Subcooling Dependence on Condenser Airflow for Unitary Air Conditioning Systems with TxV Expansion Device

Subtask 2.1B

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Covering Tasks 3 Subtask 2

2013 RESIDENTIAL BUILDING ENERGY EFFICIENCY STANDARDS

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1. Introduction

1.1. This technical brief is presented in response to the proposed low outdoor air temperature protocol (Condenser Outlet Air Restriction Winter Testing protocol) for refrigerant charge verification of TxV systems. The proposed protocol is presented in the CASE report prepared by Wilcox and Proctor and dated December 2010.

1.2. The protocol is based on using subcooling as the refrigerant charge verification method. This method is generally applied with a signal target value applicable to all operating conditions. This includes the implicit assumption that the subcooling variation with operating conditions, in particular outdoor air temperature, is relatively small.

1.3. The proposed protocol for testing at low outdoor air temperature reduces the airflow across the condenser to maintain a pressure drop across the TxV of approximately 160 psi based on the statement that "Bringing the pressure drop across the TXV to at least 160 psi has the same effect as higher test temperatures." This implies the assumption that subcooling is relatively constant when condenser airflow is reduced or alternately that the combination of the reduced outdoor air temperature and reduced condenser airflow produce a relatively small net change in subcooling (relative some reference point). There are published datasets for TxV sets which suggest that neither of these assumptions is valid for all TxV systems.

2. Experimental and Simulation Data for Air Conditioning Systems with TxV

2.1. Data were presented by Temple (2008) showing the variation of subcooling (SC) with outdoor air temperature (OAT) for several TxV systems and are reproduced in Figure 1. System information is provided in Temple (2008) and the source documents identified in the list of References. All the systems have fin-tube condenser coils.

2.2. Data Nomenclature: The performance data are presented primarily in the form of figures with the datasets being labeled as identified in the Glossary. An example dataset designation is "PA_SH_80/67/_". The test system is identified using a two or three character designation (e.g., PA). The dataset is laboratory (experimental) test data unless preceded by the designation "mo" which indicates simulation (model) data. The performance parameter being presented is identified with a two-letter (or three-letter) designation as defined in the Glossary. Performance parameters include Superheat (SH), Subcooling (SC), and Indoor Temperature Difference (ITD). The test condition is identified as 80°F indoor air dry-bulb temperature and 67°F indoor air wet-bulb temperature in this example. A "_" in the outdoor air temperature location indicates a parametric study in outdoor air temperature. Other parametric studies are identified as indicated in the Glossary, e.g., OCFM indicates a parametric study in outdoor (condenser) airflow.

2.3. The data in Figure 1show a general trend of increasing SC with decreasing OAT. For the OA3 system the SC increases approximately 1.3°F from 95°F to 55°F OAT. The

experimental trend for the PA system is not consistent. For the NI system the SC would increase approximately 3.4°F from 95°F to 55°F OAT.

2.4. Available data for subcooling variation with condenser (outdoor) airflow were assembled for the same systems and are presented in Figure 2 (OA3 system), Figure 3 (NI system), and Figure 4 (PA system). The OA3 data are for an outdoor air temperature of 65°F and show a trend of decreasing SC with decreasing condenser airflow. The NI data are for an outdoor air temperature of 100°F and show a trend of increasing SC with decreasing condenser airflow. The PA data are for an outdoor air temperature of 95°F and do not show a consistent trend with decreasing outdoor airflow. Note: For systems NI and PA the condenser airflow was measured and "Relative Outdoor Airflow" represents the relative airflow based on the nominal value. For system OA3 the condenser airflow was not measured and "Condenser Free Coil Area" represents the part of the coil that was not blocked.

2.5. Model benchmarking and simulation data were presented by Temple (2008) using the DOE/ORNL Heat Pump Design Model MarkVII from ORNL (Rice 1997). Additional simulations were performed to investigate subcooling behavior with outdoor airflow and are presented in Figure 5 and Figure 6. The data in Figure 5 indicate an increase in predicted subcooling from 12.7°F to15.6°F (+2.9°F) with a decrease in OAT from 95°F to 55°F. At 55°F and the nominal outdoor airflow the TxV pressure drop is 98 psi. The data in Figure 6 show the performance as the outdoor airflow is decreased (at 55°F OAT). The 43% airflow corresponds to a TxV pressure drop of 166 psi. From the nominal outdoor airflow rate to 43% airflow the predicted subcooling increased from 15.6°F to 19.5°F (+3.9°F). If the reference point is 95°F, then subcooling for the low temperature test condition is 6.8°F high (19.5°F compared to12.7°F). If the reference point is for the nominal outdoor airflow and an OAT that results in a TxV pressure drop of approximately 160 psi (75°F), then subcooling for the low temperature test condition is 4.8°F high (19.5°F compared to14.7°F).

3. Conclusions

3.1. In general the observed trend is for subcooling to increase with decreasing outdoor air temperature and also to increase with decreasing airflow; although, these trends are not consistent for all TxV systems investigated. The data presented do not support the assumption that subcooling remains constant with reduced outdoor airflow or with the combination of reduced outdoor air temperature and reduced outdoor airflow.

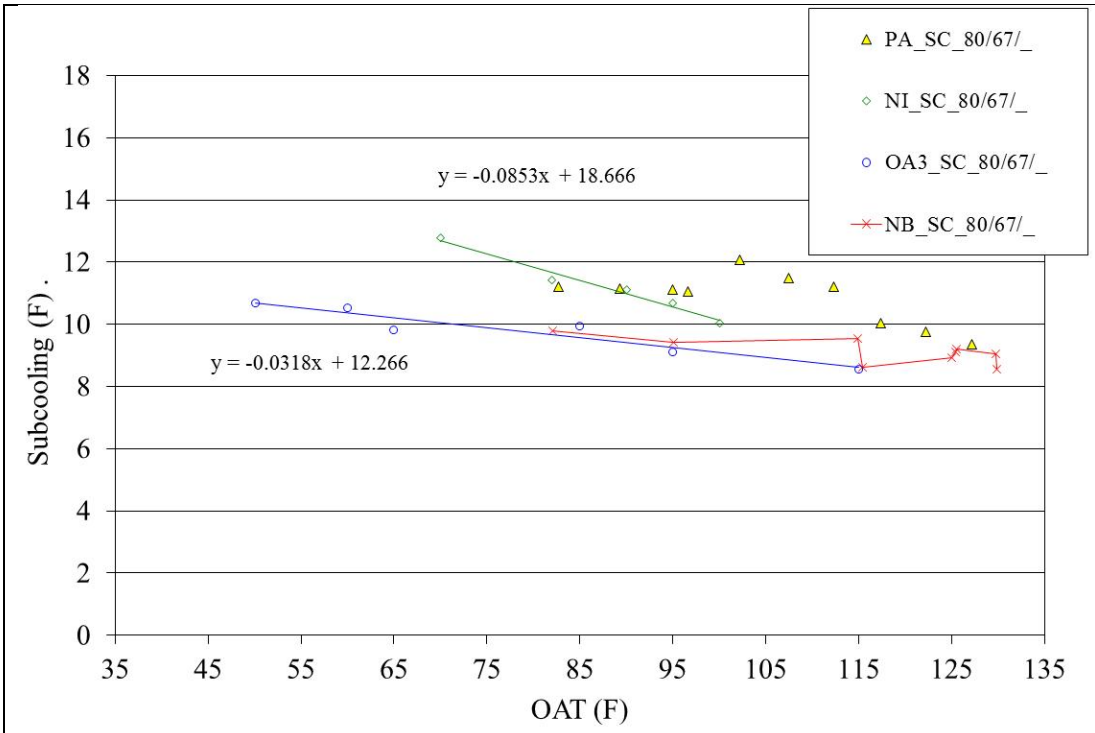

Figure 1. Subcooling variation with OAT – Experimental data, TxV

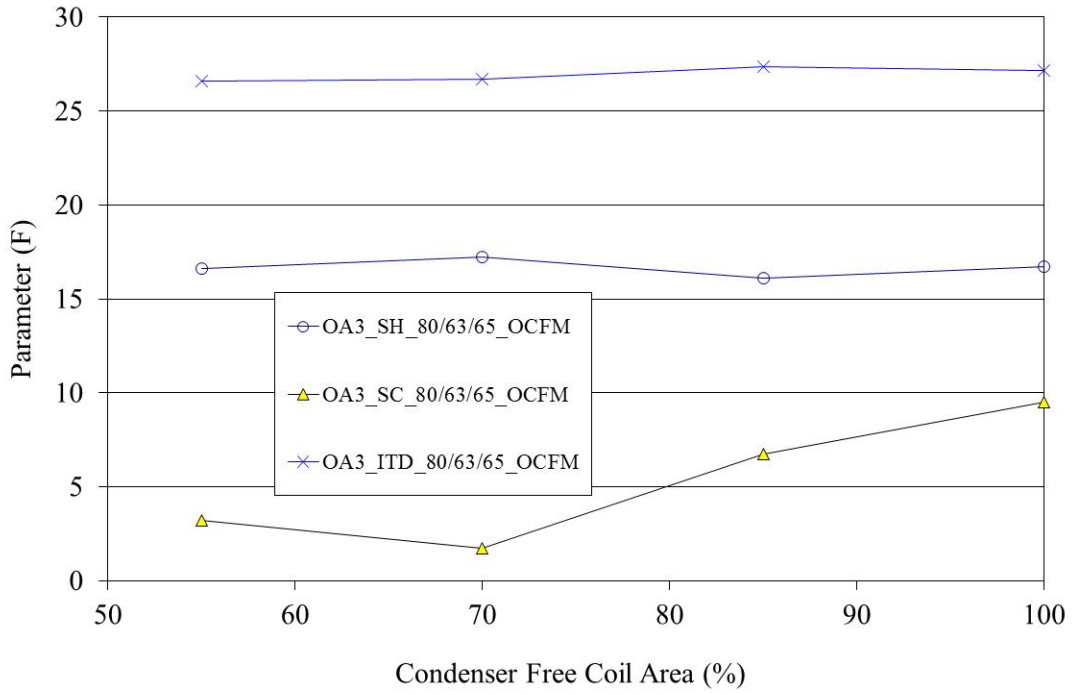

Figure 2. Performance Variation with Outdoor Airflow – Experimental Data, OA3 TxV

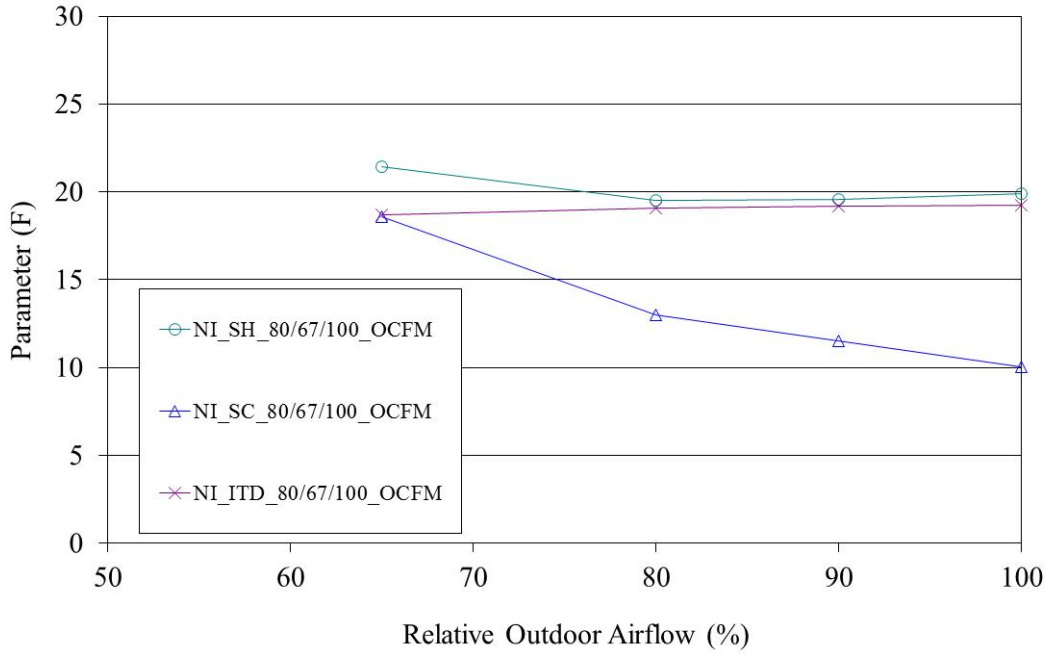

Figure 3. Performance Variation with Outdoor Airflow – Experimental Data, NI TxV

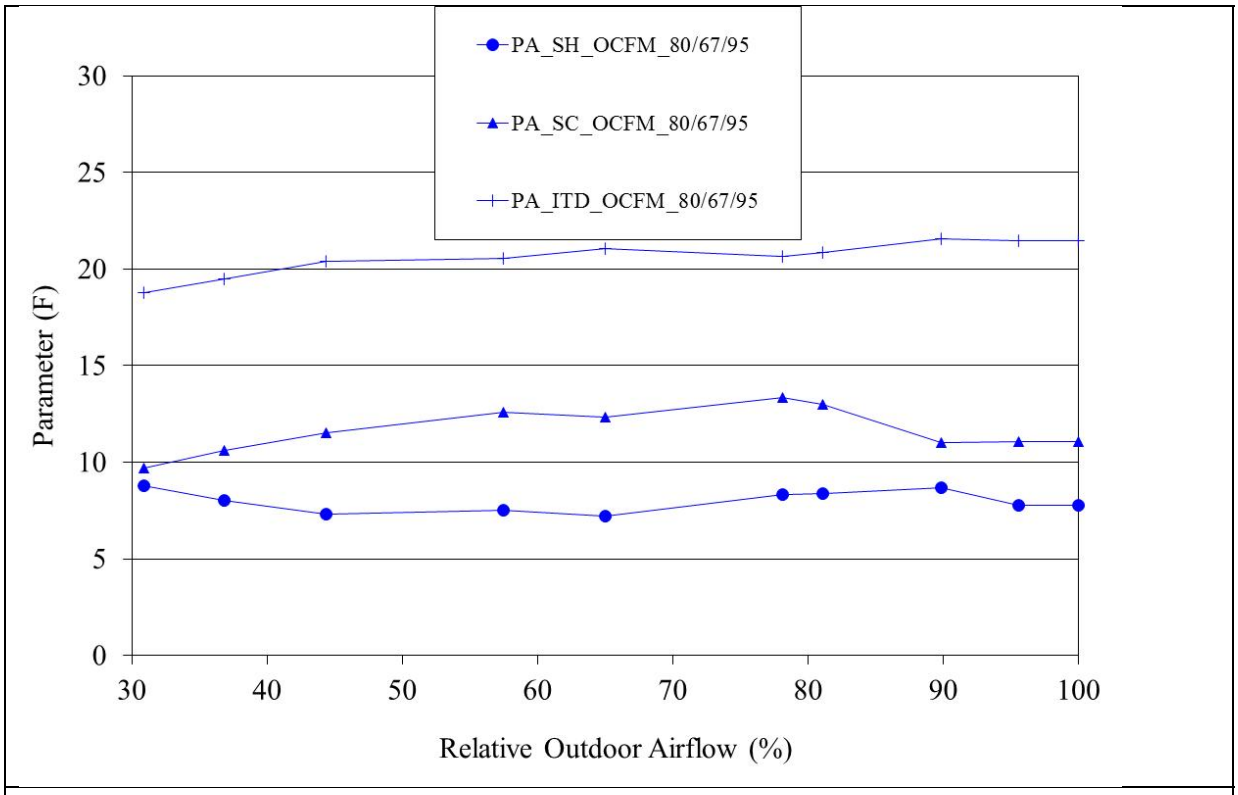

Figure 4. Performance Variation with Outdoor Airflow – Experimental Data, PA TxV

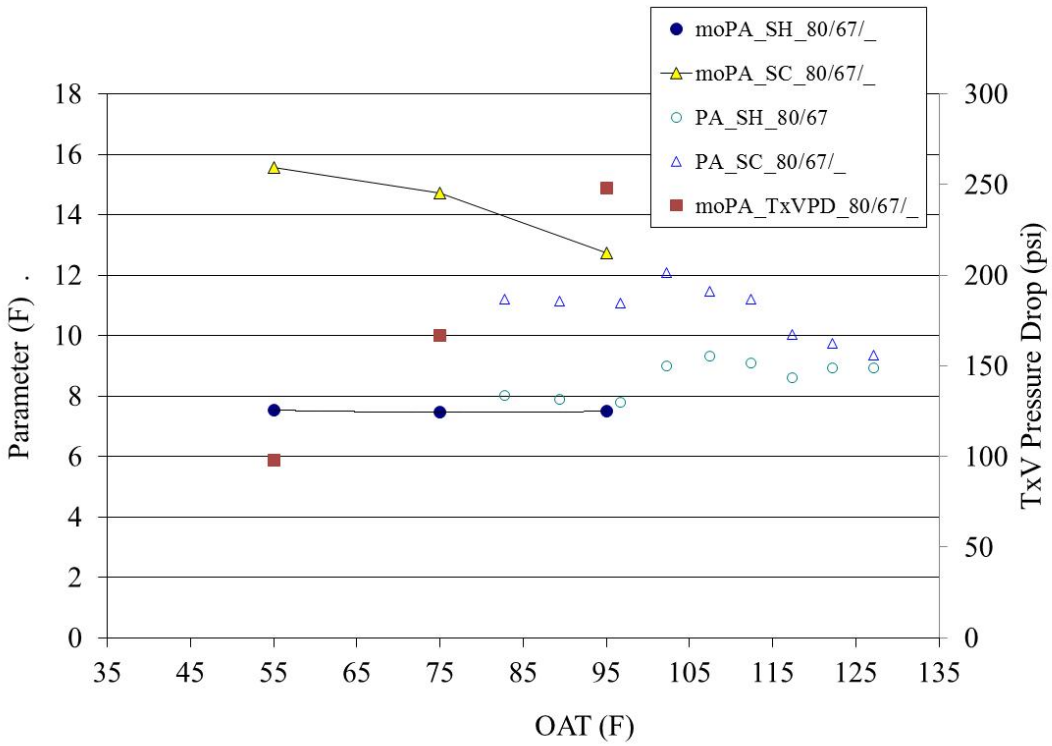

Figure 5. Performance Variation With OAT – Simulation and Experimental Data, PA TxV

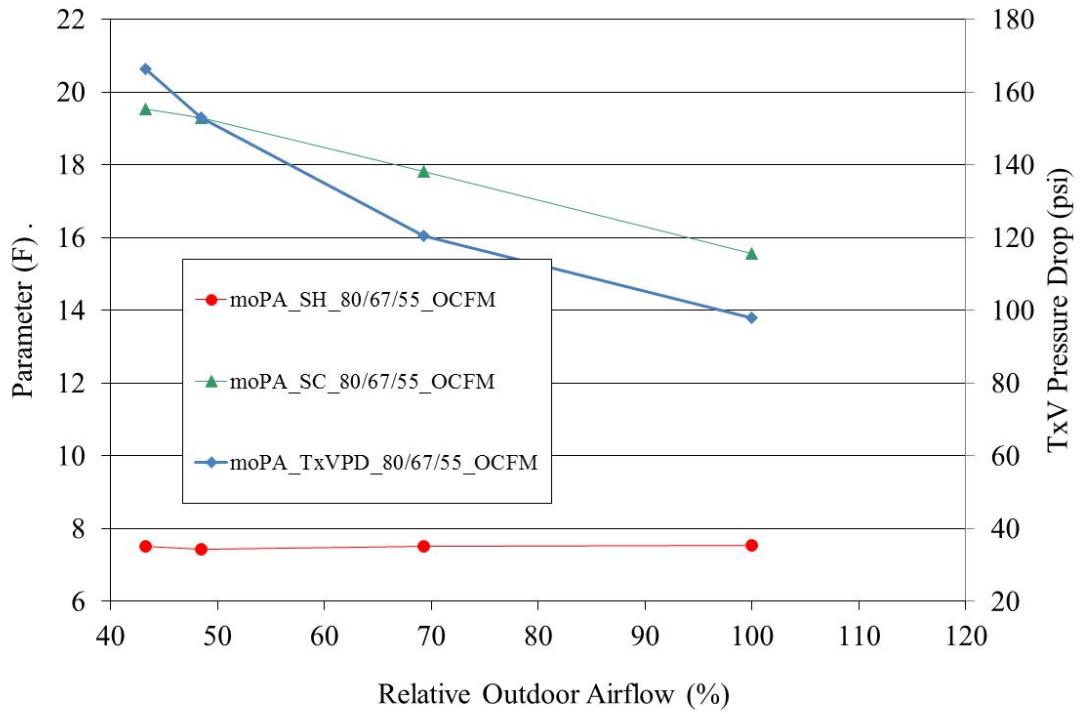

Figure 6. Performance Variation With Outdoor Airflow – Simulation Data, PA TxV

4. Glossary

Parameter	Description	English Units	SI Units
AC	Air Conditioner		
CT	Refrigerant Condensing Temperature based on saturation temperature for LP	F	C
ET	Refrigerant Evaporating Temperature based on saturation temperature for SP	F	C
ICFM	Indoor (evaporator) Airflow	ft^3/min (cfm)	L/s
ITD	Indoor air Temperature Difference (RDB-SDB), corresponds to gross capacity (does not include fan heat)	F	C

Parameter	Description	English Units	SI Units
LP	Refrigerant Liquid Pressure	psia	kPa
OAT	Outdoor Air dry-bulb Temperature	F	C
OCFM	Outdoor (condenser) Airflow	ft^3/min (cfm)	L/s
RDB	Return (indoor) air Dry-bulb Temperature	F	C
RWB	Return (indoor) air Wet-bulb Temperature (entering evaporator)	F	C
SC	Refrigerant Subcooling (condenser exit)	F	C
SDB	Supply air Dry-bulb Temperature (leaving evaporator, does not include fan heat)	F	C
SH	Refrigerant Superheat (compressor suction)	F	C
SHR	Sensible Heat Ratio	none	
SP	Refrigerant Suction Pressure	psia	kPA
TS	Temperature Split, dry-bulb temperature difference from return plenum to supply plenum (RDB-ADB), includes the effect of fan heat $TS \approx ITD - 1.1°F$	F	C
TxV	Thermal expansion valve		
TxVPD	TxV pressure drop	psi	
80/67/95	Test (or simulation) condition designation – example RDB/RWB/OAT	F	C
moPA_SH_80/67/95_Charge	ORNL model results – example mo indicates ORNL model data PA indicates system SH indicates parameter 80/67/95 indicates test condition Charge, ICFM, or OCFM indicates corresponding parametric study		
PA_SH_80/67/95_Charge	Laboratory (experimental) data Charge parametric study		

5. References

5.1. Domanski, P.A. and Payne, W.V. 2002. Properties and Cycle Performance of Refrigerant Blends Operating Near and Above the Refrigerant Critical Point, Task 2: Air Conditioner System Study, Final Report, ARTI-21CR/605-50010-01-Pt. 2, NIST, October 2002. [NIST, 2002; NB data]

5.2. Kim, M. Payne, W.V. and Domanski, P.A. 2006. Performance of a Residential Heat Pump Operating in Cooling Mode with Single Faults Imposed. Report NISTIR-7350, NIST, September 2006. [NIST 2006; NI data]

5.3. Rice, C.K. 1997. "DOE/ORNL Heat Pump Design Model, Overview and Application to R-22 Alternatives," 3rd International Conference on Heat Pumps in Cold Climates, Wolfville, Nova Scotia, Canada, Aug. 11-12, 1997; Caneta Research, Inc., Mississauga, Ontario, Canada, November, pp.43-66.

5.4. Shen B., Groll E. A., Braun J. E., 2006, Improvement and Validation of Unitary Air Conditioner and Heat Pump Simulation Models for R-22 and HFC Alternatives at Off-Design Conditions, Final Report for 1173-RP, ASHRAE. [Shen 2006; PA data]

5.5. Temple, K. 2008. Expanded Test Protocols for Low Ambient Testing of Unitary AC Systems. BERG Project Final Report. Grant 54921 A/06-09B. September 2008. [Temple 2008]