun (2012) found that a refrigerant charge reduction of 25% led to an average energy efficiency reduction of about 15% and capacity degradation of about 20%. These studies showed that improper refrigerant charge could significantly decrease energy efficiency and capacity and lead to operating conditions that decrease equipment lifespan. Furthermore, refrigerant charge leakage can contribute to global warming in the long term. The leakage of refrigerant released to the atmosphere contributes to the greenhouse effect. The other long-term impact is caused by the extra carbon dioxide emissions from fossil fuel power plants due to lower energy efficiency.

Packaged air conditioners are widely used in 46% of all commercial buildings, serving over 60% of the commercial building floor space in the U.S. (EIA, 2003). The survey data indicates that annual cooling energy consumption related to packaged air conditioner is about 160 trillion Btus. Therefore, small improvements in packaged air conditioner performance can lead to significant reductions in overall energy use and environmental impact. Based on a survey and analysis of 215 rooftop units on 75 buildings in California, it has been shown that 46% of the units were not properly charged, which resulted in reductions in capacity and energy efficiency. The average energy impact of refrigerant charge problems was about 5% of the annual cooling capacity. Based on research of more than 4,000 residential cooling systems in California, only 38 % have correct charge (Downey, 2002) and the data from Blasnik et al. (1996) have indicated that an undercharge of 15 % is common.

The typical approach used to verify refrigerant charge for systems having variable-speed compressors was reviewed. Despite the fact that there are slight differences between manufacturers, the basic methods are based on using measured pressure at the service valve determined with a manifold gauge, when system is operating at fixed speed in a test mode set by remote controller. A technician decides to add or remove refrigerant based on the difference between a pressure measurement and a target pressure specified by technical data provided by the manufacturer. These approaches can only determine whether the charge is high or low, not the level of charge. In addition, the current charge verification protocols utilize pressure gauges or transducers installed at the service valve. The installation of these gauges or transducers can lead to refrigerant leakage. Because of these limitations, the current protocols for checking refrigerant charge may be doing more harm than good in many situations.

The original VRC sensor (Li and Braun: 2007, 2009) uses a correlation in terms of superheat and subcooling that are determined using low-cost surface mounted temperature sensors. Parameters of the method can be estimated using readily available manufacturers' data. Furthermore, the charge estimates are relatively insensitive to the existence of other system faults. Based on previous research (Kim and Braun: 2010), the VRC sensor was found to work well in estimating the refrigerant charge for a range of air conditioner and heat pumps having fixed speed compressors. However, none of the tested equipment included variable-speed compressors. The current research extends the VRC sensor for systems with variable-speed compressors and fans. The VRC sensor is evaluated over a wide range of ambient conditions for both heating and cooling. In particular, the modified VRC sensor represents an improvement over the original method at extreme conditions such as low compressor speed, and low outdoor temperature conditions.

2. EXTENSION of VRC SENSOR

2.1 Original VRC sensor for fixed speed compressor

Li and Braun (2007) developed the VRC sensor (termed model I) for correlating refrigerant charge level in terms of superheat and superheating. Deviations from nominal charge can be obtained by using four measurements and four parameters. The charge deviation relative to the rated charge is expressed as

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

where m_{total} is the actual total charge, m_{rated} is the nominal total refrigerant charge, $K_{sh/sc}$ and K_{ch} are two constants that are 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.

$$K_{ch} = \frac{m_{total,rated}}{K_{sc}} = \frac{T_{sc,rated}}{\left(1-\alpha_o\right)\cdot X_{hs,rated}} \tag{2}$$

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

$$m_{total,o} = \alpha_o \cdot m_{total,rated} \tag{4}$$

$$m_{hs,rated} = X_{hs,rated} \cdot m_{total,rated} \tag{5}$$

where $m_{total,o}$ is the total refrigerant charge of the system when subcooling and superheat are zero. $X_{hs,rated}$ 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 the current paper, three different approaches are considered for determining the empirical parameters: default parameters, simulation parameters, and tuned parameters. Based on data available from Harms (2002), Li and Braun (2009) found that a reasonable estimate for $X_{hs,rated}$ is 0.73 whereas a value of 0.75 was determined for α_o as a default parameter. A reasonable estimate for $K_{sh/sc}$ for systems using a thermal expansion valve (TXV) or fixed orifice

(FXO) as the expansion device is 1/2.5 based on previous test results. For a system using an electronic expansion valve (EEV), superheat remains constant regardless of charge, and refrigerant inventory in the evaporator is relatively constant. In this case, a reasonable estimate for $K_{sh/sc}$ is 0.

To improve charge predictions, a simulation method for estimating K_{ch} was developed for extremely over and under refrigerant charge level (Kim and Braun: 2010). The simulation method determines this empirical parameter using a physical description of the heat exchangers and piping along with measurements at rated conditions and the nominal charge level. 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. Based on those findings, K_{ch} can be calculated from the refrigerant mass distribution in the system. For these calculations, the void fraction correlations based on the slip ratio correlated equation from Zivi (1964) was found to give the best results.

Alternatively, the empirical parameters within the VRC sensor algorithm can be tuned to improve accuracy if data are available at different refrigerant charge levels and operating conditions. The parameter tuning method minimizes the errors between predicted and known refrigerant charge by using linear regression techniques. In the current study, linear regression techniques were 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.

2.2 Modified VRC sensor for variable-speed compressors

Based on previous researches (Li and Braun: 2009, Kim and Braun: 2010), the VRC sensor for equipment with fixed speed compressor worked well with tuned parameters, unless the system was extremely over or undercharged. To extend the VRC sensor for equipment with variable-speed compressors, modified VRC sensor (term model II) were developed in this research. For model II, a correlation for refrigerant charge in term of evaporator inlet quality was added to the model I.

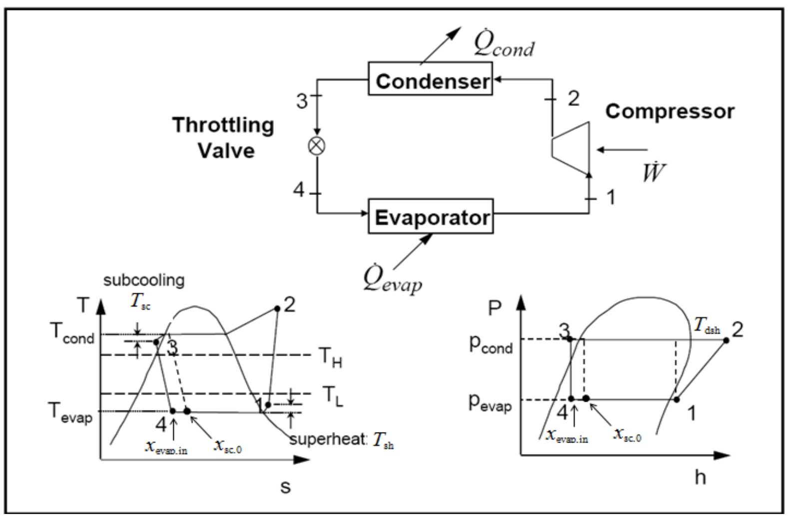

Figure 1 Operating states of vapor compression cycle.

Figure 1 shows the operating states of vapor compression cycle. The high-side refrigerant charge is related to subcooling using

$$m_{hs} = K_{sc} \cdot T_{sc} + m_{hs,o} \tag{6}$$

where m_{hs} is the charge in the high-pressure side of the system, $m_{hs,0}$ is the high-side refrigerant mass for the case of zero subcooling 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.

The low-side charge is related to superheat and inlet quality for the evaporator based on

$$m_{ls} = m_{ls,o} - K_{sh} \cdot T_{sh} + K_X \cdot (x_{sc,o} - x_{evap,in}) \tag{7}$$

where $m_{ls,o}$ is low-side refrigerant mass for the case of zero superheat and zero subcooling, $x_{evap,in}$ is evaporator inlet quality, K_x is a constant characteristic of a given system, 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. The subscript 'sc,o' denotes that the case of zero subcooling is employed.

Equations 6 and 7 can be applied to all operating condition including the rated condition so that

$$m_{hs,rated} = K_{sc} \cdot T_{sc,rated} + m_{hs,o} \tag{8}$$

$$m_{ls,rated} = m_{ls,o} - K_{sh} \cdot T_{sh,rated} + K_X \cdot (x_{sc,o} - x_{evap,in,rated}) \tag{9}$$

where the subscript '*rated*' denotes that the rating operating conditions are employed. $T_{sc,rated}$ and $T_{sh,rated}$ are liquid line subcooling and suction line superheat at rated conditions with the nominal charge, respectively.

Equations 6 to 9 can be combined to give expressions for changes in subcooling, superheat, and inlet quality of evaporator from rated conditions in terms of charge variation by elimniating $m_{hs,o}$ and $m_{ls,o}$.

$$K_{sc} \cdot (T_{sc} - T_{sc,rated}) = (m_{hs} - m_{hs,rated}) \tag{10}$$

$$-K_{sh} \cdot (T_{sh} - T_{sh,rated}) + K_x \cdot (x_{evap,in} - x_{evap,in,rated}) = (m_{ls} - m_{ls,rated}) \tag{11}$$

Equations 10 and 11 are combined and the result manipulated to give a single expression that relates subcooling, inlet quality of evaporator, and superheat to the total refrigerant charge.

$$(m_{total} - m_{total,rated}) = K_{sc} \cdot (T_{sc} - T_{sc,rated}) - K_{sh} \cdot (T_{sh} - T_{sh,rated}) + K_x \cdot (x_{evap,in} - x_{evap,in,rated}) \tag{12}$$

Equation 12 is then manipulated to give charge deviation from rated charge relative to rated charge as a function of three inputs determined from measurements (T_{sc}, T_{sh}, $x_{evap,in}$) and seven constants (K_{sc}, $K_{sh/sc}$, $K_{x/sc}$, $m_{total,rated}$, $T_{sc,rated}$, $T_{sh,rated}$, $x_{evap,in,rated}$).

$$\frac{(m_{total} - m_{total,rated})}{m_{total,rated}} = \frac{1}{K_{ch}} \left\{ (T_{sc} - T_{sc,rated}) - K_{sh/sc}(T_{sh} - T_{sh,rated}) + K_{x/sc}(x_{evap,in} - x_{evap,in,rated}) \right\} \tag{13}$$

where $K_{sh/sc}$, $K_{x/sc}$ and K_{ch} are three empirical constants that must be estimated for a given system and the other four parameters are directly determined from measurements at the rated condition and nominal charge level. The quality entering the evaporator can be readily estimated using measurements exiting the condenser and assuming an isenthalpic expansion process,

In Equations 13, it is necessary to know $K_{x/sc}$. Default estimates of this empirical parameter can be determined from data at the rated condition. Using equations 4 and 5 along with equation 14 with the case of zero superheat and zero subcooling, an expression for estimating $K_{x/sc}$ is determined using the following steps,

$$(m_{total,rated} - m_{total,o}) = K_{sc} \cdot T_{sc,rated} - K_{sh} \cdot T_{sh,rated} + K_x \cdot (x_{evap,in} - x_{evap,in,rated}) \tag{14}$$

$$\frac{(m_{total,rated} \cdot (1 - \alpha_o))}{K_{sc}} = T_{sc,rated} - \frac{K_{sh}}{K_{sc}} \cdot T_{sh,rated} + \frac{K_x}{K_{sc}} \cdot (x_{evap,in} - x_{evap,in,rated}) \tag{15}$$

$$\frac{T_{sc,rated}}{X_{hs,rated}} = T_{sc,rated} - \frac{K_{sh}}{K_{sc}} \cdot T_{sh,rated} + \frac{K_x}{K_{sc}} \cdot (x_{evap,in} - x_{evap,in,rated}) \tag{16}$$

$$K_{x/sc} = \frac{(1 - X_{hs,rated})}{X_{hs,rated}} \cdot \frac{T_{sc,rated}}{(x_{hs,o} - x_{evap,in,rated})} + K_{sh/sc} \frac{T_{sh,rated}}{(x_{hs,o} - x_{evap,in,rated})} \tag{17}$$

Under low compressor speed conditions with low ambient temperature, the laboratory test results for this study had zero subcooling and superheat. In these cases, neither the model I or model II approaches, which use subcooling and superheat measurement as input parameters, can accurately predict the charge level. Therefore, a model III approach was developed to provide improved performance in these situations. Model III is a modification of the model II equation that includes a correlation for refrigerant charge in term of discharge superheat of compressor.

$$\frac{\left(m_{total}-m_{total,rated}\right)}{m_{total,rated}}=\frac{1}{K_{ch}}\left\{\left(T_{sc}-T_{sc,rated}\right)-K_{sh/sc}\left(T_{sh}-T_{sh,rated}\right)+K_{x/sc}\left(x_{evap,in}-x_{evap,in,rated}\right)+K_{dsh/sc}\left(T_{dsh}-T_{dsh,rated}\right)\right\} \quad (17)$$

where $K_{dsh/sc}$ is a constant characteristic of a given system, and $T_{dsh,rated}$ is discharge superheat of the compressor at rated conditions with the nominal charge.

3. EVALUATION of VRC SENSOR for COOLING EQUIPMENT

3.1 System descriptions and test conditions

To evaluate the VRC sensor, data for air conditioners and chillers with variable-speed compressors were collected where the effects of refrigerant charge on performance were considered. Table 1 gives specifications for three units where data were obtained by laboratory testing. This includes data for two water-to-water chiller units and a conventional split air conditioner unit. R-22 and R-410A were used as the refrigerant and scroll and reciprocating type compressors and electronic expansion valves (EEV) were employed. The system test conditions considered for the systems are listed in Table 2. The test data were all obtained at different compressor speeds. Refrigerant charge levels were varied between about 70 % and 130 % of nominal charge levels. However, most of the tests were performed at a single indoor and ambient temperature.

Table 1 System descriptions for existing refrigerant charge level test data

System		Capacity (kW)	Refrigerant	Compressor	Expansion device	System
I	Choi (2001)	3.5	R-22	Scroll	EEV	Water to Water
II	Kim (2003)	3	R-22	Reciprocating	EEV	Water to Water
III	Cho (2005)	7.2	R-410A	Scroll	EEV	Air Split Type

Table 2 Test conditions for cooling equipment having a variable-speed compressor

System		Indoor Water / Air Inlet Temperature	Indoor Air Wet Temperature	Outdoor Water/ Air Inlet Temperature	Comp Speed	Refrigerant Charge
		C	C	C	Hz	%
I	Choi (2001)	25	-	30, 34, 38, 42	30 ~ 60	80 ~ 120
II	Kim (2003)	26.7	-	35	20 ~ 60	70 ~ 120
III	Cho (2005)	27	19.51	35	40 ~ 60	70 ~ 130

3.2 Evaluation of VRC sensor for cooling

The VRC sensor was evaluated in terms of RMS deviation between predicted and actual charge levels relative to nominal charges for the cooling systems having variable-speed compressors. The test data did not provide information necessary to estimate parameters using the simulation approach. Therefore, models I and II were only evaluated based on the use of default and tuned parameters. Figures 2 to 5 show the performance of the VRC sensor model I and II based on the default and tuned parameters.

Figures 2 shows the performance of the VRC sensor based on model I with default parameters. Overall, the RMS errors of the VRC sensor algorithm for model I were 8% based on default parameters. In many cases, the accuracy of the refrigerant charge predictions is good when using default parameters. However, the use of the default parameters led to some significant errors greater than 10% in refrigerant charge estimates at both low and high charge levels with low compressor frequencies. When the VRC sensor model II with default parameters was applied, there was an improvement compared to using model I with RMS errors of 6 % in figure 3. Model II with default parameters can also lead to significant improvements in cases where model I does not work well, such as at extremely low outdoor temperatures and high charge level. However, there were still some points with significant refrigerant charge estimate errors at high charge level with low compressor frequencies.

To increase the accuracy of the VRC sensor, the parameters were tuned for each specific system based on measurements obtained at different refrigerant charge levels. When tuned parameters were applied to the model I and II, the VRC sensor showed better performance than when the default parameters were applied, as shown in Figure 4 and 5. The RMS errors were reduced to 4 % for model I and 3 % for model II. The results verified that tuned parameters significantly improve the accuracy of the VRC sensor. It can be seen that when the system is not over charged, model I with tuned parameters has good performance under various compressor speeds. However, when the system is extremely over charged, model I may have significant errors. Compared to model I, model II led to some improvements in cases where model I did not work well, such as low compressor speed and high charge level. Overall, the VRC sensor using model II with tuned parameters can provide very accurate estiamtes of refrigerant charge levels for cooling systems having variable-speed compressors.

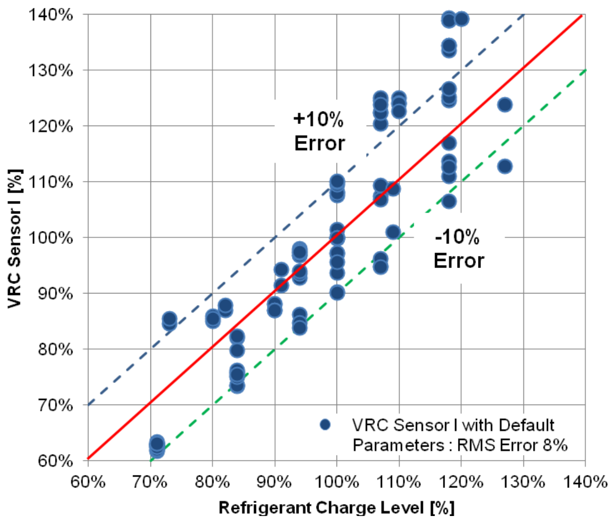

Figure 2 Performance of VRC sensor model I based on default parameters for cooling

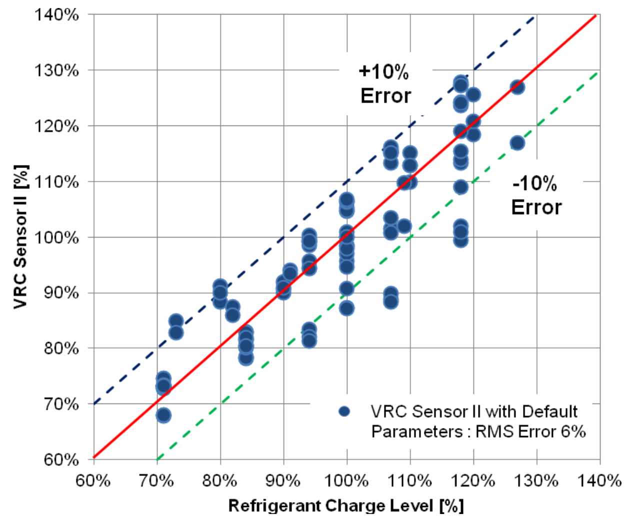

Figure 3 Performance of VRC sensor model II based on default parameters for cooling

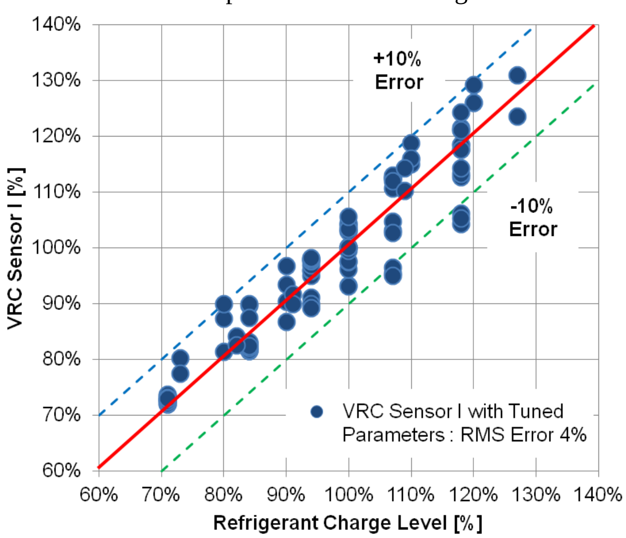

Figure 4 Performance of VRC sensor model I based on tuned parameters for cooling

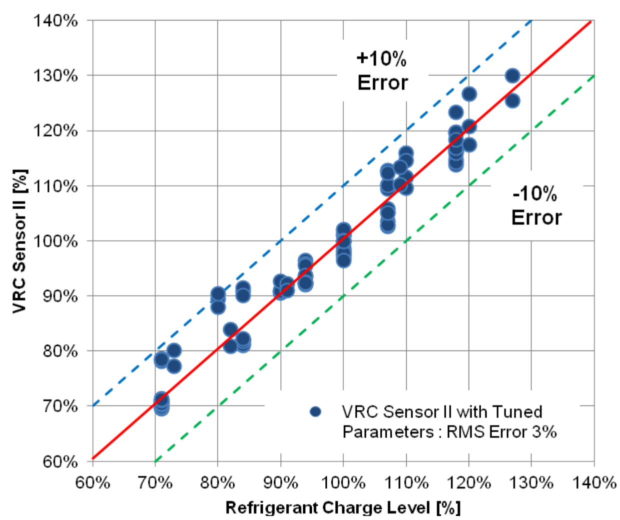

Figure 5 Performance of VRC sensor model II based on tuned parameters for cooling

4. EVALUATION of VRC SENSOR for HEAT PUMPS

4.1 System description and test conditions

The primary limitations of the previously available data for systems with variable-speed compressors are that the test conditions were limited to 1) cooling mode only, 2) with 70% as the lowest refrigerant charge level, 3) one ambient temperature condition, and 4) for systems that do not incorporate multi-speed fans. To better assess the accuracy and broaden the application of the VRC sensor, new test plans were established to consider the following key issues: 1) heating mode operation for heat pumps, 2) various ambient temperature conditions, 4) lower levels of refrigerant charge, and 5) systems with multi speed fans. Two heat pump systems having a variable-speed compressor and multi-speed fans were selected for testing and installed within the psychrometric chambers at Herrick Laboratories. Tables 3 provide an overview of the two systems that were tested. R-410A was used as the refrigerant for both systems and a hermetic type compressor and EEV were employed.

The ranges of test conditions in cooling and heating mode are given in Tables 17. The test matrix was designed to consider both cooling and heating mode operation with low levels of refrigerant charge and low ambient temperatures. Data for relatively low ambient temperatures in cooling were necessary to test the validity of the algorithm during off-season when regular maintenance procedures are often performed. Refrigerant charge levels were varied between 50% and 130% of nominal charge levels with outdoor temperatures between about 67 F and 110 F for cooling mode and 17 F and 47 F for heating mode. The effects of reduced indoor air flow were also considered for system III, with airflow rates from 260 to 430 [CFM] for heating and cooling mode. The variable compressor speeds were considered from 18 to 65 Hz in cooling mode and from 18 to 130 Hz in heating mode.

Table 3 System description for heat pump systems having a variable-speed compressor

System	Size (kW)	Refrigerant Type	Compressor	Expansion Device	Accumulator	Assembling Type
III	3.5	R-410A	Hermetic type	EEV	Yes	Split
IV	3.5	R-410A	Hermetic type	EEV	Yes	Split

Table 4 Test conditions for heat pump systems having a variable-speed compressor

<table>
<tr><th rowspan="3">System</th><th rowspan="3">Mode</th><th colspan="2">Indoor Temp.</th><th>Outdoor Temp.</th><th rowspan="2">Compressor Speed</th><th rowspan="2">Indoor
Fan Speed</th><th rowspan="2">Refrigerant
Charge Level</th></tr>
<tr><th>Dry</th><th>Wet</th><th>Dry</th></tr>
<tr><th>(F)</th><th>(F)</th><th>(F)</th><th>[Hz]</th><th>[CFM]</th><th>(%)</th></tr>
<tr><td rowspan="2">III</td><td>Cooling</td><td>80</td><td>67</td><td>110 / 95 / 67</td><td>21 ~ 65</td><td>430 / 260</td><td>50 ~ 130</td></tr>
<tr><td>Heating</td><td>70</td><td>-</td><td>47 / 37 / 17</td><td>21 ~ 130</td><td>430 /260</td><td>50 ~ 130</td></tr>
<tr><td rowspan="2">IV</td><td>Cooling</td><td>80</td><td>67</td><td>105 / 95 / 67</td><td>18 ~ 49</td><td>410</td><td>50 ~ 130</td></tr>
<tr><td>Heating</td><td>70</td><td>-</td><td>47 / 27 / 17</td><td>18 ~105</td><td>410</td><td>50 ~ 150</td></tr>
</table>

4.2 Evaluation of VRC sensor for heat pump systems

Figures 6 to 9 show the accuracy of the VRC sensor for the heat pump systems in cooling and heating mode. The performance was evaluated in terms of RMS deviation from the actual charge levels presented on a percentage basis for models II and III.

Figure 6 shows performance of the VRC sensor based on model II and default parameters in cooling and heating mode. Based on the RMS errors of 16 % for cooling mode and 22 % for heating mode, the VRC sensor did not perform well in predicting the charge level. As the fault level of refrigerant charge increased or decreased, there was bigger difference between estimated and actual charge amounts. For example, the model II with default parameters predicts 20 % undercharge when the system is charged at 50% of nominal charge. When the ambient temperature and compressor speed were low, the refrigerant charge error increased compared to other test conditions.

Figure 7 shows results based on the use of parameters that were determined using the simulation approach. The model II based on simulation parameters showed RMS errors of 15 % for cooling mode and 20 % for heating mode. The use of simulation parameters led to significant errors in refrigerant charge estimates at low and high charge level. The errors were relatively large at low compressor speed conditions at overcharge conditions. Overall, model

II with simulation parameters did not improve the performance of the VRC sensor compared to the default parameters for the heat pump system.

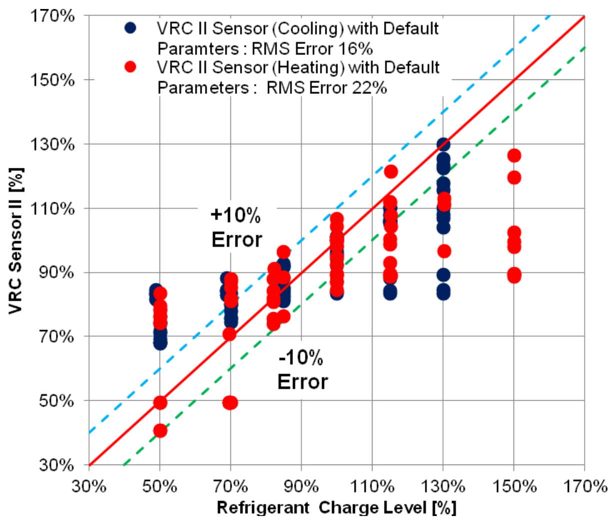

Figure 6 Performance of VRC sensor model II based on default parameters for heat pumps

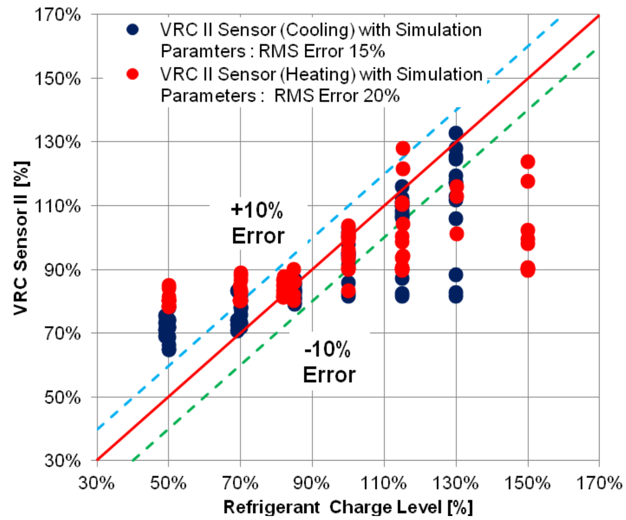

Figure 7 Performance of VRC sensor model II based on simulation parameters for heat pumps

Figure 8 shows performance based on tuned parameters. The RMS errors were reduced to 13 % for cooling mode and 12 % for heating mode. When tuned parameters were applied in heating mode, there was a significant improvement compared to using the default and simulation parameters. Although the RMS error is reduced, the errors at high charge levels are greater with more variability in the predictions. The errors were still large at high charge levels because the superheat exiting the compressor was nearly zero for various operating conditions. The VRC sensor II with tuned parameters underestimates charge when the system is highly overcharged with errors up to 30% at a charge level of 130 %. In cooling mode, the large deviations still remained at conditions having zero subcooling.

Figure 9 shows performance of model III based on tuned parameters. In this case, the RMS errors were reduced to 10% for cooling and 7 % for heating mode. In heating mode, model III can lead to significant improvements in cases where models I and II do not work well, such as at overcharge conditions with extremely low outdoor temperatures and low speed compressor. Overall, the VRC model III is better than the other two models for characterizing refrigerant charge levels for heat pumps with variable-speed compressor. However, there were still some significant errors (over 5%) at low ambient and low speeds when subcooling was zero.

5. COMPARISONS WITH MANUFACTURERS' CHARGING METHOD

The charging method specified by the manufacturer for system IV was applied and compared with the VRC sensor based on model III for cooling mode. The approach used to verify refrigerant charge in the field involves the use of pressure at the service valve. Suction pressure for cooling mode and discharge pressure for heating mode are used to indicate the charge level with the compressor operating at a fixed speed in a test mode. The technicians can evaluate whether to add or remove refrigerant based on a difference between the pressure measurement and a target pressure.

Figure 10 shows measurements associated with applying the company refrigerant charge protocol for system IV in cooling mode at three different ambient temperatures. The three horizontal lines correspond to the target suction pressures at the three temperatures. Although the suction pressure increases with charge level, it doesn't achieve the target even at 130% of normal charge. The deviation between the measured and target pressure is greatest at the lowest outdoor temperature. It appears that current approaches would have difficulty in identifying the proper charge amount during off-season maintenance. Figure 11 shows performance of model IV based on tuned parameters using the data at maximum compressor speed in cooling mode. The VRC sensor provides accurate refrigerant charge estimates in cooling mode regardless of the ambient temperature.

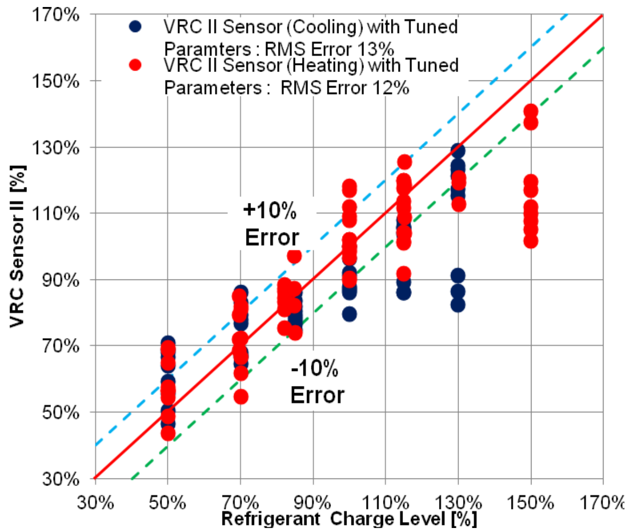

Figure 8 Performance of VRC sensor model II based on tuned parameters for heat pumps

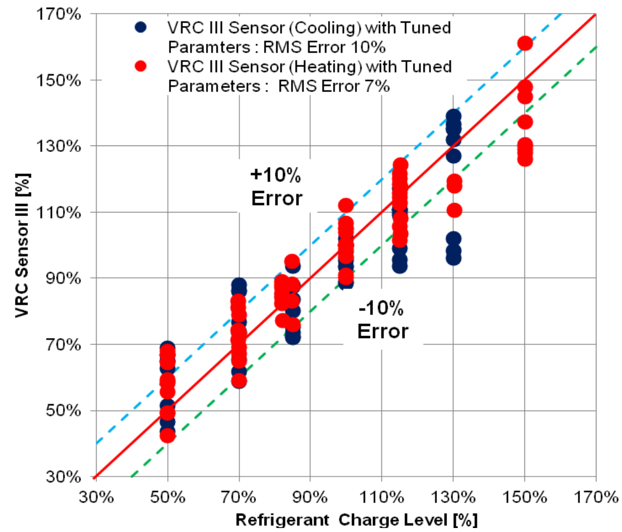

Figure 9 Performance of VRC sensor model III based on tuned parameters for heat pumps

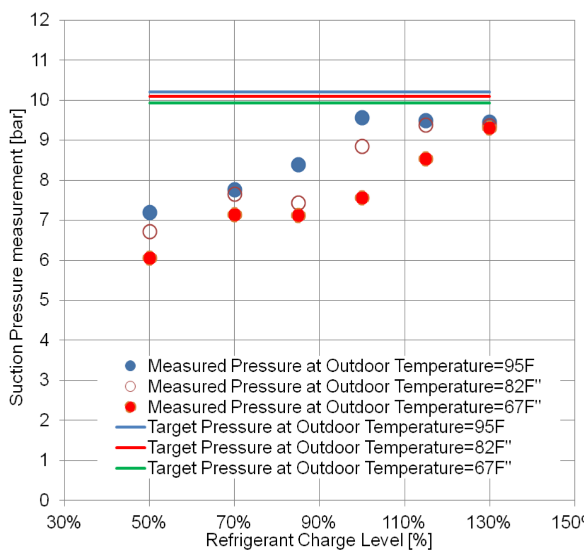

Figure 10 Refrigerant charge method based on company method for cooling mode (System IV)

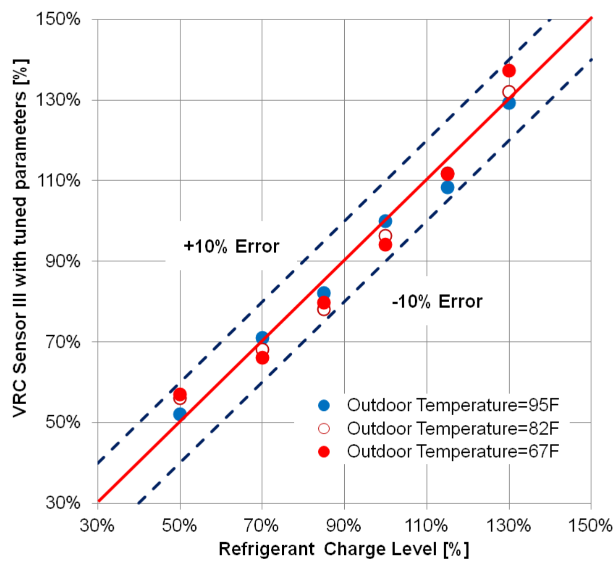

Figure 11 Performance of VRC sensor model III based on tuned parameters for the cooling mode (System IV)

6. CONCLUSIONS

The original VRC sensor (termed model I) using tuned parameters worked well for different systems at many operating conditions but the performance was significantly worse for low compressor speed and at low ambient temperatures in both cooling and heating mode. Improved performance was achieved with a modification that accounts for variations in the quality of refrigerant entering the evaporator (termed model II) but tended to fail under conditions with zero condenser subcooling and evaporator superheat for variable-speed heat pumps. Better performance was achieved for those conditions when compressor discharge superheat was included (termed model III).

For cooling equipment with variable-speed compressors, the model I and II approaches were evaluated based on the use of default and tuned parameters. The RMS errors based on default parameters were 8% and 6% for model I and model II, respectively. When tuned parameters were used, the RMS errors were 4% and 3% for model I and model II, respectively. The model II showed better performance when the charge levels were small and large.

For the laboratory testing results from heat pump systems with both variable-speed compressors and fans, the model I and II methods did not work well so the method was extended to include an additional input (model III). When the model III algorithm was tuned using all available data, the overall RMS errors were 10% for cooling mode and 7% for heating mode, compared to over 10% for both cooling and heating mode when model I and II were used. The cases where the VRC sensor with model III had difficulty were when the system was operated with zero subcooling at low compressor speed.

The VRC 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. Continuous or frequent monitoring of charge level should lead to early detection of refrigerant leakage and avoidance of under or overcharging. 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. The current charge protocols that are based on low pressure can only indicate whether refrigerant charge is high or low, whereas the VRC sensor provides a measure of the quantity of charge. The technician in the field could easily use the tool to determine the correct amount of charge to add to the unit.

NOMENCLATURE

EEV	Electronic expansion valve	(-)		**Subscripts**
FXO	Fixed orifice	(-)	*dsh*	Discharge superheat of compressor
$K_{dsh/sc}$	Constant characteristic of a given system related to discharge superheat of compressor	(-)	*dsh,rated*	Discharge superheat of compressor at rated condition
K_{ch}	Empirical constant	(-)	*evap,in*	Inlet of evaporator
K_{sc}	Constant related to condenser subcooling and depending on the condenser geometry	(-)	*hs*	High side
K_{sh}	Constant related to evaporator superheat and depending on the evaporator geometry	(-)	*hs,o*	High side for zero-subcooling
$K_{sh/sc}$	Empirical constant	(-)	*ls*	Low side
$K_{x/sc}$	Constant characteristic of a given system related to inlet quality of evaporator	(-)	*ls,o*	Low side for zero-superheat
m	Refrigerant charge	(-)	*rated*	Rating operating conditions
m_{total}	Total refrigerant charge	(kg)	*sc*	Subcooling
$m_{total,rated}$	Total refrigerant charge at rated condition	(kg)	*sc,rated*	Rated subcooling
T	Temperature	(kg)	*sh*	Superheat
T_{sc}	Liquid line subcooling	(C)	*sh,rated*	Rated superheat
$T_{sc,rated}$	Liquid line subcooling at rated condition	(C)	*tot*	Total
T_{sh}	Evaporator superheat	(C)	*tot,o*	Total for zero-subcooling and zero-superheat
$T_{sh,rated}$	Evaporator superheat at rated condition	(C)		**Greek**
TXV	Thermostatic expansion valve	(C)	α_o	Ratio of refrigerant charge necessary to have saturated liquid existing the condenser at rating conditions to the rated refrigerant charge
$X_{hs,rated}$	Ratio of high side charge to the total	(-)		

refrigerant charge at rating conditions

x	Refrigerant quality	(-)

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