Article

Virtual Pressure Sensor for Electronic Expansion Valve Control in a Vapor Compression Refrigeration System

Jian Sun [1],*, Jin Dong [1], Bo Shen [1] and Wenhua Li [2]

[1]   Oak Ridge National Laboratory, Oak Ridge, TN 37830, USA; dongj@ornl.gov (J.D.); shenb@ornl.gov (B.S.)

[2]   UTC—Carrier Corporation, Charlotte NC 28269, USA; wenhua.li@carrier.com

*   Correspondence: sunj2@ornl.gov; Tel.: +1-865-341-0078

check for updates

Abstract: Virtual sensor technology, which uses simulation models/algorithms to calculate a value to represent an unmeasured variable or replace a directly measured reading, has attracted many studies in the heating, ventilation, air conditioning and refrigeration (HVAC&R) industry. However, most virtual sensor technologies are developed for fault detection and diagnostics (FDD) purposes, which generally compare the virtual sensor values with actual measured values to detect if any fault occurred and identify the causes that led to the fault. It is rare to see studies focus on control performance of virtual sensors after substituting an actual sensor. This is particularly important for the system with no redundant sensor since a virtual sensor is the most effective way to operate the system in the desirable region when any sensor failure occurs. To address this gap, this paper develops a new virtual pressure sensor technology to substitute the actual pressure measurement for electronic expansion valve (EXV) control in a vapor compression refrigeration system by integrating compressor and valve characteristics. The control performance of this proposed virtual pressure sensor technology under various operating conditions is validated with experimental data. Closed loop EXV control simulations with the proposed virtual pressure sensor are conducted, and the results are analyzed.

Keywords: virtual pressure sensor; vapor compression refrigeration; model in the loop; electronic expansion valve

1. Introduction

Superheat control is critical for refrigeration systems to prevent compressor flooding while maintaining system capacity and efficiency. In general, two types of control devices are widely used: thermal expansion valves (TXVs) and electric expansion valves (EXVs). Due to the superior control characteristics, EXVs have become more popular in the refrigeration industry. Using the EXV technology to control system superheating requires both temperature and pressure sensors to be installed on the system. In the situation of a pressure transducer malfunctioning, which is not unusual, maintaining a proper superheat conditions becomes difficult and very crucial for system operation. If there is no redundant pressure transducer installed, the superheat temperature cannot be calculated directly based on the malfunctioning pressure sensor. An alternative approach is to use the estimated pressure, called virtual pressure, to calculate the superheat condition to supporting the EXV control.

Virtual sensor technology, which uses simulation models/algorithms to process what a physical sensor otherwise would, has attracted many studies in the heating, ventilation, air conditioning and refrigeration (HVAC&R) industry due to several significant advantages, such as cost effectiveness, easy maintenance and lack of constraints by the environment or operating condition. Andiroglu et al. [1] developed a virtual pump water flow sensor using either the variable frequency drive input power or motor input power with a calibrated pump, motor and variable frequency drive efficiencies along

with the consideration of additional harmonic energy loss. Eric et al. [2] presented a low cost, practical and non-intrusive virtual flow meter to estimate the chilled and condenser water mass flow rate using data from a building automation system. The virtual flow meter is used in fault detection and diagnostics (FDD) of the cooling plant coefficient of performance (COP). Christian et al. [3] developed a virtual flow sensor using a model, measurements of boundary conditions and an EXV control signal. Compared to other studies which focus on the single-phase inlet conditions, Christian et al.'s virtual flow sensor can be used for both two-phase and subcooled valve inlet conditions through continuous correction. Virtual pressure sensors [4] were developed for estimating compressor discharge line pressure, condensing pressure, liquid line pressure, evaporating pressure and suction line pressure for the purpose of FDD. Several virtual refrigerant charge sensors [5–7] were developed to evaluate the refrigerant charge level of different types of vapor compressor systems, such as those with/without an accumulator and fixed/variable speed compressors. Zhao et al. [8] used commonly available onboard chiller measurements to develop a virtual chiller condenser fouling sensor to monitor the condenser fouling status for the purpose of chiller FDD. Li et al. [9] described a virtual compressor power consumption sensor based on a 10-coefficient polynomial equation and virtual refrigerant flow sensor calculated through a compressor volumetric efficiency prediction. Hjortland et al. [10] developed virtual sensors for supply air flow rate and indoor fan power used for rooftop unit air-side diagnostics based on indoor fan differential pressure and speed measurements. Recently, virtual in situ sensor calibration (VIC) and sensor fault detection and diagnostics (SFDD) have been developed for the continuous correction of building system operation [11,12].

Almost all these virtual sensor technologies were developed for FDD purposes, which generally compare the virtual sensor values with actual measured values to identify if any fault occurred and the causes that led to the fault. It is rare for research to focus directly on replacing the actual sensor with the virtual sensor through integrating the virtual sensor into the control system and evaluating the impact of the virtual sensor on the control performance. This is particularly important for the system with no redundant sensors since a virtual sensor is the most effective way to operate the system in the desirable region when any sensor failure occurs. To address this issue, this study develops a new virtual pressure sensor technology to substitute the actual pressure measurement in the EXV control of a refrigeration system. The control performance of using this proposed virtual pressure sensor technology under various operating conditions is also investigated.

2. Mathematic Model of Virtual Pressure Sensor

In a typical vapor compression refrigeration system, the EXV adjusts the refrigerant volume across the EXV to maintain a desired superheat temperature at the outlet of the evaporator. The compressor is to transport a certain volume of refrigerant from a certain state (suction) to another state (discharge). The refrigerant volume through the EXV and compressor is balanced based on the EXV and compressor operation characteristics (Figure 1), which can be formulated to predict the suction pressure or evaporator superheat temperature under various operating conditions.
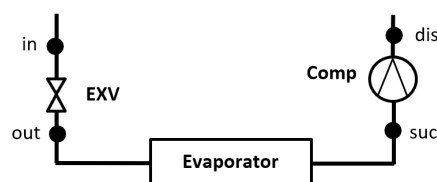

Figure 1. Compressor and Electronic expansion valve (EXV) scheme in vapor compression refrigeration system.

The following assumptions are used for simplifying the theoretical identification of the independent variables which influence the pressure prediction.

- The refrigerant suction side pressure loss through the evaporator is a function of mass flow rate.

- The refrigerant discharge side pressure loss through the condenser and economized heat exchanger is a function of mass flow rate.
- The energy loss when the refrigerant passes through the expansion valve is negligible.
- It should be noted that the approaches discussed in this paper assume the refrigerant does not have a varying saturation temperature for a specific pressure (generally referred to as "glide").
- The condenser fan is at a constant speed.
- The physical sensors for inputs of this model are assumed to be fault free.

According compressor characteristics, the compressor suction side mass flow is

$$m = \rho_{suc} V_{suc} \tag{1}$$

The suction volumetric flow rate V_{suc} can be estimated by [13]

$$V_{suc} = \left\{ 1 - C_0 \left[\left(\frac{P_d}{P_s} \right)^{\frac{1}{\gamma}} - 1 \right] \right\} N_{comp} V_{disp} \tag{2}$$

where C_0 is a constant, γ is an isentropic exponent and a constant for a specific refrigerant, P_d is compressor discharge side refrigerant pressure, P_s is compressor suction side refrigerant pressure, N_{comp} is compressor speed and V_{disp} is compressor displacement volume.

According to EXV throttling characteristics [14], the refrigerant mass flow crossing the EXV is

$$m = C_{exv} A_{exv} \sqrt{2\rho_{in}(P_{in} - P_{out})} = C_{exv} A_{exv} \sqrt{2\left(1 - \frac{P_{in}}{P_{out}}\right)} \sqrt{\rho_{in} P_{in}} \tag{3}$$

where A_{exv} is valve throat area, C_{exv} is flow coefficient, ρ_{in} is EXV inlet refrigerant density, P_{in} is EXV inlet refrigerant pressure and P_{out} is EXV outlet refrigerant pressure.

A correction factor, CF, is introduced to rewrite the above equation as

$$m_{exv} = CF \sqrt{\rho_{in} P_{in}} \tag{4}$$

where

$$CF = C_{exv} A_{exv} \sqrt{2\left(1 - \frac{P_{in}}{P_{out}}\right)} \tag{5}$$

A polynomial function is developed to calculate the correction factor

$$CF = C_1 + C_2 Exv\% + C_3 Exv\%^2 \tag{6}$$

where C_1, C_2, C_3 are constants and $Exv\%$ is expansive valve opening position as a percentage.

The refrigerant density at compressor suction ρ_{suc} and EXV inlet ρ_{in} in Equations (1) and (4) can be evaluated through refrigerant property calculation functions

$$\rho_{suc} = Ref(P_s, T_{suc}) \tag{7}$$

$$\rho_{in} = Ref(P_{in}, T_{in}) \tag{8}$$

where P_s, T_{suc} are the pressure and temperature at the compressor suction port and P_{in}, T_{in} are the EXV inlet pressure and temperature.

The relationships between P_s and P_{out}, P_d and P_{in} are given by

$$dP_{suc} = dP_{suc\_norm} * (m/m_{norm})^2 \tag{9}$$

$$dP_{dis} = dP_{dis\_norm} * (m/m_{norm})^2 \tag{10}$$

where dP_{suc_norm}, dP_{dis_norm} are the normalized pressure loss of suction side and discharge side and m_{norm} is the normalized refrigerant mass flow rate.

Substitute Equations (6)−(10) into Equation (5) to receive

$$\frac{Ref(P_s, T_{suc}) \times \left\{1 - C_0\left[\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}} - 1\right]\right\} N_{comp} V_{disp}}{= \left(C_1 + C_2 Exv\% + C_3 Exv\%^2\right) \times \sqrt{2Ref\left(P_d - dP_{dis}, T_{in}\right)\left(P_d - P_s - dP_{suc} - dP_{dis}\right)}} \tag{11}$$

In Equation (11), $C_0, \ldots C_6$, dP_{suc}, dP_{dis} are constants and can be curve fitted with experimental or manufactured test data. T_{suc}, P_d, T_{in} are measurable variables, $Exv\%$ is the control signal and the only unknown variable is P_s. Thus, solving Equation (11) results in a predicted suction pressure value, then a superheat temperature value with the leaving evaporator temperature measurement. The flow chart shown in Figure 2 describes the algorithm.
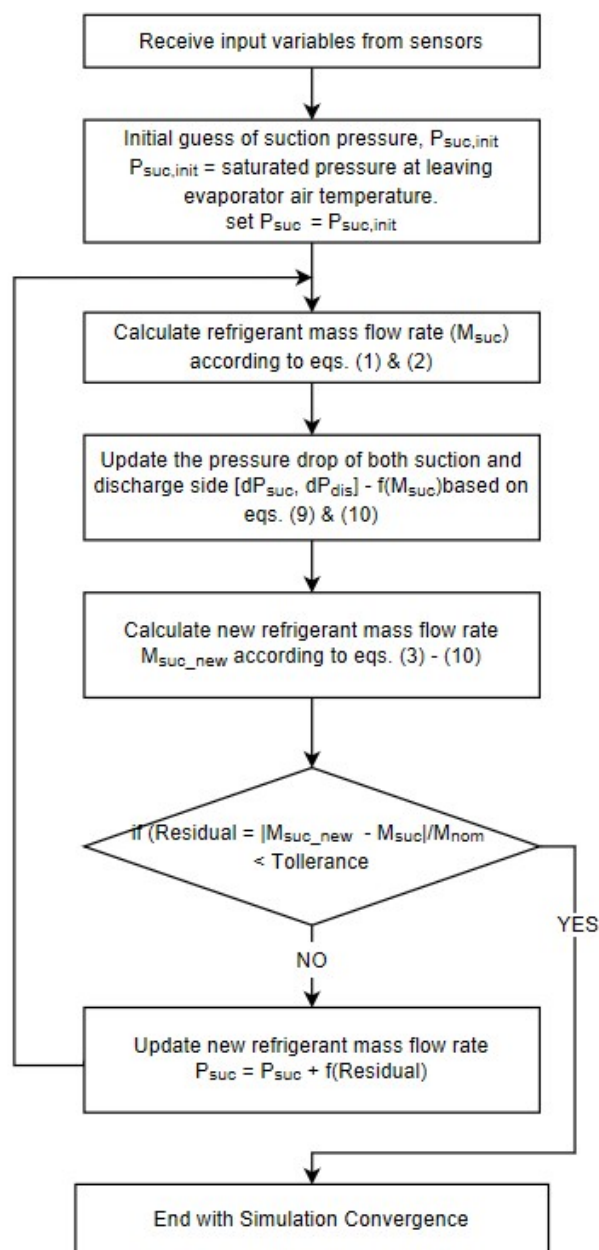
Figure 2. Pressure calculation based on compressor and EXV characteristics.

As shown in Figure 2, the virtual pressure sensor module receives sensor inputs per each time step. An initial guess of compressor suction pressure is given as the saturated pressure at the evaporator supply air temperature. The refrigerant mass flow rate can be calculated by compressor characteristics as in Equations (1) and (2). The pressure loss of both the suction side and discharge side can be calculated with Equations (9) and (10). Then a new refrigerant mass flow rate is given by solving Equations (4)–(6). The difference between these two calculated refrigerant mass flow rates is checked with a predefined convergence tolerance. If the simulation does not converge, the compressor suction pressure is updated with a new value to repeat the above steps, otherwise the calculated compressor suction pressure is the solution.

3. Model Calibration and Validation

The steady-state testing data from a typical vapor compression refrigeration system are used to calibrate the proposed virtual pressure sensor model method to identify the model parameters and evaluate its performance. These testing data cover a wide range of operation conditions: the ambient temperature can vary from 50 °C to −25 °C, and the refrigerated space can be set at any temperature between 30 °C and −20 °C. The calibrated model parameters are listed in Table 1. The tested compressor temperature and pressure (suction and discharge), mass flow rate and power consumption are used to identify the compressor constant C_0. The EXV inlet and outlet state (pressure, temperature, quality) and mass flow rate can be used to get the EXV constants C_1, C_2 and C_3.

Table 1. Model parameters.

Model Parameters	
γ	1.13
N_{comp}	3600 rpm
V_{disp}	1.45×10^{-4} m^3
C_0	0.0139
C_1	-2.45783×10^{-5}
C_2	2.7928×10^{-6}
C_3	-1.88744×10^{-8}

With the calibrated parameters, the predicted mass flow rate crossing the compressor and EXV is compared with experiment testing data, as shown in Figure 3. The accuracy of the refrigerant mass flow rate prediction of the compressor is within ±10% and can reach ±5% at relatively high load operations. The estimated refrigerant mass flow crossing the EXV is within ±5% compared to the testing data for most operating conditions except for some very low-load conditions.
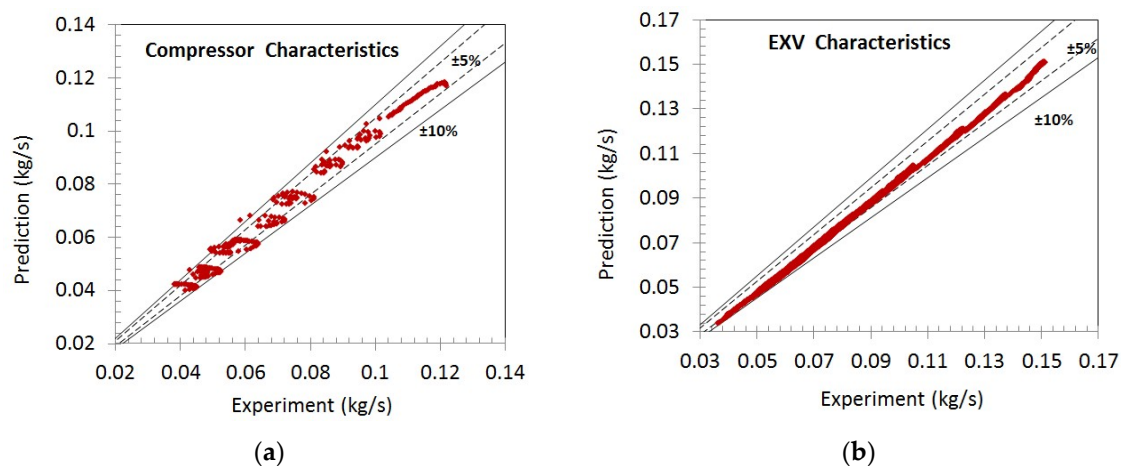

Figure 3. (**a**) Compressor; (**b**) EXV model calibration.

To validate the virtual pressure sensor (VPS), three selected testing data sets are used to compare the predicted superheat condition based on a virtual pressure sensor with the superheat condition based on the measured pressure. The three testing data sets are:

- Test Case 1: The refrigerated space temperature was cooled from 37.8 °C to −17.8 °C under 37.8 °C ambient temperature.
- Test Case 2: The refrigerated space temperature was cooled from 15.6 °C to −17.8 °C under 15.6 °C ambient temperature.
- Test Case 3: The refrigerated space temperature was cooled from 37.8 °C to 0 °C under 37.8 °C ambient temperature.

The results are plotted in Figures 4–6. As shown in Figures 4 and 5, the predicted superheat temperatures based on proposed virtual pressure sensor match well with the measured superheat temperature for the entire test process for both Test Case 1 and Test Case 2.
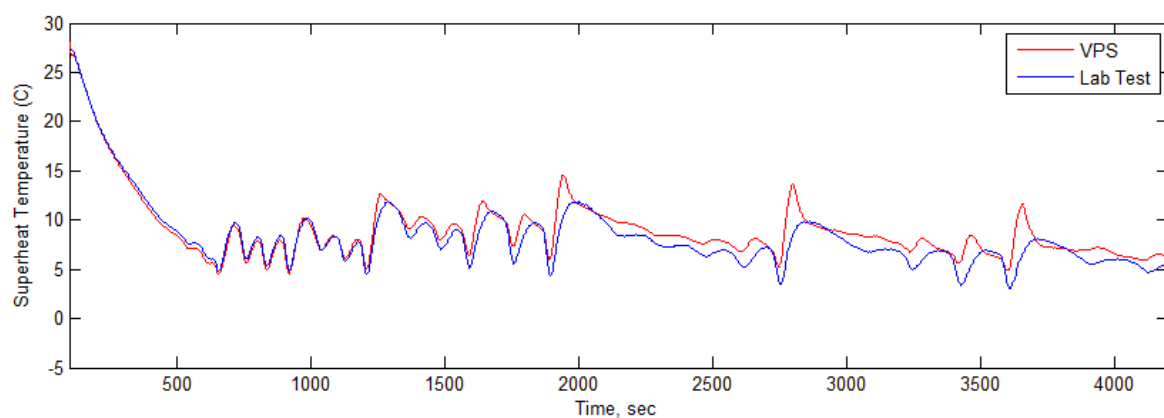
Figure 4. Test Case 1—superheat temperature estimation: Virtual Pressure Sensor (VPS vs. test.
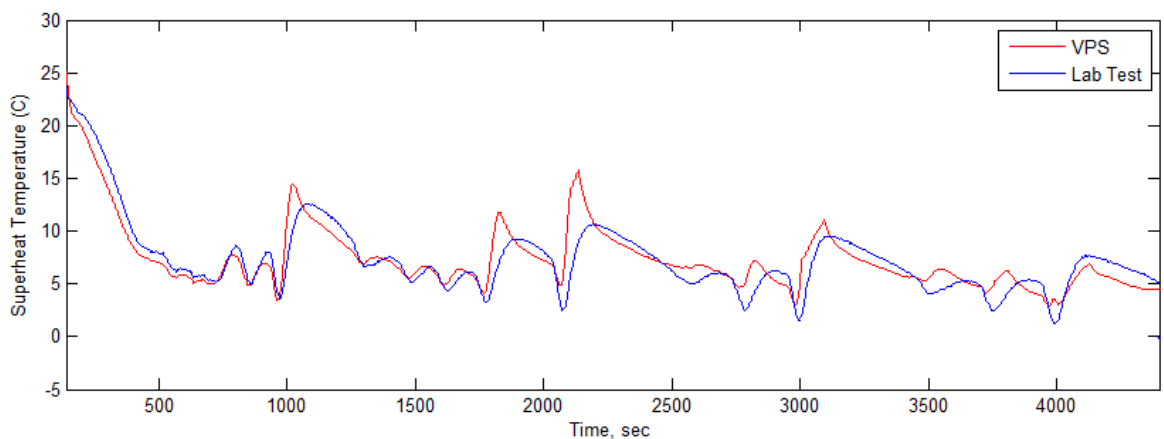
Figure 5. Test Case 2—superheat temperature estimation: VPS vs. test.

Figure 6 demonstrates some errors when using a virtual pressure sensor to predict the superheat temperature for Test Case 3. We found that errors in the first 500 s come from the valve saturation (EXV opens around 100%), which invalidates the valve model. Fortunately, the EXV control in this operating zone is not needed and the superheat value is not relevant. When the EXV moves away from the saturation state after 500 s, the results show that the virtual sensor predicts the superheat temperature well until around 1200 s, when the system is at partial load operation with the compressor running under pulse width modulation (PWM). This PWM control leads to the high frequency fluctuation of actual pressure/superheat temperature. This dynamic characteristic cannot be caught through

the steady-state model method as we proposed. Thus, the predicted superheat temperature shown in Figure 6 is the average of the actual measured superheat temperatures but cannot present its dynamic change.
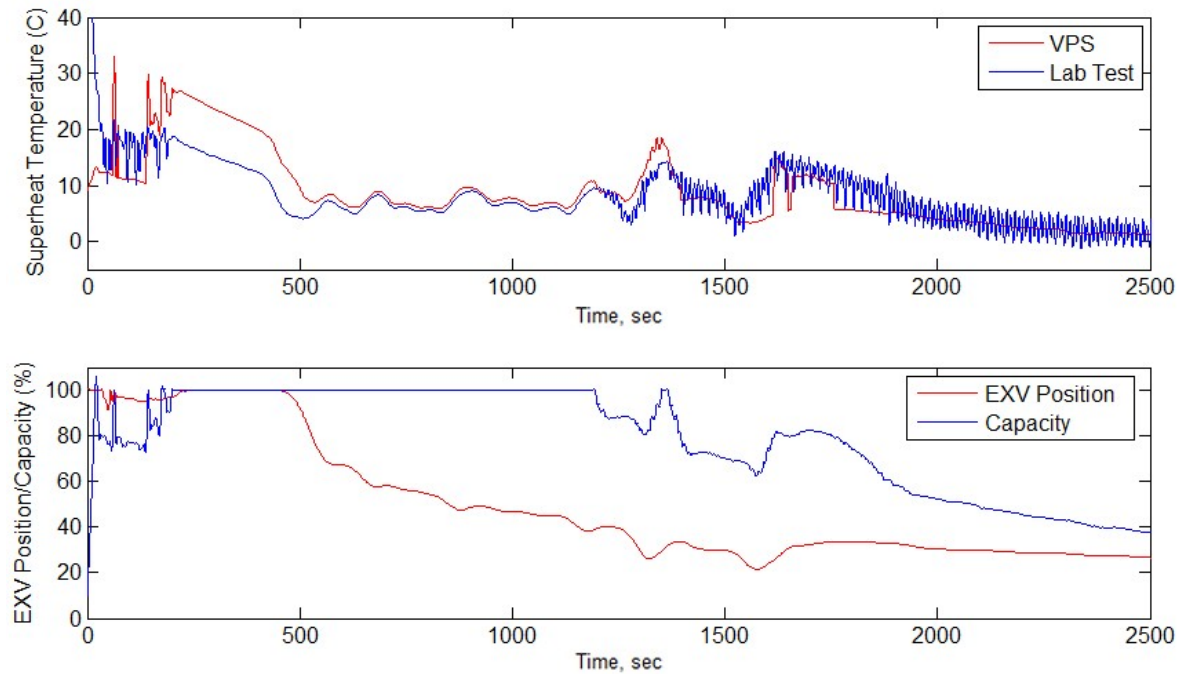

Figure 6. Test Case 3—superheat temperature estimation: VPS vs. test.

In summary, the calibration and validation results of the virtual pressure sensor performance are:

- The proposed virtual pressure sensor can estimate the pressure/superheat temperature with reasonable accuracy to support EXV control.
- The proposed method has better accuracy in the pressure/superheat temperature estimation in the full load/near full load conditions than partial load conditions (Figure 6).
- The pressure/superheat temperature estimate is not applied for the EXV saturation condition when the EXV is fully open at 100% and EXV control is not needed.

4. Electronic Expansion Valve (EXV) Control Using Virtual Pressure Sensor

After the calibration and validation of the virtual pressure sensor, we apply it to EXV control to evaluate its control performance. Instead of using the actual refrigeration system, a virtual simulation environment, called the model in the loop (MIL) simulation platform, is developed to test virtual pressure sensors for EXV control. The MIL simulation platform is shown in Figure 7. The MIL test platform is composed of five modules, including a test case generator, a Virtual Suction Pressure (VSP) pressure/superheat estimation module, control models, refrigeration system dynamic models and a simulation results module. The platform is set up in a Matlab/Simulink environment. The control model and VSP pressure/superheat estimation are built with Simulink. The system selected for this control test is a typical vapor compression refrigeration system which consists of several major components: a two-stage compressor, a fin coil evaporator and condenser, an economizer and thermal and electronic expansion valves. Its dynamic model is developed through Dymola/Modelica and imported into Matlab/Simulink through the functional mockup interface (FMI) toolbox.
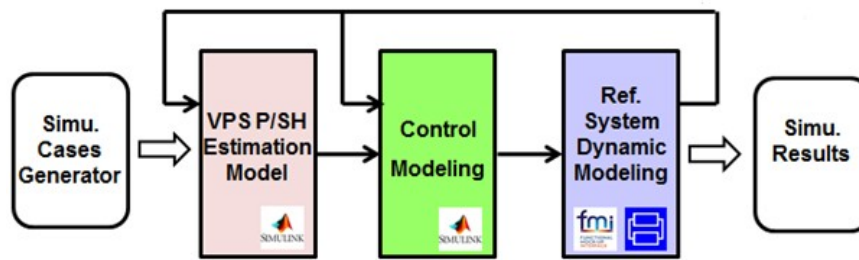
Figure 7. Model in the Loop (MIL) test framework.

Three MIL simulation cases are used to evaluate the VSP performance of estimating the pressure/superheat temperature in EXV control:

- MIL simulation scenario 1: The refrigerated space temperature was cooled from 37.8 °C to 13.9 °C under 37.8 °C ambient temperature.
- MIL simulation scenario 2: The refrigerated space temperature was cooled from 37.8 °C to 1.7 °C under 37.8 °C ambient temperature.
- MIL simulation scenario 3: The refrigerated space temperature was cooled from 37.8 °C to −17.8 °C under 37.8 °C ambient temperature.

For each MIL simulation test, the simulation results are compared between the system with measured pressure and the system with the virtual pressure sensor:

a.	Suction pressure: to evaluate the virtual pressure sensor pressure prediction accuracy.
b.	EXV position: to evaluate the impact of virtual pressure sensor performance on the EXV controller output.
c.	Evaporator supply/return air temperature: to evaluate the impact of virtual pressure sensor performance on the controlled temperature.

The EXV control performances with the proposed virtual pressure sensor are shown in Figures 8–10. Several key performance variables, supply/return air temperature, EXV opening position and suction pressure, are compared between EXV control based on the measured pressure and EXV control based on the calculated pressure based on the proposed virtual pressure sensor. The MIL simulation results indicate:

- The EXV control using the proposed virtual pressure sensor could control the supply/return temperature very well at all three test examples: the virtual pressure sensor-based EXV control maintains almost the same supply/return air temperature as the measured pressure EXV control.
- The suction pressure calculated by the virtual pressure sensor matches well with the actual measured pressure in most of operating conditions except for a few narrow operation zones, such as the transition zone from full load operation to partial load operation in MIL simulation scenario 1 and a small operation zone during full load operation in MIL simulation scenario 2. These discrepancies result in different EXV opening position. These errors are generally caused by a model error of the virtual suction pressure senor. Fortunately, the impact of these pressure calculation errors on a controlled objective, supply/return air temperature, are relatively small and can be ignored.
- When the system is running with a compressor pulse width modulation control zone, the calculated suction pressure based on the virtual pressure sensor model agrees with the average actual pressure but cannot match the dynamic characteristics of the actual pressure. The primary reason is that the proposed virtual pressure sensor is a steady-state model and cannot handle the dynamic change of pressure. This drawback of the virtual pressure sensor has little influence on the EXV control performance since the EXV control takes the average pressure signal as input to calculate the control output signal.
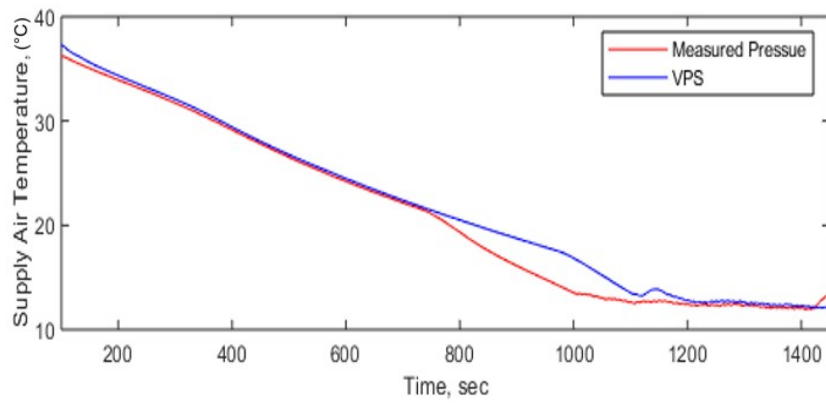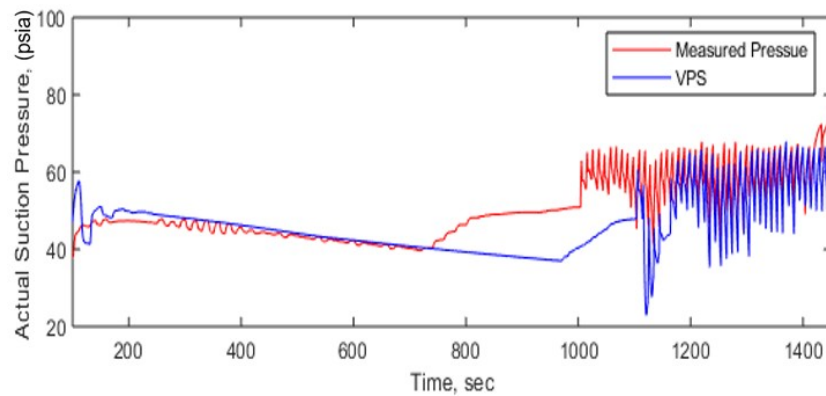
(**a**) Supply air temperature (°C).
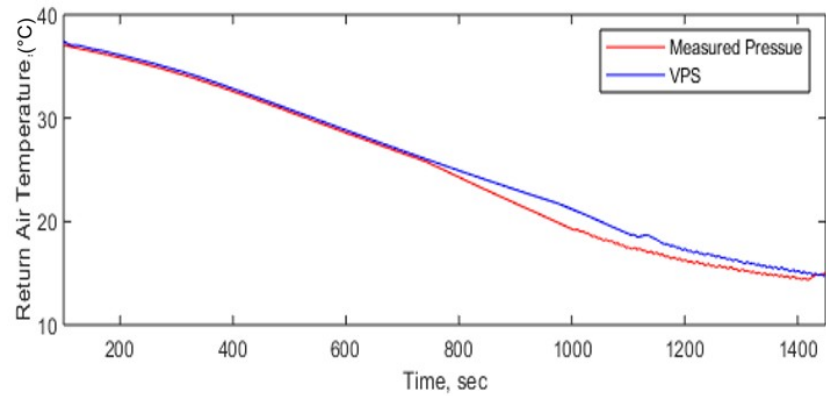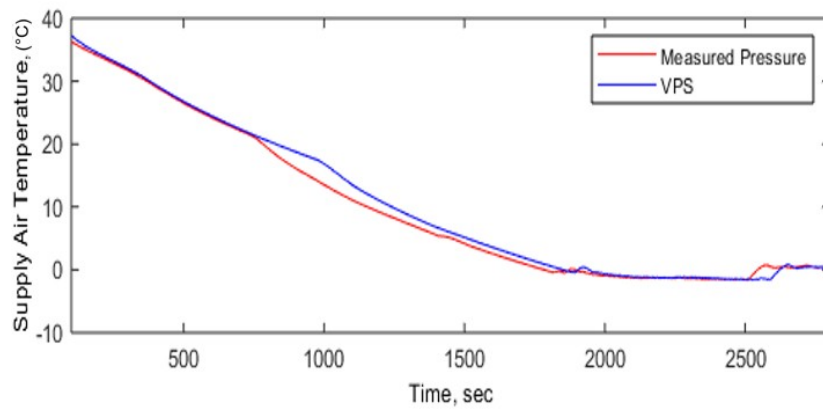

(**b**) Return air temperature (°C).
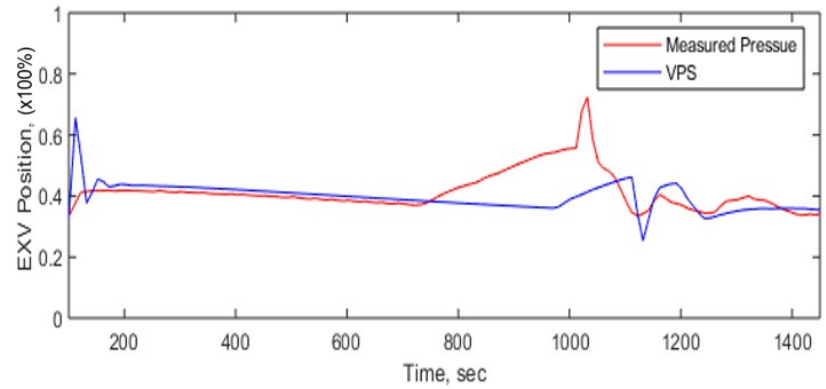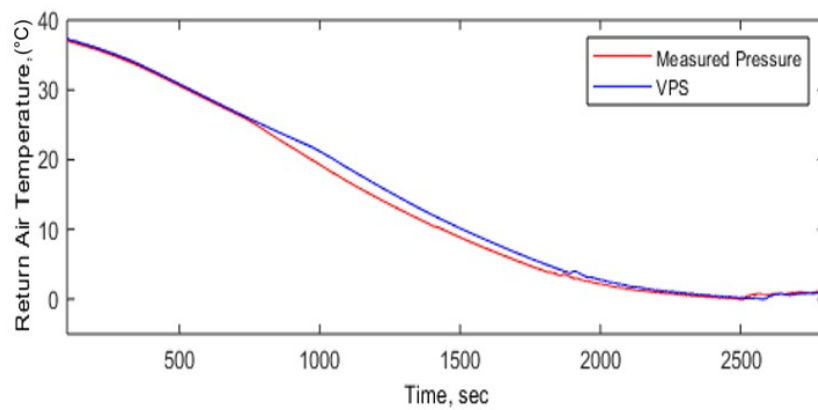

(**c**) EXV position (%).

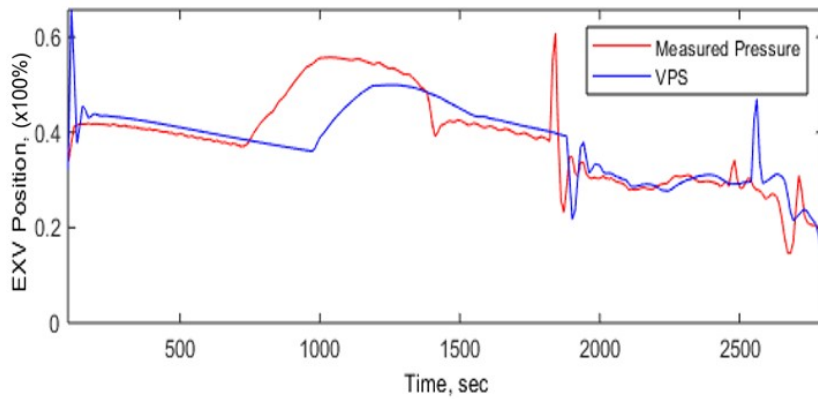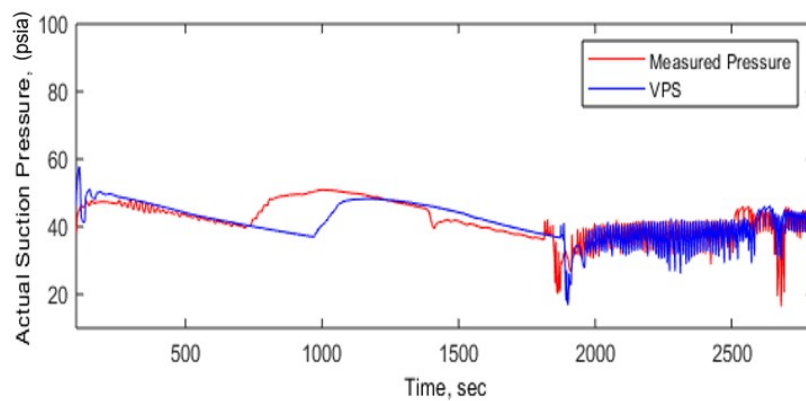(**d**) Actual suction pressure (psia).

Figure 8. MIL simulation results for scenario 1.

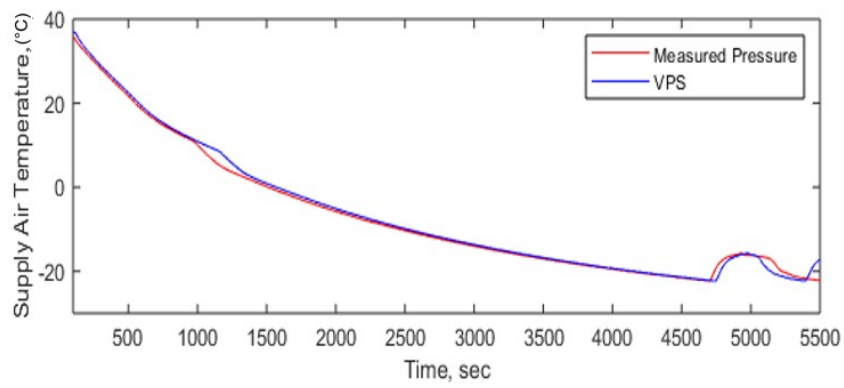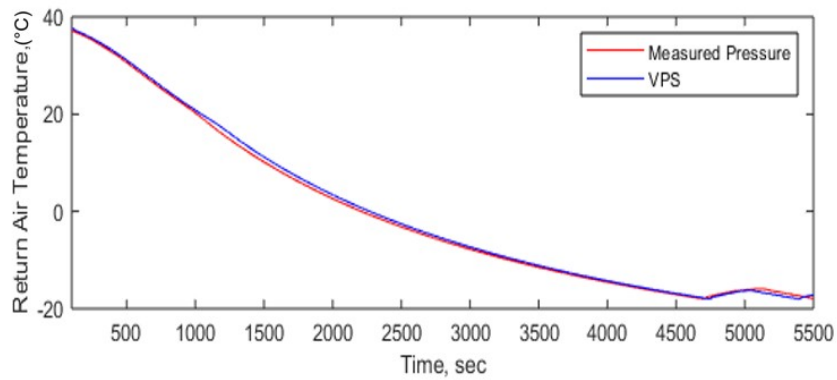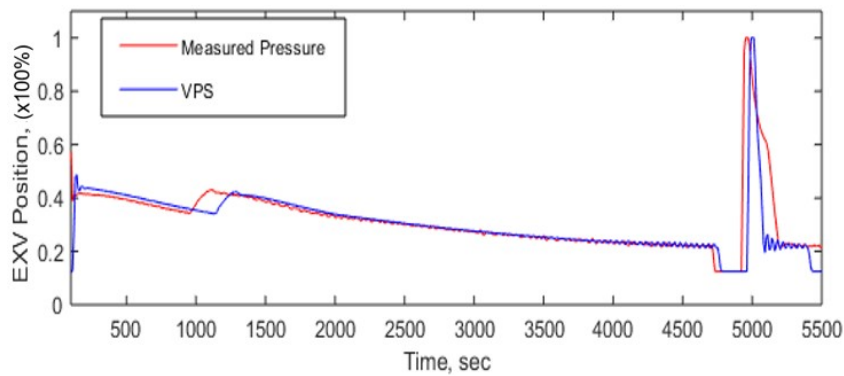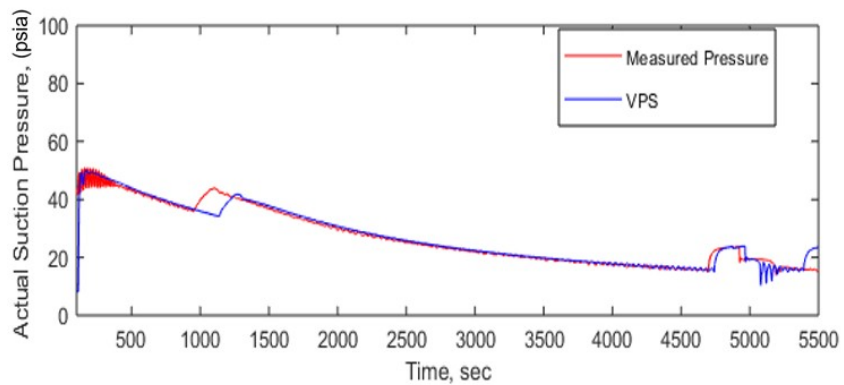(**a**) Supply air temperature (°C).

(**b**) Return air temperature (°C).

(**c**) EXV position (%).

(**d**) Actual suction pressure (psia).

Figure 9. MIL simulation results for scenario 2.

(**a**) Supply air temperature (°C).

(**b**) Return air temperature (°C).

(**c**) EXV position (%).

(**d**) Actual suction pressure (psia).

Figure 10. MIL simulation results for scenario 3.

In general, the MIL comparison simulation results indicate that the proposed virtual pressure sensor can substitute the actual pressure sensor in terms of EXV control performance. This virtual pressure sensor technology is able to provide additional redundancy and guarantee EXV control in a vapor compression refrigeration system.

5. Conclusions

Superheat control is one of the most important control functions in vapor compression refrigeration systems, since it prevents compressor flooding and maintains system capacity and efficiency. Both temperature and pressure sensors are required for EXVs to control refrigeration system superheat temperatures. It is not abnormal for pressure sensors to malfunction during operation. For a system without redundant pressure sensors, the virtual pressure sensor technology is a cost-effective and convenient method to calculate the pressure to maintain the EXV superheat control. This study developed an easily implemented virtual pressure sensor model to address this issue. The calculated pressures based on the proposed virtual pressure sensor model are compared with measured pressures in three lab tests, which verifies the accuracy of the virtual pressure sensor. A model-based control platform is used to further verify the influence of the virtual pressure sensor on the EXV control performance through three MIL simulation tests. The results are analyzed and it is concluded that the proposed virtual pressure sensor can be used to substitute the actual pressure sensor for EXV superheat control. Further studies can be conducted on how to automatically identify the sensor faults and continuously calibrate the virtual sensor.

Nomenclature

Symbols

A	Area, m^2
C	Constant/parameter
dP	Pressure drop, pa
CF	Correction factor
Exv%	Electronic expansion valve opening, %
m	Mass flow rate, kg/s
N	Compressor speed, rpm
P	Pressure, pa
T	Temperature, °C
V	Volumetric flow rate, m^3/s
ρ	Density, kg/m^3
γ	Isentropic exponent

Subscripts and abbreviations

air	Air side
comp	Compressor
dis, d	Compressor discharge port
disp	Compressor displacement
exv, EXV	Electronic expansion valve
in	Electronic expansion valve inlet
init	Initial value
norm	Normalized value
out	Electronic expansion valve outlet
suc, s	Compressor suction port
0,1,2,3	State number, constant number

References

1. Andiroglu, E.; Wang, G.; Song, L.; Kiamehr, K. Development of a Virtual Pump Water Flow Meter Using Power Derived from Comprehensive Energy Loss Analysis. *Sci. Technol. Built Environ.* **2016**, *22*, 214–226. [CrossRef]
2. McDonald, E.; Zmeureanu, R. Virtual Flow Meter for Chilled and Condenser Water for Chiller: Estimates Versus Measurements. *Sci. Technol. Built Environ.* **2016**, *22*, 178–188. [CrossRef]
3. Bach, C.K.; Groll, E.A.; Braun, J.E.; Horton, W.T. Development of a Virtual EXV Flow Sensor for Applications with Two-Phase Flow Inlet Conditions. *Int. J. Refrig.* **2014**, *45*, 243–250. [CrossRef]
4. Li, H.; Braun, J. Virtual Refrigeration Pressure Sensors for Use in Monitoring and Fault Diagnosis of Vapor-Compressor Equipment. *HAVC&R Res.* **2009**, *15*, 597–616.
5. Kim, W.; Braun, J. Performance Evaluation of a Virtual Refrigerant Charge Sensor. *Int. J. Refrig.* **2013**, *36*, 1130–1141. [CrossRef]
6. Kim, W.; Braun, J. Extension of a Virtual Refrigeration Charge Sensor. *Int. J. Refrig.* **2015**, *55*, 224–235. [CrossRef]
7. Li, H.; Braun, J. Development, Evaluation and Demonstration of a Virtual Refrigerant Charge Sensor. *HAVC&R Res.* **2009**, *15*, 117–136.
8. Zhao, X.; Yang, M.; Li, H. A Virtual Condenser Fouling Sensor for Chiller. *Energy Build.* **2012**, *52*, 68–76. [CrossRef]
9. Hjortland, A.; Braun, J. Virtual Sensors for Rooftop Unit Air-Side Diagnostics. *Sci. Technol. Built Environ.* **2016**, *22*, 189–200. [CrossRef]
10. Li, H.; Braun, J. Decoupling Features and Virtual Sensors for Diagnosis of Faults in Vapor Compression Air Conditioners. *Int. J. Refrig.* **2007**, *30*, 546–564. [CrossRef]
11. Yoon, S.; Yu, Y. Strategies for Virtual In-Situ Sensor Calibration in Building Energy Systems. *Energy Build.* **2018**, *172*, 22–34. [CrossRef]
12. Wang, P.; Yoon, S.; Wang, J.; Yu, Y. Automated Reviving Calibration Strategy for Virtual In-Situ Sensor Calibration in Building Energy Systems: Sensitivity Coefficient Optimization. *Energy Build.* **2019**, *198*, 291–304. [CrossRef]
13. Kuehn, T.H.; Ramsey, J.W.; Threlkeld, J.L. *Thermal Environmental Engineering*, 3rd ed.; Prentice Hall: Upper Saddle River, NJ, USA, 1998.
14. Xue, Z.; Shi, L.; Ou, H. Refrigeration Flow Characteristics of Electronic Expansion Valve Based on Thermodynamic Analysis and Experiment. *Appl. Therm. Eng.* **2008**, *28*, 238–243.