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ABSTRACT

The performance of the heat pump system varies greatly depending on the refrigerant charge amount. Improving the refrigerant charge Fault Detection and Diagnostics (FDD) method of vapor compression systems have the potential for increasing energy efficiency and reducing service cost. Previous studies to predict refrigerant charge amount are mostly empirical methods which require significant amount of experimental data for high accuracy. The primary goal of this research is to develop a universal charge fault detection method which requires only a few experimental data with high prediction accuracy.

Currently, pump down operations are typical practices by HVAC technicians when they need to open the refrigerant circuit to make a repairment. In addition, compressors have a built-in low-pressure cut-off protection function, and the compressor performance maps are commonly available from manufacturers. The proposed method innovatively utilizes the typical pump down operation, the compressor low-pressure cut-off protection, and the compressor performance map. It does not require any geometry information of heat exchangers, refrigerant lines, or charge buffers.

The new charge prediction method is firstly formulated through theoretical analysis, then verified and calibrated by a quasi-steady-state simulation of the pump down process for a residential heat pump system. The quasi steady-state simulation uses an HVAC system simulation framework driven by DOE/ORNL Heat Pump Design Model (HPDM). Preliminary experiment validations with heat pump refrigerant leakage tests demonstrate the deviation of the proposed charge prediction method compared with measurement is within 8%. This technology makes refrigerant charge amount available at the technician’s fingertips and leads to shorter maintenance time and fewer site visits.

Key words: refrigerant charge, fault detection and diagnostics, pump down, Quasi steady-state modeling

1. INTRODUCTION

Operation reliability of the HVAC systems plays an essential role in ensuring energy efficiency and indoor thermal comfort. To operate and maintain heat pump systems efficiently, the energy efficiency should be maximized through optimal control and the reliability should be maintained by continual fault diagnosis technology. For the residential and commercial HVAC systems, faults occur inevitably after long term operation. The occurrence of a fault is usually related to the improper initial installation or the degradation of components during the operation. Faults in heat pump systems can be classified into hard and soft faults. Hard faults lead to system halt and can be easily detected and diagnosed. However, soft faults such as refrigerant leakage are difficult to detect and diagnose before the fault accumulates to severe consequences.

According to Hong et al. (2019), refrigerant charge fault is the most costly soft fault among all different types of heat pump faults and has become a significant hurdle of heat pump efficiency. Rossi (2004) showed that refrigerant charge accounts for 63% of tune-up faults of air conditioners. The refrigerant charge fault mainly attributes to refrigerant leakage (Madani and Roccatello (2014)). Kim and Braun (2012) revealed that the cooling capacity of a heat pump with 75% refrigerant charge, i.e., low-charged system, degrades by 20% than the same system with nominal refrigerant charge. And the system with 75% charge yields 16% Seasonal Energy Efficiency Ratio (SEER) degradation than nominal charged system. Domanski et al. (2015) found a heat pump with 10%-16% over-charge results in up to 30% increase in annual electricity consumption. As a conclusion, low-charge and over-charge systems induce efficiency degradation. To avoid operating heat pumps off-design conditions, the systems should always be charged with the appropriate amount of refrigerant.

The research on developing refrigerant charge fault detection methods have been paid great attention for last few decades. Grace et al. (2005) and Kocyigit et al. (2014) found the superheat degree at the compressor suction and subcooling degree at the condenser outlet are both affected by the amount of refrigerant charge in the system. Thus, superheat degree and subcooling degree can be good predictors of refrigerant charge. Despite their method shows high reliability, the method requires many experimental data. Considering the operation of heat pump is continuously fluctuating due to the changes of ambient temperature and building load, the extraction of steady-state operation data requires long data acquisition time. Tassou and Grace (2005) predicted refrigerant leakage in a vapor compression system using Artificial Neural Network (ANN). In addition to a large amount of experimental data required to train their ANN model, their model is only applicable for the specific heat pump system because the trained model is system dependent. Li and Braun (2007) developed a mathematical decoupling framework for charge fault detection of vapor compression systems. Their method was effective to detect the refrigerant undercharge and overcharge faults. However, their method cannot quantify the total refrigerant charge amount. That is, if their method detects a charge fault, a technician cannot know how much refrigerant should be charged to or released from the system. Later, Li and Braun (2009) presented the virtual refrigerant charge sensor for the vapor compression systems. The subcooling and superheat temperatures were used as the input variables to estimate the refrigerant charge amount. However, this model was only suitable for the heat pumps with the fixed speed compressor. According to Kim and Braun (2013), the calculation accuracy of their model was not satisfying when applied for the variable speed heat pump systems. After that, Kim and Braun (2015) improved the model by adding the evaporator inlet quality and the discharge superheat temperature as two additional predictors. However, under the conditions of lower outdoor temperatures or higher fault intensities (such as 70% low-charge or 130% over-charge), the performance of their Fault Detection and Diagnostics (FDD) method has large deviation compared with the actual charge amount.

The objective of this paper is to develop a universal charge identification approach. It is an accurate, fast, cost-effective refrigerant charge fault detection method which can be executed automatically during existing vapor compression system commissioning operations. The new charge FDD method will utilize pump-down operations in the field and well-known compressor information (compressor maps) to identify key parameters and characterize a system charge model on the fly. It does not require geometry details of any components such as heat exchanger internal volumes, length of pipelines, and internal volume of accumulator and receiver as the previous charge models need. It is universally applicable to a variety of vapor compression systems, i.e., packaged systems and split systems under various field conditions.

2. METHODOLOGY

2.1 Pump-down Operation

Currently, HVAC technicians are regularly performing refrigerant pump down operations when they need to open the refrigerant circuit to make a repairment. Figure 1 shows a typical residential heat pump system. There are two cut-off valves, one valve in the suction line and one valve in the liquid line. As the technician closes the liquid line valve and turns the air conditioning on, the compressor pushes all the refrigerant into the high-pressure side of the system. Usually, the compressor has low-pressure protection mechanism, e.g., 30 Psig low pressure limit for a typical R-410A system. When the suction pressure gets to this predetermined lower limit, the compressor will be turned off by the unit control.

Most of the vapor compression systems using scroll compressors have been equipped with the low-pressure protection feature. Thus, the termination of the pump down will be executed automatically at a constant low suction pressure without extra cost. At the end of the pump down, the suction pressure is extremely low. And the refrigerant mass flow rate is very small, because the compressor mass flow rate is dictated by the compressor displacement volume and low suction pressure setting (vapor density). At the termination, the vapor compression system reaches near steady-state balance points.

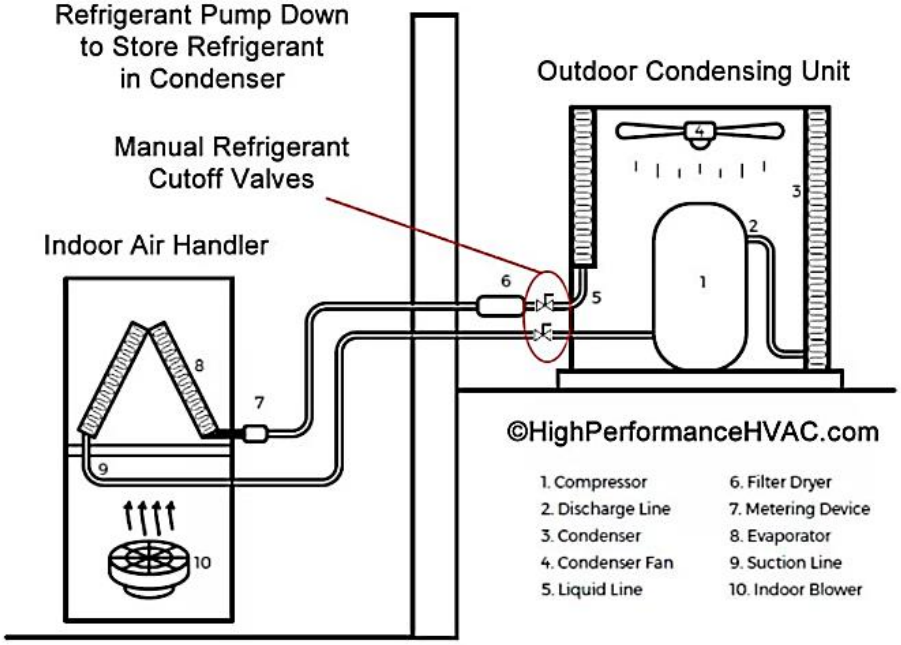

Figure 1: Schematic of Pump Down Operation of Heat Pump

Automatic pump down operation can be conducted by systems equipped with one solenoid cut-off valve at the condenser exit. After the pump down operation, the refrigerant can be released back to low-pressure side by opening the valve. The novel charge fault detection method is realized by analyzing the data recorded by a controller which monitors the compressor and the pressures and temperatures at the suction and liquid line during the pump down operation.

2.2 Proposed Charge Prediction Method based on Theoretical Analysis

There is an advantage of predicting charging using measured data from pump down operation instead of using data from steady state heat pump regular operation. During steady state operation, the amount of refrigerant in refrigerant buffer, i.e., receiver or accumulator, does not circulate in the system. Therefore, the refrigerant stored in the receiver or accumulator, does not affect the sensor measured information. In other words, the system operation information is identical when the receiver is filled with liquid refrigerant or completely empty, as long as the circulated refrigerant is of the same amount. Considering the pump down operation circulates all refrigerant into high-pressure side including the refrigerant in refrigerant buffer, one highlight of this charge prediction technology is its capability to predict the total refrigerant charge including refrigerant in accumulator and receiver. At the end of a pump down operation, the refrigerant status in the high-pressure side is schematized in Figure 2. The tube represents the total refrigerant flow path in the high-pressure side.

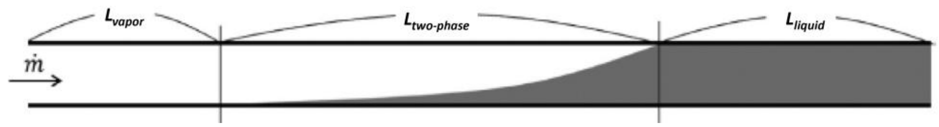

Figure 2: Refrigerant Distribution in High-pressure Side Flow Path

In the refrigerant flow path, the longer a section is occupied by a specific-phase of refrigerant, the greater amount of refrigerant under that phase is stored. The phase-specific refrigerant charge can be calculated as the product of refrigerant average density, the cross sectional area of the path and the length of the path occupied by the specific phase as shown in Equation (1).

$$m_{phaseSpecific} = (\rho_{phaseSpecific} A_{crossSection}) L_{phaseSpecific} \tag{1}$$

For a vapor compression system, the phase-specific average density and the cross-sectional area can be regarded as constants between multiple pump-down operations. Equation (2) is an abstract form of Equation (1) and shows the linear relationship between the refrigerant charge and the length of the path occupied by different phases.

$$\begin{aligned} m_{twoPhase} &= k_1 L_{twoPhase} \\ m_{liquid} &= k_2 L_{liquid} \\ m_{vapor} &= k_3 L_{vapor} \end{aligned} \tag{2}$$

The summation of Equation (2) is the total refrigerant charge as shown in Equation (3).

$$m_{totalCharge} = m_{twoPhase} + m_{liquid} + m_{vapor} \tag{3}$$

Despite the pump down process is a dynamic process, at the end of the operation, the mass flow rate in the compressor changes very slowly and the system can be regarded to reach quasi steady state. And there is very small amount of refrigerant left in the low-pressure side after evacuating the refrigerant. Figure 3 shows the P-h diagram of the quasi-steady state at the last moment of pump down operation.

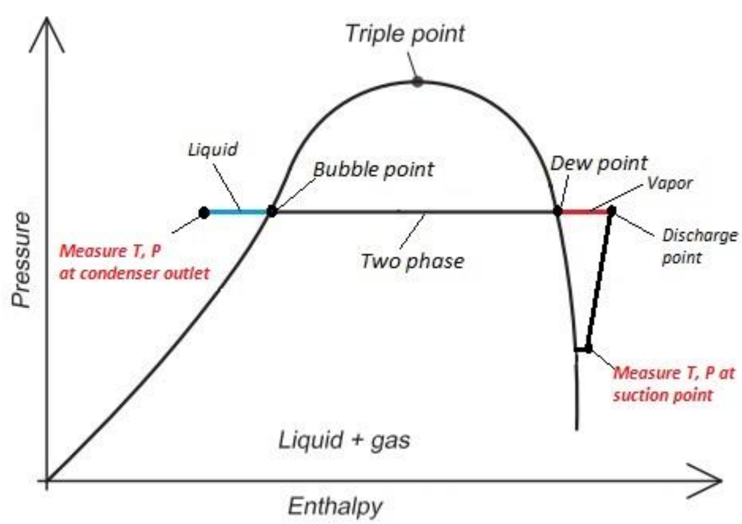

Figure 3: P-h diagram for the Quasi Steady State at the End of Pump-down

Based on the energy conservation, the capacity of each phase can be calculated by the enthalpy difference between the phase transition points, governed by the measured condensing pressure at the condenser exit. Equation (4) shows how to calculate the heat transfer rate. h_{fg} is the latent heat of refrigerant at the condensation pressure, i.e., discharge pressure. $h_{bubblepoint}$, $h_{dewpoint}$, $h_{dischargePoint}$ and $h_{condenserOutlet}$ are the enthalpy at bubble point, dew point, discharge point and condenser outlet, respectively.

$$\begin{aligned} Q_{twoPhase} &= \dot{m} \times h_{fg} \\ Q_{liquid} &= \dot{m} \times (h_{bubblePoint} - h_{condenserOutlet}) \\ Q_{vapor} &= \dot{m} \times (h_{dischargePoint} - h_{dewPoint}) \end{aligned} \tag{4}$$

From the heat transfer point of view, the heat transfer rate between air and refrigerant can be expressed as Equation (5), where $U_{twoPhase}$, U_{liquid}, U_{vapor} are the total heat transfer coefficient per unit tube length for different phase sections, $LMTD_{twoPhase}$, $LMTD_{liquid}$, $LMTD_{vapor}$ are the logarithmic mean temperature difference between the outdoor air and refrigerant of different phase.

$$\begin{aligned} Q_{twoPhase} &= U_{twoPhase} L_{twoPhase} LMTD_{twoPhase} \\ Q_{liquid} &= U_{liquid} L_{liquid} LMTD_{liquid} \\ Q_{vapor} &= U_{vapor} L_{vapor} LMTD_{vapor} \end{aligned} \tag{5}$$

Since the heat transfer coefficient of the refrigerant is much larger than the heat transfer coefficient of air, the air heat transfer coefficient dominates $U_{twoPhase}$, U_{liquid}, U_{vapor}. The dominance of airside thermal resistance is true for forced convection when the fans are on. If the fans are closed during the pump down operation, the airside heat transfer coefficient for natural convention is even smaller than forced convection, and the airside thermal resistance is even more dominant. It is worthwhile to mention that, the variation of heat transfer coefficient of air is very small under different ambient temperature for forced convection Wang et al. (1999) as well as for natural convection Churchill and Chu (1975) in the typical application temperature range of the heat pump systems. Therefore, $U_{twoPhase}$, U_{liquid}, U_{vapor} can be regarded as constants which are not sensitive to the variation of ambient temperature. By equalizing Equation (4) and Equation (5) and rearrange the equations, the phase specific path length can be expressed in Equation (6).

$$\begin{aligned} L_{twoPhase} &= \frac{\dot{m} \times h_{fg}}{U_{twoPhase} LMTD_{twoPhase}} \\ L_{liquid} &= \frac{\dot{m} \times (h_{bubblePoint} - h_{condenserOutlet})}{U_{liquid} LMTD_{liquid}} \\ L_{vapor} &= \frac{\dot{m} \times (h_{dischargePoint} - h_{dewPoint})}{U_{vapor} LMTD_{vapor}} \end{aligned} \tag{6}$$

Plug Equation (6) into Equation (2), then sum up refrigerant charge in different phases as Equation (3), the total refrigerant charge is shown in Equation (7).

$$m_{totalCharge} = k_1 \times \frac{\dot{m} \times h_{fg}}{U_{twoPhase} LMTD_{twoPhase}} + k_2 \times \frac{\dot{m} \times (h_{bubblePoint} - h_{condenserOutlet})}{U_{liquid} LMTD_{liquid}} + k_3 \times \frac{\dot{m} \times (h_{dischargePoint} - h_{dewPoint})}{U_{vapor} LMTD_{vapor}} \tag{7}$$

The constants k_1, k_2, k_3 and constants $U_{twoPhase}$, U_{liquid}, U_{vapor} can be grouped as shown in Equation (8).

$$m_{totalCharge} = (\frac{k_1}{U_{twoPhase}}) \times \frac{\dot{m} \times h_{fg}}{LMTD_{twoPhase}} + (\frac{k_2}{U_{liquid}}) \times \frac{\dot{m} \times (h_{bubblePoint} - h_{condenserOutlet})}{LMTD_{liquid}} + (\frac{k_3}{U_{vapor}}) \times \frac{\dot{m} \times (h_{dischargePoint} - h_{dewPoint})}{LMTD_{vapor}} \tag{8}$$

By lumping the constants in Equation (8), Equation (9) shows the refrigerant charge calculation.

$$m_{totalCharge} = C_1 \times \frac{\dot{m} \times h_{fg}}{LMTD_{twoPhase}} + C_2 \times \frac{\dot{m} \times (h_{bubblePoint} - h_{condenserOutlet})}{LMTD_{liquid}} + C_3 \times \frac{\dot{m} \times (h_{dischargePoint} - h_{dewPoint})}{LMTD_{vapor}} \tag{9}$$

In Equation (9), for a specific heat pump system there are four constants C_1, C_2, C_3 and $m_{totalCharge}$. These four constants are not sensitive to the variation of the ambient condition. They are specific values for that vapor compression system. Therefore, the four constants can be obtained through a few pump-down operations at different ambient conditions.

To be specific, since there are four unknowns, four equations are needed. That is, minimum four pump-down operations under different ambient temperature will be conducted to form four equations. After the four pump-down runs, the total refrigerant charge can be solved. The experiment information required to assign into Equation (9) is obtained by system controller at the end of pump down operation when the system reaches quasi steady state. The constants of C_1, C_2, C_3 can be obtained on the day of the unit installation. Since they are not sensitive to the charge variation, they can be assumed being constants and treated as inputs to a controller of this specific heat pump unit. Later for continuous monitoring, the unit will only need to go through one pump down operation to calculate $m_{totalCharge}$ in Equation (9).

To obtain the average temperature of different phases and the refrigerant enthalpy values required in Equation (9), only the suction state point and the condenser outlet state point are necessary, because the pressure drop inside condenser and pipes are negligible due to very small refrigerant mass flow rate. For the compression process, the refrigerant mass flow rate and compressor power can be obtained using the compressor map from the manufacturer. The compressor discharge temperature can be decided by the energy balance. As a result, all state points in Figure 3 are available.

As stated in previous section, the novel charge FDD method can take advantage of the 'low-pressure protection' capability of the compressor to furtherly ease the data acquisition process. Low-pressure protection indicates that the controller will shut down the compressor at a predetermined low suction pressure limit. This can further ease the experiments by only measuring three variables, i.e., the suction temperature and condenser outlet temperature and pressure. And the pump down operation can be scheduled by the controller to automatically perform at different ambient conditions. Once the charge amount is predicted, the refrigerant charge fault can be diagnosed by comparing the name-plate charge amount suggested by the manufacturer with the measured charge amount.

2.3 Quasi steady-state simulation of pump-down process

To develop and validate the charge identification approach, a detailed quasi steady-state system model to simulate pump down operation is conducted. The quasi steady-state simulation framework is developed using DOE/ORNL Heat Pump Design Model (HPDM), which is a component-based building equipment system modeling platform. As an industry leading system simulation, HPDM integrates expertise in thermodynamics, heat transfer, fluid dynamics, numerical analysis, nonlinear equation solving, and object-oriented programming with optimization tools for the design and control of highly efficient vapor compression systems. Firstly, the system is dissembled into two half systems as shown in Figure 4.

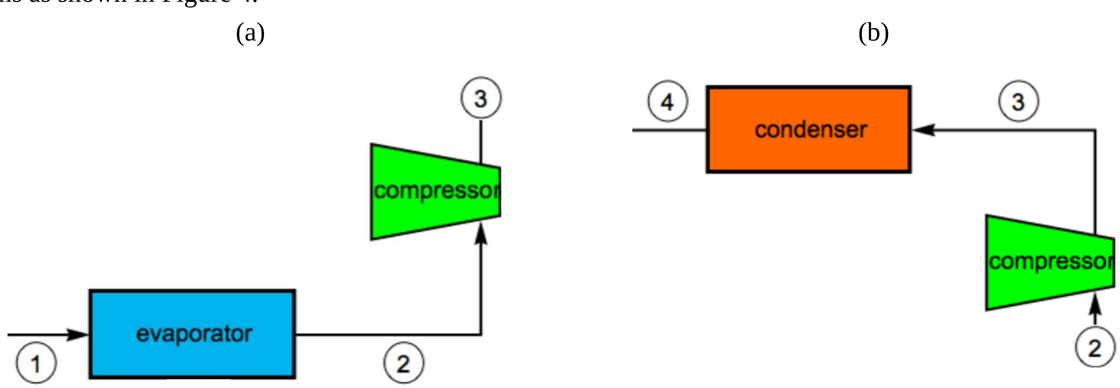

Figure 4: (a) low-side half system; (b) high-side half system.

In the quasi steady-state simulation, the low-side refrigerant temperature is assumed to maintain at indoor temperature during the pump-down process and the refrigerant mass is distributed evenly per component inner volumes at start up. Figure 5 shows the algorithm of quasi steady-state simulation of pump-down operation. The simulation starts by specifying the low-side inlet enthalpy as the saturated liquid enthalpy under room temperature. Then the low-side and high-side half systems are simulated to obtain the charges in both half system. At each time step, by assuming a guessed evaporator inlet enthalpy, the new charges at low side and high side are calculated. The charge increase in high-side is compared to the charge reduction in low-side. And the difference of them are the criteria to adjust the guessed evaporator inlet enthalpy. The algorithm converges when the suction pressure reaches a predefined threshold which is determined by the compressor low-pressure protection mechanism.

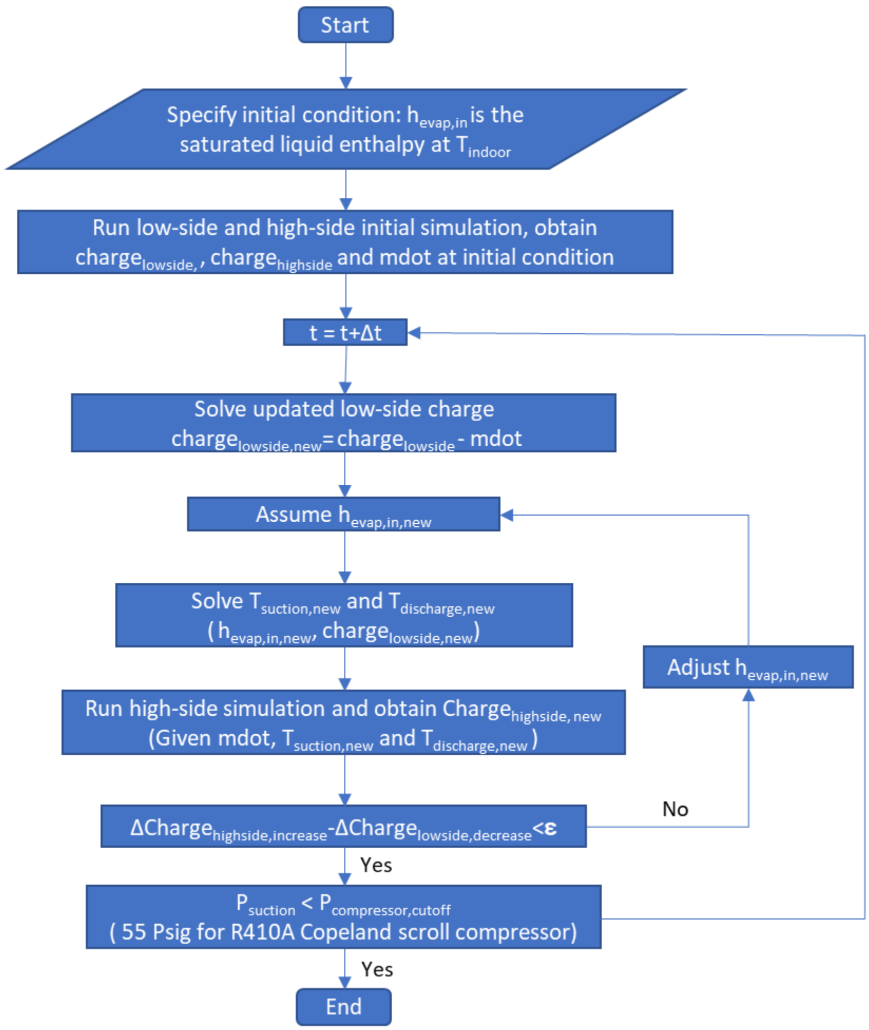

Figure 5: Algorithm Flowchart of quasi steady-state simulation for pump-down operation

3. RESULTS

3.1 Preliminary validation of pump-down prediction method

The charge prediction method was validated using experiment data obtained from laboratory refrigerant leakage test from two residential heat pump systems (Butler et al. (2021)). The measured charge level in the leakage test is shown in Figure 6. Three points highlighted in red circles are sampled from the experiment data to solve the constants C_1, C_2, C_3 in Equation (9) for each of the test heat pump system. The goal is to test whether the fitted Equation (9) can predict other points in leakage tests.

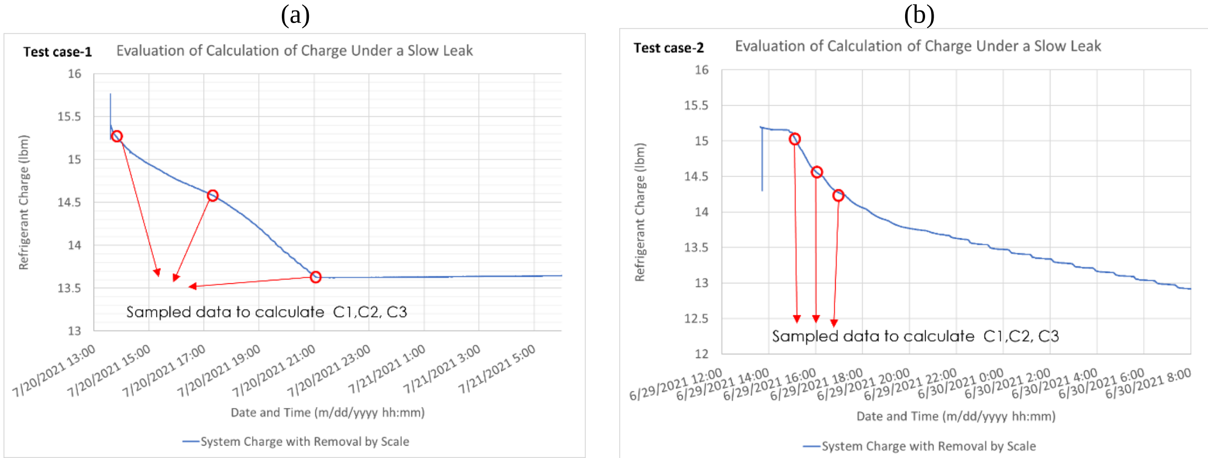

Figure 6: Sampling data to solve the constants for theoretical charge prediction equation (a) Test case-1; (b) Test case-2.

Table 1 shows the solved constants in Equation (9) using the sampled data from Figure 6.

Table 1: Solved constants in charge prediction equation using experiment data

Test Number	C_1	C_2	C_3
Test case-1	3.84	10.37	28.89
Test case-2	6.19	1.68	12.50

Figure 7 shows the prediction of other charge levels using the established charge prediction equation for different test cases. The discrepancy between measured charge and predicted charge is within 8% and 5%, respectively with two different test cases.

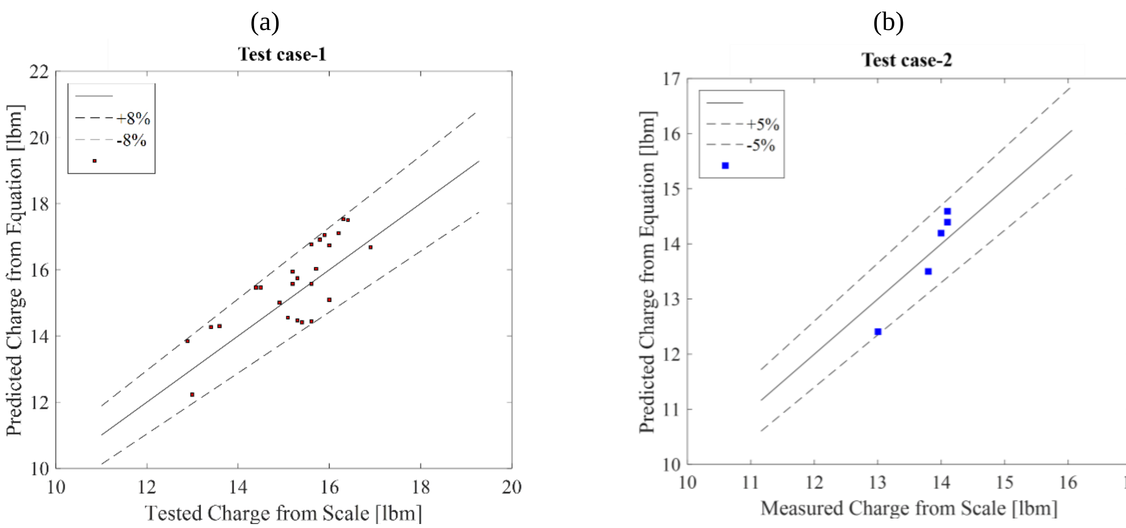

Figure 7: Preliminary Validation of Charge Prediction Method (a) Test case-1; (b) Test case-2.

The quasi steady-state pump-down simulation is conducted on a 5-ton residential heat pump system which has total charge of 13.28 lb. of R410-A. Figure 8 shows the transients of charge migration. Pump-down operation takes 24 seconds until compressor cut-off initiates at 55 psi suction pressure. At the end of 24s, the remaining charge in low-side is 0.69 lb., which is 5.2% of the total charge. these result matches with the observation during laboratory pump-down experiment tests. Detailed experimental validation of the quasi steady-state simulation for pump-down process will be presented in subsequent publications.

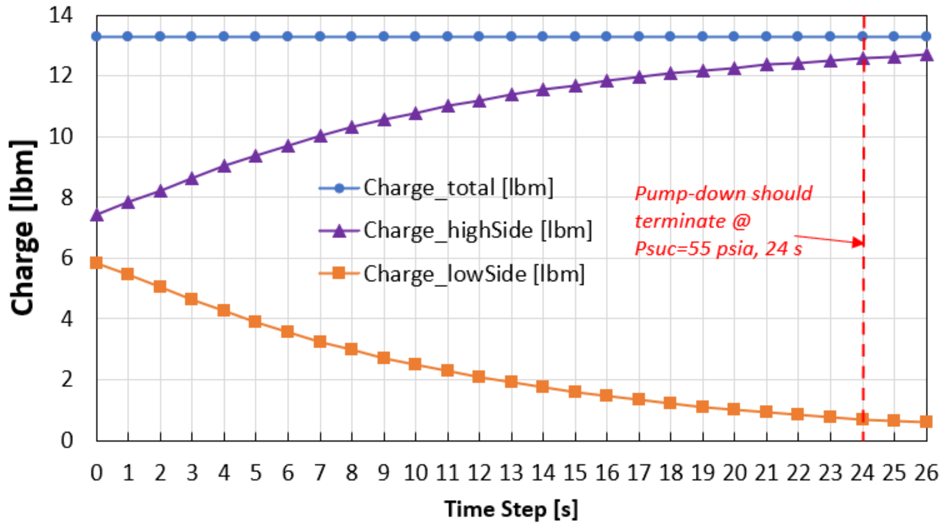

Figure 8: Transients of charge migration

Figure 9 shows the transients of suction pressure, from 0 s to 4 s, the evaporator inlet is in two-phase, so the pressure does not decrease as dramatically as latter session when the entire low side is occupied with superheated vapor.

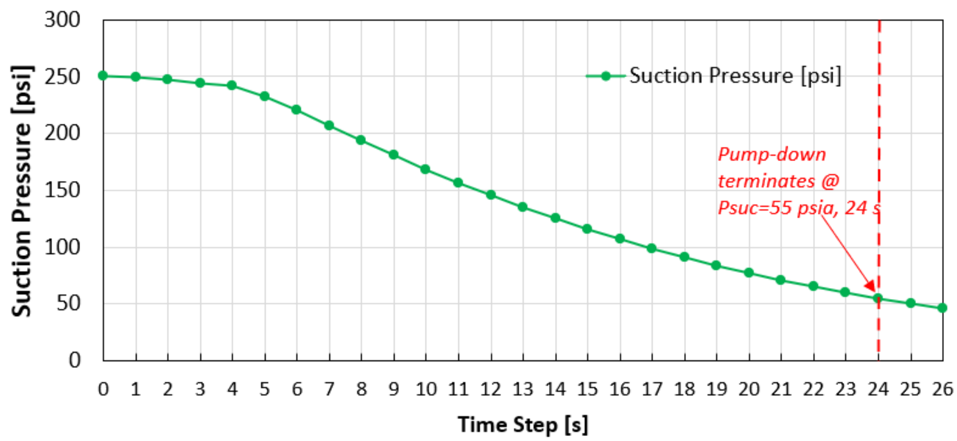

Figure 9: Transients of suction pressure

Figure 10 shows the mass flow rate transients. As indicated, the mass flow rate drops significantly fast after the low-side is superheated.

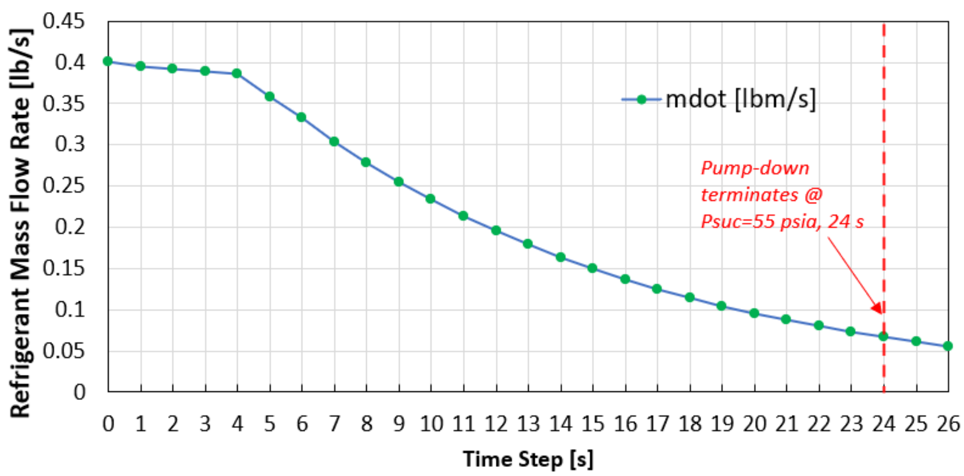

Figure 10: Transients of mass flow rate in compressor

4. CONCLUSION

The performance of the heat pump system varies greatly depending on the refrigerant charge amount. Improving the refrigerant charge fault detection and diagnostics method of vapor compression systems have the potential for increasing energy efficiency and reducing service cost. Previous studies to predict refrigerant charge amount are mostly empirical methods which require significant amount of experimental data for high accuracy. The primary goal of this research is to develop a universal charge fault detection method which requires only a few experimental data with high prediction accuracy.

The proposed charge prediction method is based on running through refrigerant pumping down cycles. With inputting the compressor model number, the fault diagnosis module knows the exact compressor power and mass flow rate map at measured suction and discharge pressure. When pumping down, a valve at the liquid line would close to stop the refrigerant flow, and the refrigerant is pumped by the compressor from the evaporator side to the condenser side. By monitoring the real time suction/discharge pressures and temperatures, the refrigerant mass at the evaporating and condensing sides can be accurately calculated.

As a result, the system charge has been predicted and compared to the nameplate to indicate a charge fault. Preliminary experiment validation shows the charge prediction accuracy is within 8%. The quasi-steady-state system model to simulate the pumping down process successfully can capture the transients of system during pump-down operation and demonstrates the efficacy of the new charge prediction method.

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