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# REFRIGERANT SUBCOOLING CORRELATIONS FOR DIFFERENT EXPANSION DEVICES FOR A NON-INTRUSIVE CHARGE INDICATOR

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**Abstract**

The most common problems affecting residential and light commercial heating, ventilation, and air-conditioning (HVAC) systems are slow refrigerant leaks. Equipment users are usually not aware of the problem until most of the refrigerant has escaped.

A low-cost, non-intrusive refrigerant charge indicator has been developed, based on temperature measurements and correlations formed to interpret the measured temperatures. It can be used to provide real time warnings to the equipment users before the majority of refrigerant is escaped. It could be inexpensive and easy to incorporate into existing heat pumps and air conditioners. Extensive laboratory experimental work was performed on a 2-ton window air conditioner and on a 2.5 ton split heat pump system. It was found that the heat pump was not sensitive to slow refrigerant leak because of the long liquid line. Liquid subcooling was measured to determine the system charge status before a substantial amount of refrigerant was leaked. This study reports the finding of correlations formed for liquid subcooling for the orifice plate and thermal expansion valve used on the heat pump system for both heating and cooling mode operation.

## 1. Introduction

Slow leak on air conditioning and heat pump systems is a common problem for many units in the field, and are usually unnoticed until the majority of refrigerant has leaked out, particularly for split heat pump systems with long liquid refrigerant lines and for air conditioning systems, such as window air conditioners with thermal expansion valves (TXV). However, such leaks can still be detected by measuring the level of liquid subcooling (Domanski, 2006, and Mei, Chen and Gao, 2003) with similar shapes of liquid subcooling curves as functions of system refrigerant charges, even though one tested with a 3 ton system and the other a 2.5 system. This gives some confidence that the method of measuring the heat pump liquid subcooling to detect the system leakage is not very sensitive to the size of the equipment. For systems that are refrigerant charging sensitive, such as the window air conditioners with capillary tube, or orifice, refrigerant expansion devices, the system capacity will be sensitive to charge. A 20% refrigerant leak could reduce the cooling capacity for a window air conditioner, for example, by 50% or more (Mei, Chen, and Gao, 2003). For heat pump heating mode operation, there is practically no published data available on public domain. For TVA region where heat pumps are popular, for example, the system charges are based on summer cooling mode. The systems are overcharged for winter operation. Therefore, using the level of liquid subcooling to detect the system leaks are not as sensitive as that for cooling modem until a substantial amount of refrigerant, 40 to 50% of the charge, is lost. Domanski (2006) also mentioned that the level of liquid subcoolings as functions of system charge for different refrigerants, such as R-22 and R-410A, are similar to each other. This method is then not very sensitive to the type of refrigerants used. However, if the equipment users, usually the residents of the house, will often not be able to sense the problem because the unit will continuously provide cooling, even though at a reduced level.

In this study, a non-intrusive method was developed for system charging status by sensing the refrigerant and ambient temperatures at some selected locations. Based on the laboratory experiments, correlations are formed to reflect the changes of liquid subcoolings for split heat pump systems and for air conditioning units with TXVs. For charging sensitive systems, measuring the two-phase refrigerant temperature on the evaporator coil could possibly be applied.

## 2. Laboratory Test Data for a 2-Ton Window Air Conditioner

Figure 1 shows the test results for the evaporator two-phase refrigerant temperature as a function of the refrigerant charge of an off-the-shelf air conditioner. It is clear that the evaporator coil's two-phase refrigerant temperature is a strong function of the refrigerant charge and a weak function of the outdoor temperature. Therefore, there is a real possibility that the refrigerant charge status can be determined by measuring only one temperature (at the evaporator coil). Measuring both coil and ambient



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temperatures will improve the accuracy, and adding the indoor temperature will provide further improvement. Measuring pressures could be much more expensive, and the error is usually higher. Moreover, pressure measurements require penetration of the refrigeration circuit, opening up the potential for refrigerant leakage from what was a sealed, reliable system. Temperature measurement is inexpensive and accurate.

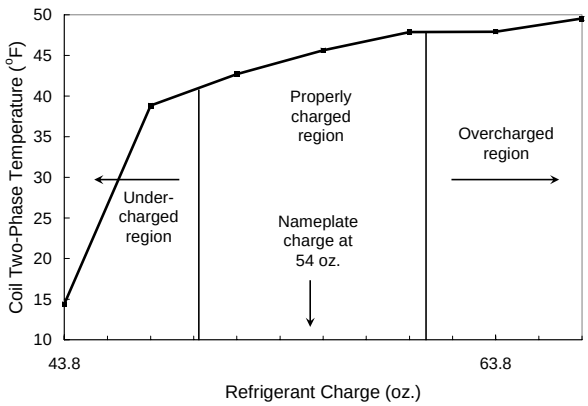


**Fig. 1. R-22 charge vs. evaporator coil temperature for a 2-ton off-the-shelf air conditioner**

### 3. Laboratory Tests on a Split Heat Pump system

Split heat pump systems, because of their long liquid lines, are less sensitive to slow refrigerant leaks. The coil temperature measurement approach, where C1 to C8 represent different u-bend temperature of the C passes (the evaporator has 4 coil passes) of the evaporator, will still work for these systems, as shown in Fig. 2, but only when most of the refrigerant has already leaked out. Figures 3 and 4 show that even when the evaporator coil temperatures are constant in these systems (as shown in Fig. 4), the liquid subcooling continues to drop when the refrigerant leaks slowly. Measurement of the liquid subcooling can be used to determine the status of the system refrigerant charge. It is anticipated that this approach can be used for heat pump heating mode operation, too. In this study, the air flow rate of 600 and 800 cfm for a two ton split heat pump were tested.

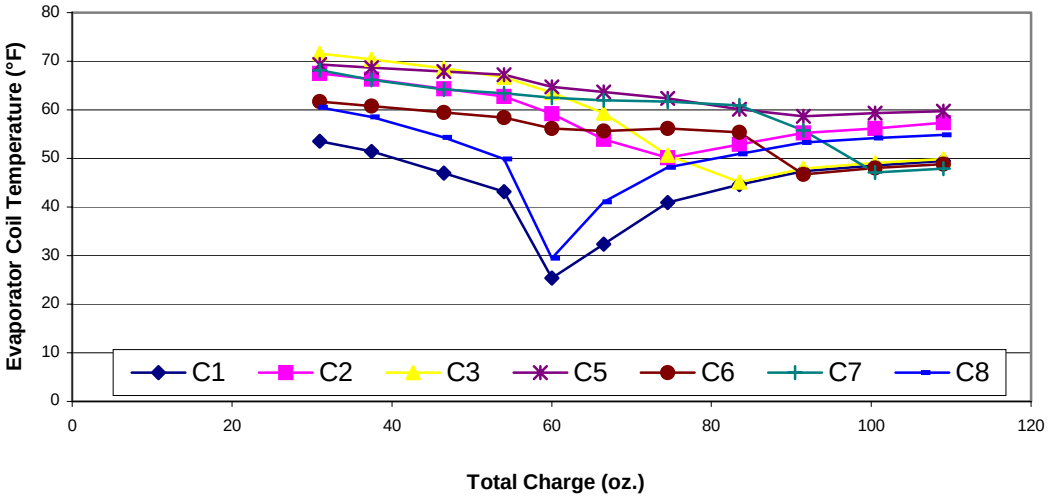


**Fig. 2. Evaporator refrigerant two-phase temperature as a function of refrigerant charge**

### 3.1. Equation for cooling mode operation with an orifice plate

Based on the laboratory test data, refrigerant charge as a function of liquid subcooling cooling can be correlated as follow:

$$R = A \cdot S^4 + B \cdot S^3 + C \cdot S^2 + D \cdot S + E$$

where *R* is a ratio of actual refrigerant charge to name charge; S: subcooling. The applicable range is from 100% to 50% charge.



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$$A = a_1 f_a + a_2;\ B = b_1 f_a + b_2;$$
$$C = c_1 f_a + c_2;\ D = d_1 f_a + d_2;\ E = e_1 f_a + e_2$$

Here $f_a$ is indoor airflow rate in cfm; the detail coefficients listed

| | a | b | c | d | e |
|---|---|---|---|---|---|
| 1 | -1.45833E-07 | 3.13E-06 | -1.5E-05 | -0.00019 | 0.0024042 |
| 2 | 8.89E-05 | -1.25E-03 | -8.21E-03 | 2.90E-01 | -1.57E+00 |

The suitable condition for the empirical equation:

Indoor condition: 80F; 52%RH
Indoor airflow: 600~800cfm
Outdoor condition: 82F ~ 95F

Note that liquid subcooling is a weak function of ambient temperature, as shown in the following figures 3 and 4, where the dotted lines are the plots by the correlations.

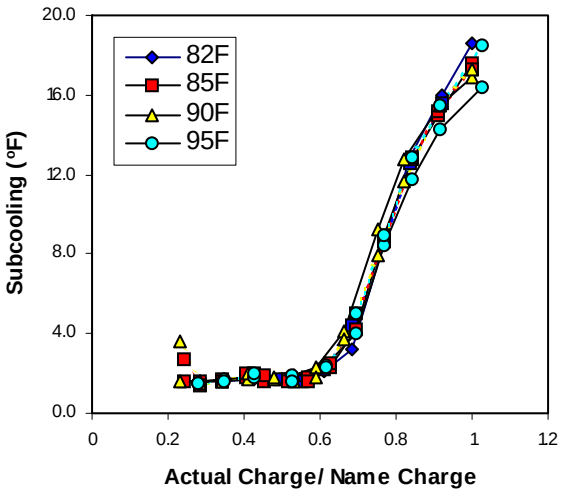


**Fig. 3. Indoor airflow ~ 800 cfm.**

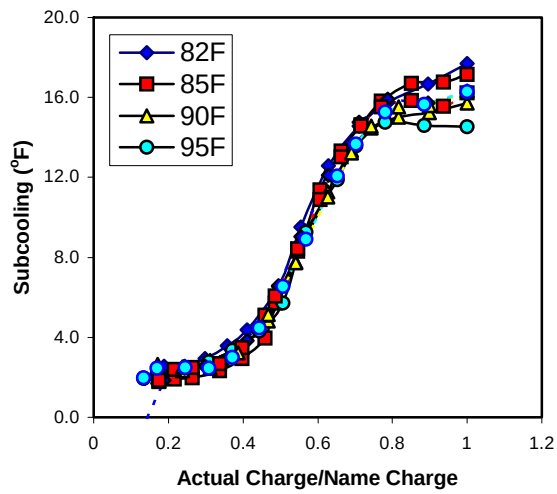


**Fig.4. Indoor airflow ~ 600 cfm.**

While liquid subcooling is a weak function of the ambient temperature, it is a strong function of the amount of refrigerant charge. The reason the level of liquid subcooling at a lower indoor air flow rate is not very sensitive to the refrigerant leak near full charge is that the reduced air flow rate causes the cooling capacity to be reduced, which also reduces the refrigerant flow rate. In other words, the system operates at a surplus of refrigerant charge at a lower airflow rate.

### 3.2. Equation for heating mode test

The original heat pump was equipped with an orifice short tube as the expansion device. The short tube is sturdy and gives the same performance as an expansion valve at the design point, but it is slightly less efficient at off-design operating conditions. After the tests were completed with the orifice plate, a TXV was installed as the outdoor coil expansion device for heating mode tests.

The indoor chamber temperature was set at 70ºF. The outdoor ambient temperatures were set at 47, 42, and 37ºF. The indoor airflow rate ranged from 400 to 800 cfm to examine the effect on heating capacity and indoor air outlet temperature. It was found that when the outdoor ambient temperature was below 35ºF, frost started accumulating on the outdoor coil, which resulted in unstable heat pump performance. The ambient temperature was thus limited to 37ºF.

It should also be noted that because the heat pump was charged for cooling mode operation in the Tennessee region, switching to heating mode operation would result in an excessive refrigerant charge. Under this overcharged operating condition, the system performance was not very sensitive to a refrigerant leak unless a substantial amount of refrigerant had escaped.

#### 3.2.1 Orifice Plate Expansion Devices on Outdoor Coil

Figure 5 shows typical test data with coil temperatures. Clearly the system performance was not affected by the refrigerant leak until 25% of the charge had leaked out, and then other parts of the coil dried out and the one coil, which still has refrigerant in it, temperature started increasing. This was not surprising

because the system was overcharged to start with for heating mode operation. Figure 6 shows a typical liquid subcooling measurement as a function of ambient temperature and refrigerant charge. As in cooling mode operation, the effect of ambient temperatures is relatively small. The dotted lines represent the plots by the correlations.



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With the effect of indoor airflow rate, the liquid subcooling, and the ambient temperature, the amount of refrigerant charge can be curve-fitted into the following equation:

$R$ = exp [(S –B)/A] , where S is the liquid subcooling.

Where $R$ is a ratio of actual refrigerant charge to name charge

$$A = a_1 T_{outdoor}^2 + a_2 T_{outdoor} + a_3$$
$$B = b_1 T_{outdoor}^2 + b_2 T_{outdoor} + b_3$$

Here $T_{outdoor}$ is outdoor temperature; $f_a$ is indoor airflow rate; the detail coefficients are as follows:

$$a_1 = 1.2236e-3 \cdot f_a + 3.246706 - 0.6306512 \cdot \log(f_a)$$
$$a_2 = 3.2345999e-3 \cdot f_a^{1.5} - 3.072797 - 1.107e-4 \cdot f_a^2$$
$$a_3 = 112.93 - 1.037328 \cdot f_a + 1.1605e-3 \cdot f_a^2$$
$$b_1 = -1.0326999e-3 \cdot f_a - 2.953461 + 0.5582497 \cdot \log(f_a)$$
$$b_2 = 4.39e-5 \cdot f_a^{2.4161215} + 8.053967 - 2.275e-4 \cdot f_a^2$$
$$b_3 = -140.5453 - 9.0530002e-4 \cdot f_a^{2.4161215} + 4.6953e-3 \cdot f_a^2$$

where "e" is the exponential term.
The suitable condition for the empirical equation:

Indoor condition: 70ºF; 50% relative humidity. Indoor airflow: 400 ~ 800cfm
Outdoor condition: 37 ~ 47ºF

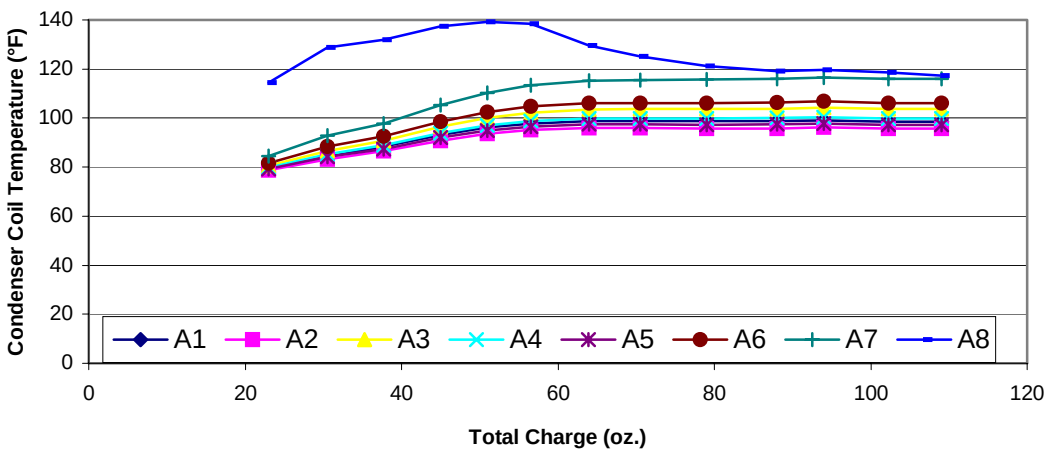


**Fig 5. Outdoor coil temperature (pass A coil) as a function of refrigerant charge at 800 cfm and 47ºF ambient. A1…A8 represent A pass of coil (the outdoor coil has A and B two passes)**



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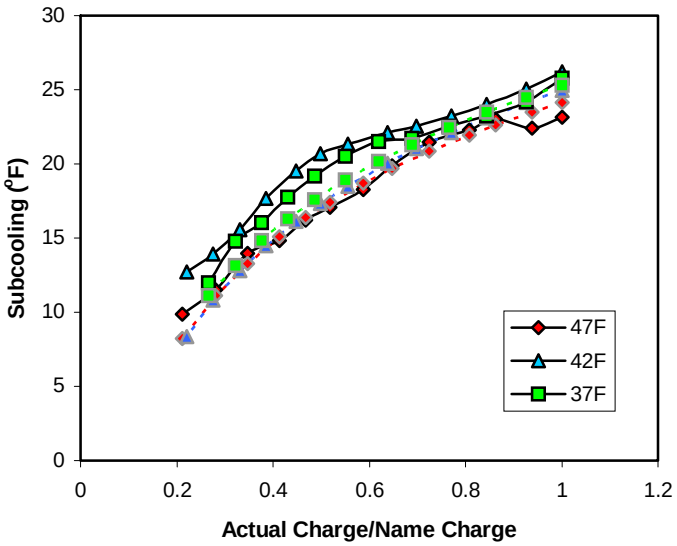


**Fig. 6. Liquid subcooling as a function of refrigerant charge and outdoor temperature at 800 cfm**

The problem of using liquid subcooling method in heat mode to determine the system charge status is that the system is not too sensitive to liquid subcooling at near full charge range. This because in Tennessee region, heat pumps are charged based on cooling mode operation, which resulted in overcharge for heating mode operation. Small leak would not affect the system efficiency at all. Even though this method is not sensitive, we can still use it when the subcooling level drops more than 2°C, for example.

### 3.2.2 Thermal Expansion Valve (TXV) on Outdoor Coil

The heat pump was modified with a TXV as the outdoor coil expansion device for tests in heating mode. Systems with TXVs can tolerate a higher level of refrigerant loss and still perform well. The only way to measure the refrigerant loss, again, is to measure the liquid subcooling, because subcooling is a function of charge regardless of the amount of refrigerant loss. Only the outdoor coil was retrofitted with a TXV to replace the original orifice plate expansion device. Figure 7 shows typical liquid subcooling values as a function of refrigerant charge at 800 cfm indoor airflow rate. At 600 cfm, test data showed that it had the same trend as that of 800 cfm.

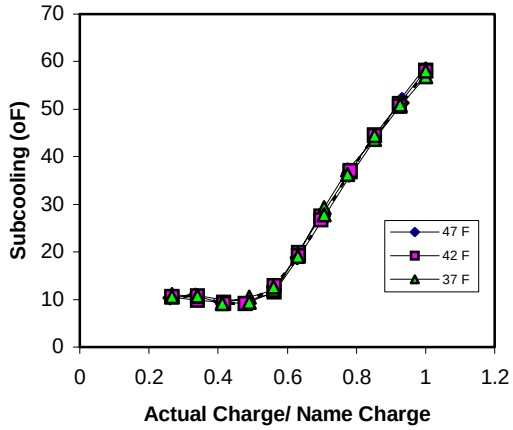


**Fig. 7. Typical heating mode operation with a TXV on outdoor coil, 800 cfm indoor airflow rate**



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When the airflow rate was low during the test, frost started accumulating on the outdoor coil, which caused the performance to become unstable. An airflow rate of 400 cfm was the lowest the heat pump could tolerate without a fast accumulation of frost on the outdoor coil. Again, the test data indicate that the liquid subcooling is a weak function of the ambient temperature. The test data were curve-fitted into the following equation.

$$R = A + B \cdot S$$

Where: *R* is a ration of actual refrigerant charge to name charge. S is subcooling. The applicable range is from 100% to 50% charge.

$$A = a_1 f_a + a_2 ;\ B = b_1 f_a + b_2$$

Here $f_a$ is indoor airflow rate in cfm; the detail coefficients listed

| | a | b |
|---|---|---|
| 1 | 1E-4 | 4E-6 |
| 2 | 0.3444 | 0.0066 |

The suitable condition for the empirical equation:

Indoor condition 70; 50%RH
Indoor airflow: 400~800cfm
Outdoor condition: 37$^o$F ~ 47$^o$F

## 4. Conclusions

A slow refrigerant leak is one of the most common problems associated with air conditioning and refrigeration systems. The traditional way of checking the refrigerant charge status of a system involves measuring high-side and low-side pressures, along with ambient temperature. It requires the use of pressure gauges and at least one temperature sensor. However, most owners or users of HVAC equipment do not have the knowledge to interpret the results of the measurements, even if they have the proper measuring tools. A method of checking a refrigeration or air conditioning system refrigerant charge status by temperature measurements was conceived to resolve these issues. Temperature measurements are accurate, easy to implement, and inexpensive. The test data can be used for the development of a refrigerant charge indicator, with the correlations been used to interpret the refrigerant charging status, based on the measured temperatures.

## 5. References

Domanski, P., 2006, personal conversion with Domanski on 8/11/06 about liquid subcooling measurement as a function of system refrigerant charge.

Mei, V.C., F.C. Chen, and Z. Gao, 2003, Development of Refrigerant Charge Indicator and Dirty Air Filter Sensor, ORNL/CON-489 report, Oak Ridge National Laboratory, Oak Ridge, TN. 37831

## 6. Conversion Factors:

1. Temperature °C = (°F-32)/1.8
2. Subcooling (a temperature differential) °C = °F/1.8
3. Air flow rate L/s (liter/sec) = cfm (cu. ft per minute) * 0.472



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