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Virtual Refrigerant Mass Flow and Power Sensors for Variable-Speed Compressors

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ABSTRACT

The use of variable-speed compressors in heat pumps and air conditioners has increased in recent years in order to improve comfort and energy efficiency. At the same time, there is a trend towards embedding more sensors in this type of equipment to facilitate real-time energy monitoring and diagnostics. Although compressor mass flow rate and power consumption are useful indices for performance monitoring and diagnostics, they are expensive to measure. The virtual variable-speed compressor sensor (VVC) was developed to estimate mass flow rate and power consumption using inexpensive temperature sensors and embedded models. The mass flow and power consumption at maximum compressor speed and rated superheat are first correlated with suction and discharge pressure. These pressures are estimated using virtual pressure sensors that use condensation and evaporation temperatures measured on return bends within the heat exchangers. The mass flow and power predictions are corrected for different speeds and inlet superheat heats using additional correlations. The VVC can predict mass flow rate and power consumption within RMS errors of ± 5% and ± 3%, respectively, under normal (no fault) conditions. The VVC also works well under conditions where faults in the system exist, since the VVC isolates these two estimates from other possible external faults such as condenser or evaporator fouling. One of the inputs to the model is compressor frequency, which can be difficult to measure for technicians in the field. However, for an embedded application, the compressor frequency would generally be available from the motor controller. Alternatively, the VVC could be used to estimate compressor frequency if mass flow rate or power consumption was provided as an input from another independent virtual sensor.

1. INTRODUCTION

Variable-speed compressors are one of the most energy efficient methods to regulate capacity for heat pumps and air conditioners because they minimize on/off cycling (Qureshi et al., 1996; Aprea et al., 2004; Ding, 2007). The fixed-speed compressor, the dominant type in the air conditioner market in the past, is now gradually replaced by the variable-speed compressor for high efficiency equipment. The use of variable-speed compressors is common in Asia and has recently started to make in-roads within the U.S. market for commercial and residential air conditioner.

Despite the expanded application of variable-speed compressors, there are only a few modeling approaches for compressor performance that have been published (Navarro-Esbrv et al., 1981; Park et al., 2001; Browne et al., 2002). Manufacturers typically provide several map-based models at different frequencies for variable-speed compressors. The compressor performance for other operating frequencies can be calculated using interpolation and extrapolation. However, this type of modeling approach may not work well over a wide range of operating conditions. To overcome this limitation, empirical functional equations were developed and validated that provide accurate estimates of mass flow rate and power consumption for variable-speed compressors. These models could use inexpensive temperature measurements as inputs and be embedded in a performance monitoring and diagnostic system as virtual sensors.

The VVC sensor consists of virtual mass flow rate (VRMF) and virtual compressor power (VCP) sensors can used for real-time monitoring of capacity and efficiency. The VVC sensor can be used in place of expensive mass flow and power meters. The VVC sensor models are based on second-order functions in terms of condensation and evaporation temperature (determined with low cost temperature sensors installed on return bends) and operating frequency. For embedded systems, operating frequency would be available from the motor controller. For existing equipment, it is difficult to measure operating frequency in the field for hermetic or semi-hermetic compressors. In this case, compressor frequency could be estimated using mass flow rate as input explained in Kim and Braun (2012).

2. VVC SENSOR MODELING

2.1 System specification and test condition

Table 1 provides an overview of systems where data was available to evaluate the performance of VVC sensor. Three different ductless split heat pump units and one water-to-water system were considered. All of systems employed electronic expansion valves (EEV) as expansion devices. A hermetic rotary type of variable-speed compressor and R-410a or R-22 refrigerant were employed. The split heat pump units had low-side accumulators.

The ranges of test conditions in cooling and heating mode are given in Table 2. Laboratory data (Nyika, 2011) were obtained for different refrigerant charge levels except for system III. Refrigerant charge levels were varied between 50% and 130% of nominal charge levels. The test data were obtained at with variations in ambient temperature. The ambient temperatures ranged between about 60 and 115F for cooling mode, and 17 and 68F for heating mode. The indoor temperature was kept at 80F for cooling and 70F for heating mode. The compressor speeds were considered from 18 to 70Hz in cooling mode and from 20 to 115Hz in heating mode. Tests for system IV were performed at different condenser and evaporator water mass flow rates to simulate fouling fault conditions. Tests for system IV (Kim 2003) also included simulation of compressor valve leakage faults that reduced refrigerant mass flow rate relative to normal compressor operation.

Table 1 System description for system with variable speed compressor

System	Size (kW)	Refrigerant Type	Compressor	Expansion Device	Accumulator	AssemblyType
I	3.5	R-410a	Hermetic rotary type	EEV	Yes	Split
II	3.5	R-410a			Yes	Split
III	3.5	R-410a			Yes	Split
IV	3	R-22			No	Water-to-Water

Table 2 Testing conditions for system with variable speed compressor

<table>
<tr><th rowspan="3">System</th><th rowspan="3">Mode</th><th colspan="2">Indoor Temp.</th><th>Ambient Temp.</th><th rowspan="2">Comp Speed</th><th rowspan="2">Indoor air flow rate</th><th rowspan="2">Outdoor air flow rate</th><th rowspan="2">Percent of normal refrigerant mass flow rate</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>%</th><th>%</th><th>%</th><th>%</th></tr>
<tr><td rowspan="2">I</td><td>Cooling</td><td>80</td><td>67/Dry</td><td>67 ~ 105</td><td>18~70</td><td rowspan="2">60~100</td><td rowspan="2">100</td><td rowspan="2">100</td><td rowspan="2">50~130</td></tr>
<tr><td>Heating</td><td>70</td><td>-</td><td>17 ~ 62</td><td>20~115</td></tr>
<tr><td rowspan="2">II</td><td>Cooling</td><td>80</td><td>67</td><td>67 ~ 105</td><td>17~55</td><td rowspan="2">55~100</td><td rowspan="2">100</td><td rowspan="2">100</td><td rowspan="2">50~130</td></tr>
<tr><td>Heating</td><td>70</td><td>-</td><td>17 ~ 62</td><td>17~110</td></tr>
<tr><td rowspan="2">III</td><td>Cooling</td><td>80</td><td>67/Dry</td><td>67 ~ 115</td><td>43~70</td><td rowspan="2">60~100</td><td rowspan="2">100</td><td rowspan="2">100</td><td rowspan="2">100</td></tr>
<tr><td>Heating</td><td>70</td><td>-</td><td>7 ~ 68</td><td>40~95</td></tr>
<tr><td>IV</td><td>Cooling</td><td>80</td><td>-</td><td>60/75/90/105</td><td>20~55</td><td>35~100</td><td>25~100</td><td>65~100</td><td>70~130</td></tr>
</table>

2.2 Refrigerant volumetric flow rate and power input at rated frequency

The VVC sensor adopts a two-step model where the maximum volumetric flow and power are correlated in terms of the evaporation and condensing temperature at the rated compressor speed and then are corrected for the actual operating speed. The rated volumetric flow rate and power are determined using the second-order polynomial equations given in equation 1 and 2.

$$\frac{\dot{m}_{rated}}{\rho_{suc,rated}} = \dot{V}_{rated} = \left(c_1 + c_2 T_e + c_3 T_c + +c_4 T_e^{\ 2} + c_5 T_c^{\ 2} + c_6 T_e \cdot T_c\right) \tag{1}$$

$$\dot{W}_{rated} = \left(d_1 + d_2 T_e + d_3 T_c + d_4 T_e^{\ 2} + d_5 T_c^{\ 2} + d_6 T_e \cdot T_c\right) \tag{2}$$

where V_{rated}, m_{rated}, $\rho_{suc,rated}$, W_{rated} are the compressor volumetric flow rate, mass flow rate, and suction density, and power input at the rated frequency and superheat; the c's and d's are empirical coefficients; T_e is evaporating saturation temperature; and T_c is condensing saturation temperature. The condensing and evaporating temperatures can be measured on return bends within the heat exchangers. The flow rate is expressed as volumetric flow because it can be readily converted to a mass flow rate using the suction density regardless of the inlet superheat even though the testing was performed at a fixed superheat. The compressor power consumption model in equation 2 has reasonable accuracy without correction of suction density.

2.3 Correction modeling at different frequencies

Correction factors for converting the rated volumetric flow rate and power consumption to volumetric flow rate and power consumption at any operating speed and superheat are defined in equations 3 and 4.

$$\frac{m_{measured}}{\left(\dot{V}_{rated} \cdot \rho_{measured}\right)} = K_{flow} \tag{3}$$

$$\frac{\dot{W}_{measured}}{\dot{W}_{rated}} = K_{power} \tag{4}$$

where $m_{measured}$, $W_{measured}$ and $\rho_{suc,measured}$ are the compressor mass flow rate, input power, and suction density determined from measurements at any operating frequency and inlet superheat.

Figures 1 and 2 show trends in K_{flow} and K_{power} in terms of evaporation temperature and compressor frequency for system II with different condensing temperatures. K_{flow} and K_{power} are relatively independent of evaporating and condensing temperature and depend primarily on compressor frequency. An empirical model for this correction factors are expressed with a second-order function of frequency as shown in equations 5 and 6. The coefficients can be estimated based on regression analysis using experimental data at different frequencies.

$$K_{flow} = \left(a_1 \left(f - f_{rated}\right)^2 + a_2 \left(f - f_{rated}\right) + a_3\right) \tag{5}$$

$$K_{power} = \left(b_1 \left(f - f_{rated}\right)^2 + b_2 \left(f - f_{rated}\right) + b_3\right) \tag{6}$$

where the a's and b's are empirical coefficients, f is operating frequency, and f_{rated} is the rated compressor frequency.

The VVC sensors for estimating refrigerant mass flow rate and compressor input power use the products of the rated outputs from equations 1 and 2 and the correction factors from equations 5 and 6 with the result presented in equations 7 and 8. The accuracy of the mass flow rate prediction is less under conditions where the compressor inlet superheat is zero, corresponding to a two-phase mixture. However, other virtual mass flow rate sensors (Kim and Braun, 2012) can be applied under these conditions.

$$\dot{m}_{map} = \left(a_1 \left(f - f_{rated}\right)^2 + a_2 \left(f - f_{rated}\right) + a_3\right) \cdot \left(\dot{V}_{rated} \cdot \rho_{measured}\right) \tag{7}$$

$$\dot{W}_{map} = \left(a_1 \left(f - f_{rated}\right)^2 + a_2 \left(f - f_{rated}\right) + a_3\right) \cdot \left(\dot{W}_{rated}\right) \tag{8}$$

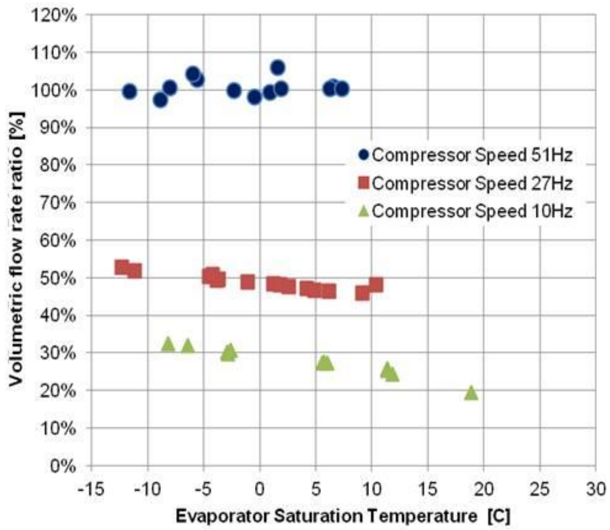

Figure 1: K_{flow} in terms of evaporation temperature for different frequencies and ambient (condensing) temperatures

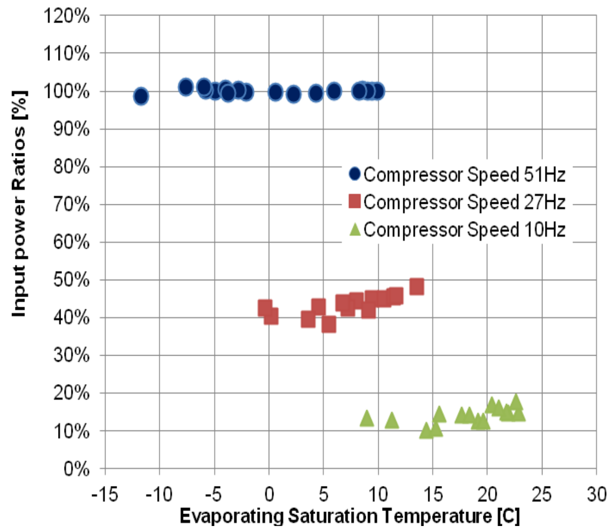

Figure 2: K_{input} in terms of evaporation temperature for different frequencies and ambient (condensing) temperatures

3. EVLAUATION of VVC SENSOR

3.1 Refrigerant mass flow rate

Figure 3 shows performance of the VVC sensor for refrigerant mass flow rate (VRMF) under no fault conditions over a range of operating frequencies and other conditions for the four heat pumps in both heating and cooling mode. The average relative errors of VRMF sensor were less than 6% for the four systems operating without faults.

Figure 4 shows performance of the VRMF sensor under various faults conditions for three of the systems. For this case, the accuracy is still within 6% except for the compressor valve leakage fault. With the compressor valve leakage faults, the overall RMS error for mass flow rate estimation was 16%. As the severity of the compressor fault level was increased, the error of estimated mass flow rate increased. For this fault, it would be necessary to have an independent virtual sensor for refrigerant flow that could be used for fault detection and diagnosis as suggested by Kim and Braun (2012).

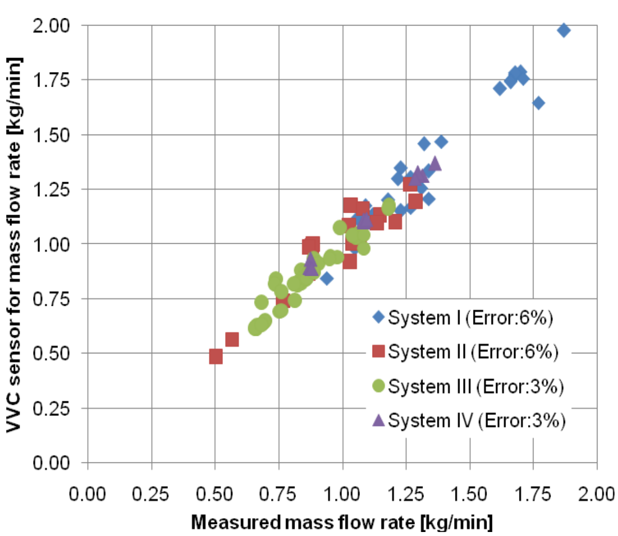

Figure 3: Performance of VRMF sensor (mass flow rate) under no fault conditions

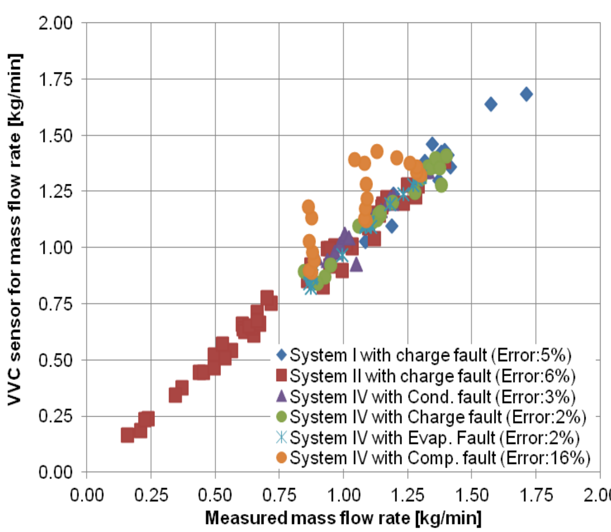

Figure 4: Performance of VRMF sensor (mass flow rate) under fault conditions

3.2 Input Power

Figure 5 shows performance of the VVC sensor for input power (VCP) under no fault conditions. In this case, the RMS error of estimated input power consumption was less than 4% for the four systems operating over a wide range of conditions in both heating and cooling mode. Figure 6 presents results under fault conditions in cooling and heating mode. For all of the faults except compressor valve leakage, the predictions and measurements are within about 5%. The errors are somewhat higher (8%) for compressor valve leakage, but the sensor outputs are still reasonable for this fault. Overall, the VCP sensor provides accurate estimates for both no fault and faulty conditions in cooling and heating mode.

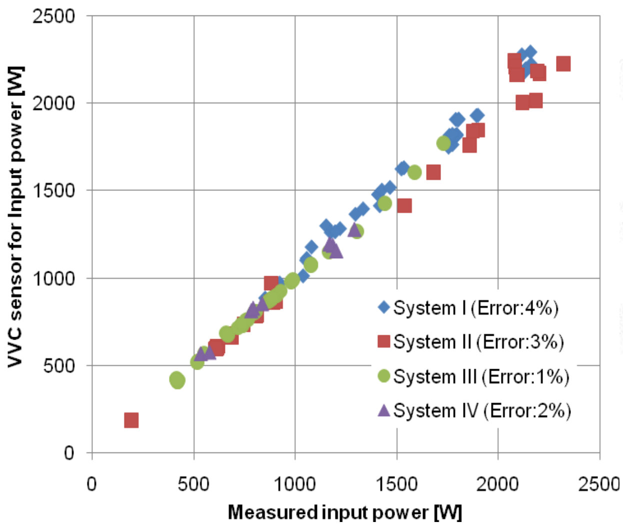

Figure 5: Performance of VCP sensor (input power) under no fault conditions

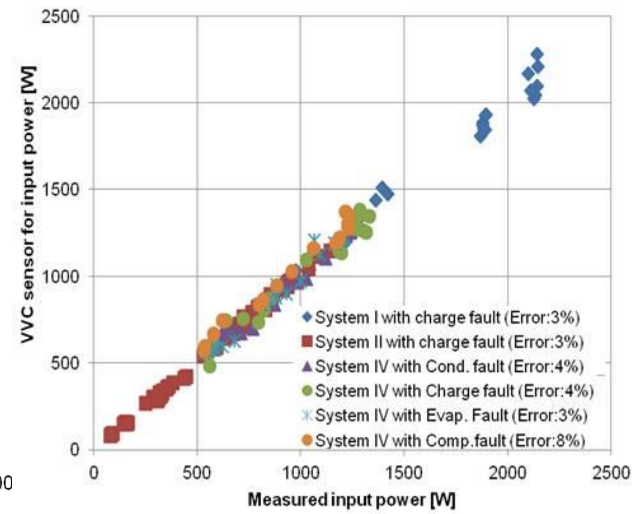

Figure 6: Performance of VCP sensor (input power) under fault conditions

4. ADDITIONAL VIRTUAL SENSORS

4.1 Prediction of compressor frequency

One of the inputs to the VCP sensor is compressor frequency, which can be difficult to measure for technicians in the field. However, for an embedded application, the compressor frequency would generally be available from the motor controller. Alternatively, the VCP sensor could be used to estimate compressor frequency if mass flow rate were provided as an input from another independent VRMF sensors. In this case, compressor frequency can be estimated by solving the second-order polynomial equation 9 using mass flow rate from a VRMF sensor as in input. The solution for operating compressor frequency is given in equation 10.

$$a_1\left(f_{estimation}-f_{rated}\right)^2+a_2\left(f_{estimation}-f_{rated}\right)+\left(a_3-\frac{\dot{m}_{VRMFsensor}}{\left(\dot{V}_{rated}\cdot\rho_{measured}\right)}\right)=0 \tag{9}$$

$$f_{estimation}=f_{rated}+\frac{-a_2+\sqrt{\left(a_2^2-4a_1\cdot\dot{m}_{VRMFsensor}\Big/\left(\dot{V}_{rated}\cdot\rho_{measured}\right)\right)}}{2a_1} \tag{10}$$

Figure 7 shows comparisons of calculated and measured compressor frequencies for system II for a range of different operating conditions. The average RMS errors are less than 6 % for cooling mode and 8% for heating mode. Estimated frequency could be used for monitoring system status and for identifying faults through comparison with frequency outputs from the motor controller. Figure 8 shows comparisons of measured and predicted compressor frequencies for system I under faulty conditions. In this case, the RMS errors of estimated compressor frequency under condenser fouling, evaporator fouling, and charge fault condition are less than 8 %, 10 % and 6 %, respectively.

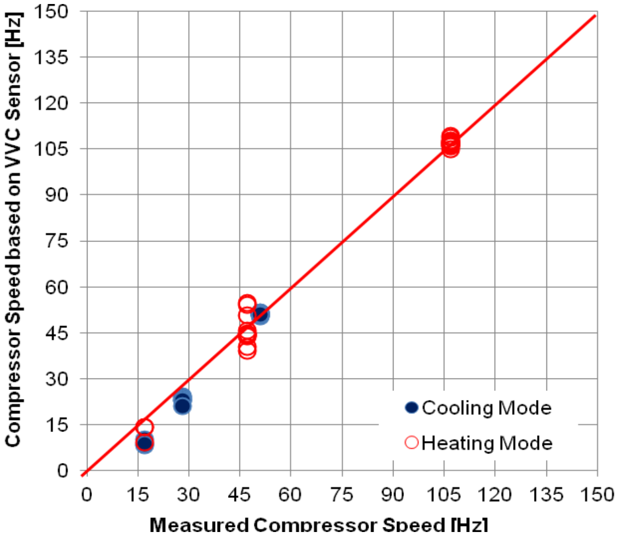

Figure 7: Comparisons of predicted and measured compressor frequencies for system II

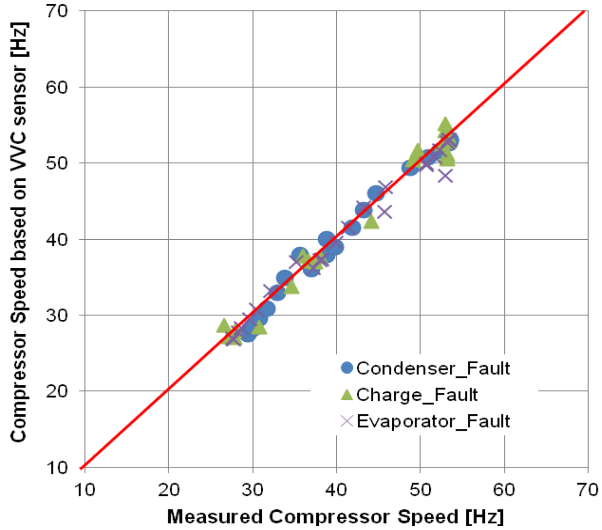

Figure 8: Comparisons of predicted and measured compressor frequencies for system VI

4.2 System performance

Capacity and energy efficiency (COP) are important parameters for evaluating the performance of air conditioner and heat pump systems. In general, physical measurements necessary to determine these parameters are not employed because of cost. However, these quantities can be determined using virtual sensors for mass flow rate and power consumption along with some other relatively low cost physical measurements using equations 11 and 12.

$$\dot{Q}_{predicted} = \dot{m}_{map}(T_c, T_e, f) \cdot \left(h_{evap,out}(P_{evap,out}, T_{evap,out}) - h_{liquid,in}\left(P_{liquid,in}, T_{liquid,in}\right) \right) \tag{11}$$

$$COP_{predicted} = \frac{Q_{predicted}}{W_{map}(T_c, T_e, f)} \tag{12}$$

where $Q_{predicted}$, and $COP_{predicted}$ are predicted capacity and energy efficiency using the virtual refrigerant mass flow rate and input power sensors.

Figures 9 and 10 show cooling capacity and COP normalized by rated values for system II that were determined using equations 11 and 12. These results are presented as a function of refrigerant charge for different ambient temperatures and operating frequencies. The nomenclature “MX” and “INT” refer to maximum (49Hz) and intermediate (27Hz) frequencies. This type of information could be used within an online tool for assessing the economics associated with servicing a unit if the refrigerant charge were not correct or if other faults existed. In this particular case, the results demonstrate that the impact of refrigerant charge on performance was relatively small if the charge was within 10 % of the rated charge. However, there was a dramatic reduction in both cooling capacity and energy efficiency with decreasing charge below about 70 % refrigerant charge. Overcharging had a somewhat larger effect on COP than undercharging within this range. When overcharged, the capacity did not show a large degradation, whereas the COP was reduced due to an increase in power consumption, which was caused by a rise of pressure ratio. There are similar trends for both frequencies, although the optimal charge for peak capacity and COP moves to lower values at lower speeds.

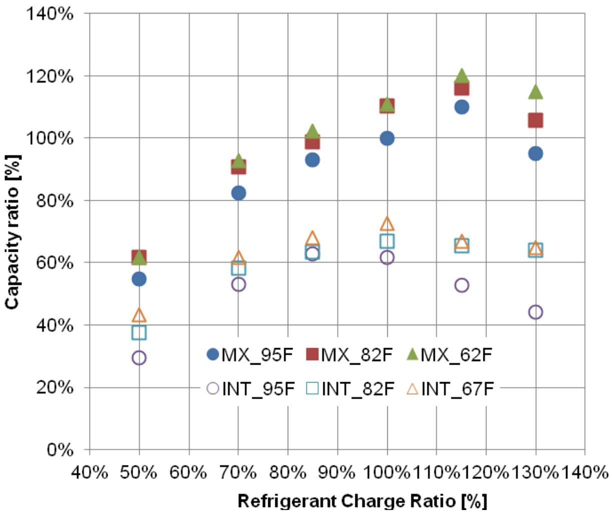

Figure 9: Cooling capacity ratio for system II based on refrigerant charge

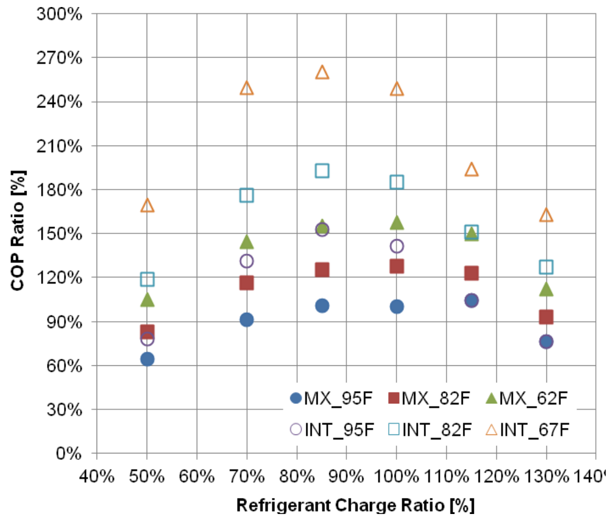

Figure 10: Cooling COP ratio for system II based on refrigerant charge

5. CONCLUSIONS

Virtual sensors were developed for variable-speed compressors to estimate mass flow rate and power consumption using low cost measurements and models. The flow rate and input power at a rated compressor speed are correlated in terms of condensation and evaporation temperature and then corrected for any operating speed using additional correlations in terms of operating frequency. The virtual sensors predicted mass flow rate and power consumption with RMS errors less than 6% and 4%, respectively, under normal conditions for four different systems over a range of operating conditions in heating and cooling mode. The virtual sensors also work well when faults are present, except in the case of compressor valve leakage faults that lead to errors in predictions of refrigerant flow rate. Overall, predictions of mass flow rate were about 16% higher than measurements under compressor fault conditions. In this case, other virtual sensors for refrigerant flow could be used to isolate this fault as described in Kim and Braun (2012).

For embedded applications, motor operating frequency would be an available input. However, for application to existing equipment in the field, compressor frequency can be difficult to measure. An alternative would be to use another virtual refrigerant mass flow measurement as an input to the compressor mass flow model in order to estimate compressor frequency for use in the virtual compressor power model. This was shown to work well using the data available in this study. The estimated frequency may be a useful index in monitoring and diagnosing faults for the system. In addition, the paper demonstrated outputs of virtual sensors for estimating system capacity and energy efficiency that are based on the virtual refrigerant flow and power sensors and other low-cost physical sensors. This information is very valuable for assessing the economic impacts of different faults.

NOMENCLATURE

a	Empirical constant for mass flow rate			**Subscripts**
b	Empirical constant for power		*c*	Condenser
c	Empirical constant for mass flow rate at rated condition		*e*	Evaporator
COP	Coefficient of Performance		*estimated*	Estimation
d	Empirical constant for power at rated condition		*evap,in*	Inlet of evaporator
f	Compressor speed	Hz	*liquid,in*	Inlet of liquid line
h	Enthalpy	kJ/kg	*map*	Mapping

K_{flow}	Ratio of refrigerant volumetric flow rate at operating speed to value at rated speed		*massflow*	Mass flow rate
K_{input}	Ratio of input power at operating speed to value at rated speed		*measured*	Measurement
Q	Capacity	W	*predicted*	Estimation
m	Refrigerant mass flow rate	kg/s	*power*	Input Power
P	Pressure	Pa	*rated*	Rated condition
T	Temperature	C	*suc*	Suction
V	Volume flow rate	m^3/s		**Greek**
VRMF	Virtual refrigerant mass flow		ρ	Density
VCP	Virtual compressor performance			
VVC	Virtual variable-speed compressor			
W	Input power	W		

REFERENCES

Aprea C., Mastrullo R., and Renno C., 2004, Fuzzy control of the compressor speed in a refrigeration plant, *International Journal of Refrigeration*, Vol.27(6), p. 639-649.

Browne M.W. and Bansal P.K., 2002, Transient simulation of vapor compression packaged liquid chillers, *International Journal of Refrigeration*, Vol.25, p. 597-610.

Guo-liang D., 2007, Recent developments in simulation techniques for vapor-compression refrigeration systems, *International Journal of Refrigeration*, Vol.30, p. 1119-1133.

Kim M. and Kim M.S., 2003, Performance investigation of a variable speed vapor compression system for fault detection and diagnosis, *International Journal of Refrigeration*, Vol.28, p. 481-488.

Kim, W.H. and Braun, J.E., 2012, Evaluation of virtual refrigerant mass flow sensors, *International Refrigerant and Air conditioning Conference*, No. 2298.

Navarro-Esbry J., Berbegall V., Verdu G., Cabello R., and Lopis R., 2007, A low data requirement model of a variable speed vapor compression refrigeration system based on neural networks, *International Journal of Refrigeration*, Vol.30, p. 1452-1459.

Park Y.C., Kim Y.C., and Min M.K., 2001, Performance analysis on a multi-type inverter air conditioner, *Energy Conversion Mgmt*, Vol.42, p. 1607-1621.

Quereshi T.Q. and Tassou S.A., 1996, Variable speed capacity controls in refrigeration system, *Applied Thermal Engng*, Vol.16(2), p. 103-113.

Nyika, S., 2011, Personal Communication, Ray W. Herrick Laboratories, Purdue University

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