



# **Airside HVAC BESTEST: Adaptation of ASHRAE RP 865 Airside HVAC Equipment Modeling Test Cases for ASHRAE Standard 140**

## **Volume 1: Cases AE101 – AE445**

J. Neymark  
*J. Neymark & Associates*

M. Kennedy  
*Mike D. Kennedy, Inc.*

R. Judkoff  
*National Renewable Energy Laboratory*

With:

J. Gall and D. Knebel  
*AAON, Inc.*

R. Henninger and M. Witte  
*GARD Analytics, Inc.*

T. Hong  
*Lawrence Berkeley National Laboratory*

T. McDowell  
*Thermal Energy System Specialists*

D. Yan and X. Zhou  
*Tsinghua University*

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This project built on ASHRAE Research Project 865 (RP 865) by Gren Yuill and Jeff Haberl, published in 2002. Both RP 865 authors attended several SSPC 140 working group meetings for the project and graciously provided additional supporting material from the original research and valuable comments, especially relating to quasi-analytical solution development and the original test specification.

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- Kentaro Kimura, Takasago Thermal Engineering Co., Ltd, Japan: LCEM
- Harunori Yoshida, Okayama University of Science, Japan: LCEM

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## Electronic Media Contents

Airside HVAC Equipment Modeling Tests (Airside HVAC BESTEST), Cases AE101 – AE445. These accompanying files are included in the .zip file link on the cover page of the posted electronic version of this document; click on “Link to Accompanying Zipped Data Files (50 MB)”.

- **README-Airside-HVAC.docx:** Electronic media contents

### *\Part I subfolder*

*Files apply as they are called out in the test procedure.*

- **AE101.TM2:** TMY2 weather data for -29.0°C/-29.0°C constant ODB/ODP
- **AE103.TM2:** TMY2 weather data for 15.5°C/-3.0°C constant ODB/ODP
- **AE104.TM2:** TMY2 weather data for 26.9°C/22.1°C constant ODB/ODP
- **AE105.TM2:** TMY2 weather data for 24.9°C/2.4°C constant ODB/ODP
- **AE106.TM2:** TMY2 weather data for 23.0°C/20.9°C constant ODB/ODP
- **Sec5-5out.xls:** Raw output data spreadsheet

### *\Part II subfolder*

*Supporting files for the Quasi-Analytical Solution (QAS), and its development.*

- **QASv2.xlsm:** QAS calculations.
- **QAS-Results-FCSZ.xlsm:** Results documenting progression and resolution of QAS results from originating PSU and TAMU solution results, FC and SZ systems
- **QAS-Results-CVVV.xlsm:** Results documenting progression and resolution of QAS results from originating PSU and TAMU solution results, CV and VAV systems
- **\RP-865-Spreadsheets subfolder:** Spreadsheets developed for the original ASHRAE RP 865 project. (Original file dates not shown in folder – see file properties details for dates.)
  - \PSU
  - \TAMU (Included .xls files are updates by TAMU to their original files.)

### *\Part IV subfolder*

*Supporting files for the example simulation results, and their development.*

- **Results-FCSZ.xlsm:** Results workbook for FC and SZ systems, to assist users with plotting their results versus the QAS and example simulation results.
  - See tab “Readme” for navigation instructions
  - See tab “Adding Results” for instructions about adding new results.
- **Results-CVVV.xlsm:** Results workbook for CV and VAV systems, to assist users with plotting their results versus the QAS and example simulation results.
  - See tab “Readme” for navigation instructions
  - See tab “Adding Results” for instructions about adding new results.
- **\InpFiles-Airside-HVAC subfolder:** (ASHRAE SSPC 140 Airside HVAC Working Group participant simulation input files)
  - \DEEAP
  - \DeST
  - \DOE22
  - \EnergyPlus
  - \IES-VE
  - \LCEM
  - \TRNSYS

## Executive Summary

This report documents a set of diagnostic analytical verification cases for testing the ability of whole building simulation software to model the air distribution side of typical heating, ventilating and air conditioning (HVAC) equipment. These cases complement the unitary equipment cases included in American National Standards Institute (ANSI)/American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) Standard 140, *Standard Method of Test for the Evaluation of Building Energy Analysis Computer Programs*, which test the ability to model the heat-transfer fluid side of HVAC equipment.<sup>1,2,3,4</sup> This new work was conducted by the National Renewable Energy Laboratory (NREL) in collaboration with a working group of international experts associated with the ASHRAE Standing Standard Project Committee (SSPC) 140. SSPC 140 is responsible for ANSI/ASHRAE Standard 140. The work builds off of ASHRAE Research Project 865 (RP 865).<sup>5</sup> RP 865 developed a test specification, and spreadsheet solutions intended as quasi-analytical solutions, for a number of typical HVAC configurations such as constant volume and variable air volume reheat systems. At the time RP 865 was developed, the scope for input descriptions in its test specification was limited to two prominent whole-building energy simulation programs. The new work, presented in this report, defined the test cases and specifications such that most building energy simulation programs with time-steps of one hour or less can perform the tests, and completed the process of verifying and reconciling the quasi-analytical solutions.

## Background

The development of practical procedures and data for tool evaluation and improvement is part of an overall validation methodology that NREL,<sup>6,7,8</sup> the International Energy Agency (IEA),<sup>9,10,11</sup> and ASHRAE<sup>12,13,14</sup> have been developing for many years. The methodology combines empirical validation, analytical verification, and comparative analysis techniques. Details are discussed in the Background section of HVAC BESTEST Volume 1,<sup>2</sup> with the most recent updates, as of this writing, published in the *2013 ASHRAE Handbook of Fundamentals*<sup>13</sup> and elsewhere.<sup>8,14</sup> NREL originally developed the BESTEST method in IEA Solar Heating and Cooling Programme Task 12 to test building thermal fabric (envelope) models and to diagnose sources of predictive disagreements.<sup>15</sup> This method of testing was adopted with some refinements by ASHRAE in accordance with procedures of ANSI, and now forms the basis for ANSI/ASHRAE Standard 140, *Method of Test for the Evaluation of Building Energy Analysis Computer Programs*.<sup>1,16,17,18,19</sup>

Since Standard 140 was first published, three HVAC BESTEST test suites developed within IEA Solar Heating and Cooling Programme, Task 22 have been added: two that address unitary space cooling equipment<sup>2,3</sup> and one that addresses fuel-fired furnaces.<sup>4</sup> Other test suites that have been added to Standard 140 include building thermal fabric in-depth test cases for ground-coupled slab models<sup>20</sup> and building thermal fabric test cases applicable to more simplified analysis programs commonly used for Home Energy Rating Systems.<sup>21</sup> The new in-depth airside HVAC equipment model test cases described in this report are also planned for inclusion in Standard 140.

## Importance of the HVAC Air-Distribution System Modeling Problem

Most buildings in the United States, especially larger buildings with substantial space cooling loads, have ducted HVAC air distribution systems. Therefore, practical energy simulation models of larger buildings require the presence of ducted HVAC air distribution systems, either for modeling a typical planned building design, or for developing comparative energy savings predictions for an advanced HVAC system (e.g., chilled beams) versus a more typical ducted air distribution system.

As noted above, HVAC BESTEST cases originally published by NREL in 2002<sup>2</sup> and 2004,<sup>3</sup> and Natural Resources Canada in 2003,<sup>4</sup> were previously added to ASHRAE Standard 140.<sup>1</sup> These are analytical verification and comparative cases that test the ability to model unitary space cooling and space heating equipment. For the space cooling systems, these cases test the ability of programs to model behavior of the working fluid side of the system, using manufacturer design data presented as empirically derived performance maps. Many whole-building energy simulation programs are designed to work with this type of data because there is very little manufacturer's data that would support the alternative of first principles modeling. Similarly, the space heating cases test the operation of a furnace based on summary manufacturer performance data. These test suites only address air distribution systems superficially, and the original final reports for the HVAC BESTEST procedures indicate that complementary test cases for typical HVAC air distribution systems are needed. This means that if a model has good agreement for the current set of working-fluid-side mechanical equipment test cases, phenomena specific to airside HVAC distribution systems are not necessarily being correctly modeled.

The work of ASHRAE RP 865,<sup>5</sup> which was published in 2002, developed test cases that address airside HVAC distribution system modeling and are complementary to the current HVAC BESTEST cases. RP 865 developed a set of test cases with two independently developed external spreadsheet solutions to evaluate HVAC air distribution system models utilized by building energy analysis computer programs. The test cases focus on system airflow, and heat and mass balance. The cases are steady-state tests done at constant zone and ambient conditions. The test cases address seven different types of air handling systems, with testing conducted at six different sets of steady state outdoor and zone conditions, and with various economizer outdoor air control strategies. The external spreadsheet solutions were initially produced independently by two different analysts during the original RP 865 project work for all the test case configurations. All the spreadsheet calculations are based on the laws of conservation of mass and energy and on the properties of air and water presented in the *1993 ASHRAE Handbook of Fundamentals*.<sup>22</sup>

### ***The Current Airside HVAC Distribution System Diagnostic Test Cases***

This project adapted ASHRAE RP 865<sup>5</sup> described above. The adaptation, as described in Part I of this report, includes a set of 24 diagnostic test cases for airside HVAC distribution system models. The test cases are summarized in Tables B-1 through B-8 of Appendix B. These are steady-state analytical verification tests, where simulation results are compared to a quasi-analytical solution (secondary mathematical truth standard as defined in ASHRAE Standard 140<sup>1</sup>), and may also be compared to other example simulation results. All results are provided in Part IV. The importance of the quasi-analytical solution as a benchmark for comparing simulation tools is discussed in Part III, Section 3.5.1.

The test cases are a subset of those developed as part of RP 865. The original research provided a technically sound conceptual starting point for the test specifications along with two separately developed solution spreadsheets intended as quasi-analytical solutions (QASs). However, from this starting point NREL had three main technical challenges: a) to reconcile differences in the two analytical solutions into a single final QAS, b) to rework the test specifications such that they would be unambiguous for the input structures of most whole building energy simulation programs with time steps of one hour or less, and c) to field test the usability of the test specifications with a variety of different simulation programs and associated software development groups around the world.

After we began to adapt the test specification, it became apparent that some of the assumptions included in the RP 865 spreadsheet solutions were not clearly stated in the original RP 865 test specification, and these details needed to be brought forward, clarified, or expanded in the final version of the specification. A summary of the process of improving the test specification is described in Part III, Section 3.3, along with a summary topical listing of the improvements. There were also a few substantive technical revisions



to the original RP 865 test specification and revisions to the original external spreadsheet solutions leading to the final merged quasi-analytical solution. The process of vetting the QAS is described in Part II, Sections 2.2.2 and 2.3.

The test systems include the following: Four-Pipe Fan Coil (FC), Single-Zone Air Conditioner (SZ), Constant Volume Terminal Reheat (CV), and Variable Air Volume Reheat (VAV). The FC system is the simplest of the RP 865 systems: it is a single-zone system with heating and cooling coils, zone air exhaust, and limited outdoor air (no economizer control), and it does not include a return air fan. The FC system provides a good starting point for testing basic mass flow and heat balance modeling before addressing more complex air systems. In these test cases the SZ system adds an economizer, and a return air fan; the CV system further applies multiple (two) zones, system supply air temperature control, and terminal reheat coils; and the VAV system further applies a variable airflow supply fan and terminal zone supply air dampers. The final test cases are conducted at five different sets of steady state outdoor and zone conditions in heating, dry-coil cooling, and wet-coil cooling modes, and with temperature and enthalpy economizer outdoor air control strategies applied to selected conditions. Primary compared output for these test cases includes coil sensible, latent and total loads; zone sensible and latent loads; and cooling-coil leaving-air relative humidity. Additional diagnostic outputs at various points in the systems include dry-bulb temperature (and the ability to isolate fan heat effects), humidity ratio, specific volume, enthalpy, and mass flow rate. For these in-depth cases, plant energy use related to coil loads and fan electricity consumption is not considered.

NREL led the collaborative effort to vet and extensively revise the test case specifications for external use as a standard method of test that can be integrated with the tests of ASHRAE Standard 140. The collaboration included a number of software developer members of SSPC 140 along with other international software developers and participants. NREL also vetted the original RP 865 external spreadsheet solutions, and merged them into a single QAS. The vetting process followed the procedure for developing analytical and quasi-analytical solutions defined by NREL in previous work<sup>2</sup> and involved checking all original equations by a third independent analyst. Spreadsheet results disagreements from the original work were reconciled by the third analyst; where needed, some details were reconciled in collaboration with the original solution developers. The vetting process also included comparison with previous solutions as QAS versions progressed. Details of QAS development are described in Part II. The QAS also has good agreement with two of the simulation models that were able to most closely match the assumptions of the QAS, and observable differences between the QAS and the other simulation programs were explainable by modeling assumptions of those programs, consistent with the test case diagnostics. Perfect agreement among simulations and the QAS is not necessarily expected: this is because many programs contain simplifying assumptions to ease calculation burden (e.g., constant air density), and the QAS contains idealized simplifying assumptions (in order to be solved analytically) that cannot always be exactly reproduced by some simulation programs that are conceived and hardcoded with more realistic assumptions.

## **Results and Conclusions**

The new airside HVAC equipment model test cases have been vetted via field trials with a number of detailed state-of-the-art whole-building energy simulation programs from around the world (see Table ES-1). The field-trial process was iterative in that executing the simulations led to refinement of the test cases, and the results of the tests led to improving and debugging the models. Improvements to simulation programs or simulation inputs made by participants must have a mathematical and a physical basis and must be applied consistently across tests. Arbitrary modification of a simulation program's input or internal code just for the purpose of more closely matching a given set of results is not allowed. All improvements were requested to be documented and justified in the modeler reports. Improvements to

simulation models are summarized in Table 3-18 (see Part III, Section 3.6.1.2), with additional details in Part III, Section 3.4 and in the participant modeler reports of Part III, Section 3.9.

Improvements to the simulation models are evident when the initial results set is compared to the final results set. Initial simulation results for total coil loads obtained are shown for the FC and SZ systems in Figure ES-1 and for the CV and VAV systems in Figure ES-3 (abbreviations along these figures' x-axes are shorthand for the case descriptions given in Part I). Figures ES-1 and ES-3 show each participant's results after the first "blind" round of simulations, and before the QAS results were distributed to the working group, for most of the programs. Two late-entering participating programs (IES-VE and LCEM) were allowed an initial non-blind round to accelerate their simulation trials and asked to document intermediate revisions to their input files or programs in their modeler reports; these participants also benefitted from first seeing the test specification after revisions engendered by the initial simulation trials. The late entering participants were also a benefit to the test specification vetting process, as this introduced new test specification reviewers after its initial revisions.

The results of Figure ES-1 indicate that for the FC and SZ systems there was initially 1% to 19% *average* disagreement for a given program versus the QAS results. The results of Figure ES-3 indicate that for the CV and VAV systems there was initially 2% to 37% average disagreement for a given program versus the QAS results. The additional complexity of specifying the CV and VAV systems (multi-zone with reheat) and inputting them in the models generated more initial disagreements than for the simpler FC and SZ systems. This emphasizes the importance of testing the different systems.

The final set of total coil load results for all the simulations and the QAS are shown in Figure ES-2 for the FC and SZ systems and in Figure ES-4 for the CV and VAV systems (abbreviations along these figures' x-axes are the same as in Figure ES-1, except the temperature values are ODB/ODP [outdoor dry bulb/outdoor dew point] instead of ODB/OWB [outdoor wet bulb]). After correcting software errors and other model improvements using the diagnostic output, the mean of all simulated results of total cooling coil load for the tested programs are, on average, within 1.3% of the QAS results, with average variations among the test cases for a given program of up to 3%.

## **Major Accomplishments**

The major accomplishments of this project were:

- Extension of the BESTEST procedures to include in-depth diagnostic analytical verification test cases for the air distribution side of typical HVAC equipment, based on ASHRAE research project RP 865.<sup>5</sup>
- Development of clear unambiguous test specifications such that a variety of state-of-the-art building simulation programs with a variety of input structures and modeling approaches can perform the tests.
- Development of QAS results for all test cases by comparing, reconciling, and merging the two original external spreadsheet solutions from RP 865.
- Improved accuracy of, or potential improvements identified for, all but one of the models that participated in the simulation trials of the test cases.
- Development of a set of national and international simulation results representative of the range of legitimate modeling differences for the current state of the art in whole building energy simulation computer programs.
- Development of a new test suite suitable to initiate the ANSI/ASHRAE process for inclusion in Standard 140.

**Table ES-1. Airside HVAC Cases, Participating Organizations and Models**

Model	Authoring Organization	Implemented by	Abbreviation
Quasi-Analytical Solution (QAS)	PSU <sup>a</sup> /UNO <sup>b</sup> /TAMU <sup>c</sup> /NREL <sup>d</sup> / JNA <sup>e</sup> /MDK <sup>f</sup> , United States	NREL <sup>d</sup> /JNA <sup>e</sup> /MDK <sup>f</sup> , United States	QAS/PSU-TAMU-NREL
DEEAP <sup>g</sup> 1.1.2	AAON, Inc., United States	AAON, Inc., United States	DEEAP/AAON
DeST <sup>h</sup> 2	Tsinghua University, China	Tsinghua University, China / LBNL <sup>i</sup> , United States	DeST/TsinghuaU-LBNL
DOE-2.2 V48L	JJH <sup>j</sup> /LBNL <sup>i</sup> /UC <sup>k</sup> , United States	NREL <sup>d</sup> /JNA <sup>e</sup> /MDK <sup>f</sup> , United States	DOE-2.2/NREL
EnergyPlus 8.2.0	DOE-BT <sup>l</sup> , United States	GARD Analytics, Inc., United States	EnergyPlus/GARD
IES-VE <sup>m</sup> 2014.2	IES <sup>n</sup> , United Kingdom	IES <sup>n</sup> , United Kingdom	IES-VE/IES
LCEM <sup>o</sup> 3.10	MLIT <sup>p</sup> , Japan	TTE <sup>q</sup> , Japan	LCEM/MLIT-TTE
TRNSYS 17.01.0028	TESS <sup>r</sup> /UWM <sup>s</sup> , United States	TESS <sup>r</sup> , United States	TRNSYS/TESS

<sup>a</sup> PSU: The Pennsylvania State University, United States

<sup>b</sup> UNO: University of Nebraska - Omaha, United States

<sup>c</sup> TAMU: Texas A&M University, United States

<sup>d</sup> NREL: National Renewable Energy Laboratory, United States

<sup>e</sup> J. Neymark & Associates, United States

<sup>f</sup> Mike D. Kennedy, Inc., United States

<sup>g</sup> DEEAP: Detailed Energy and Economic Analysis Program

<sup>h</sup> DeST: Designer's Simulation Toolkit

<sup>i</sup> LBNL: Lawrence Berkeley National Laboratory, United States

<sup>j</sup> JJH: James J. Hirsch & Associates, United States

<sup>k</sup> UC: University of California, United States

<sup>l</sup> DOE-BT: U.S. Department of Energy, Office of Building Technologies, Energy Efficiency and Renewable Energy, United States

<sup>m</sup> IES-VE: Integrated Environmental Solutions - Virtual Environment

<sup>n</sup> IES: Integrated Environmental Solutions, United Kingdom

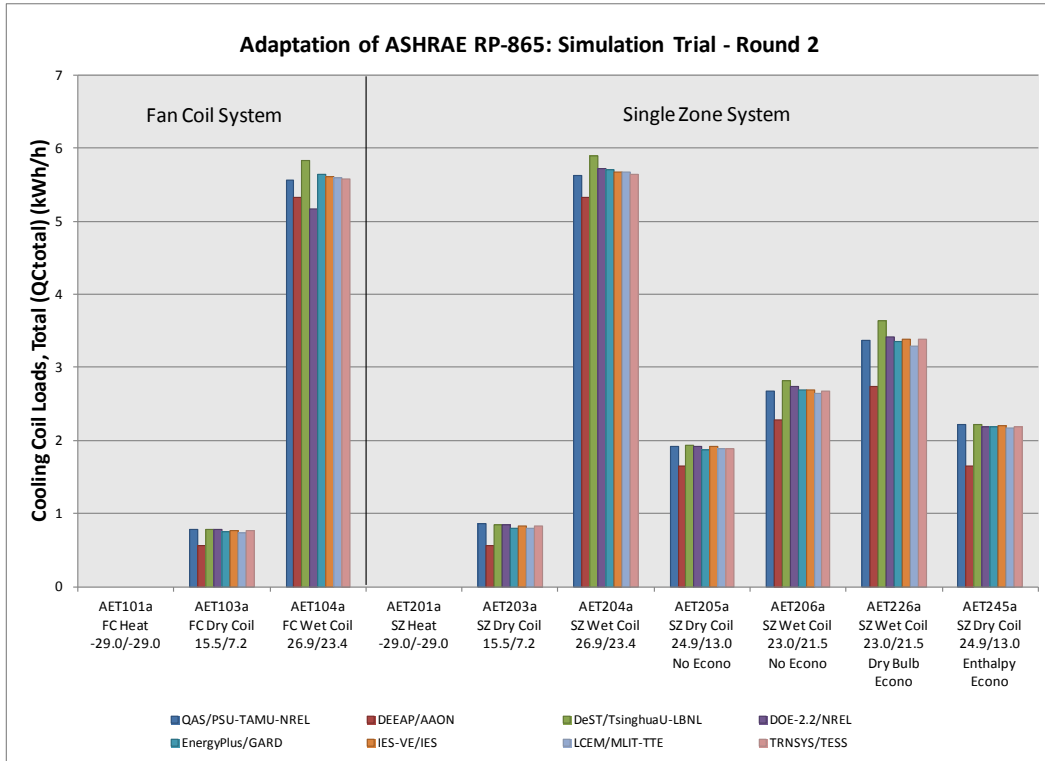
<sup>o</sup> LCEM: Life Cycle Energy Management tool

<sup>p</sup> MLIT: Ministry of Land, Infrastructure, Transportation and Tourism, Japan

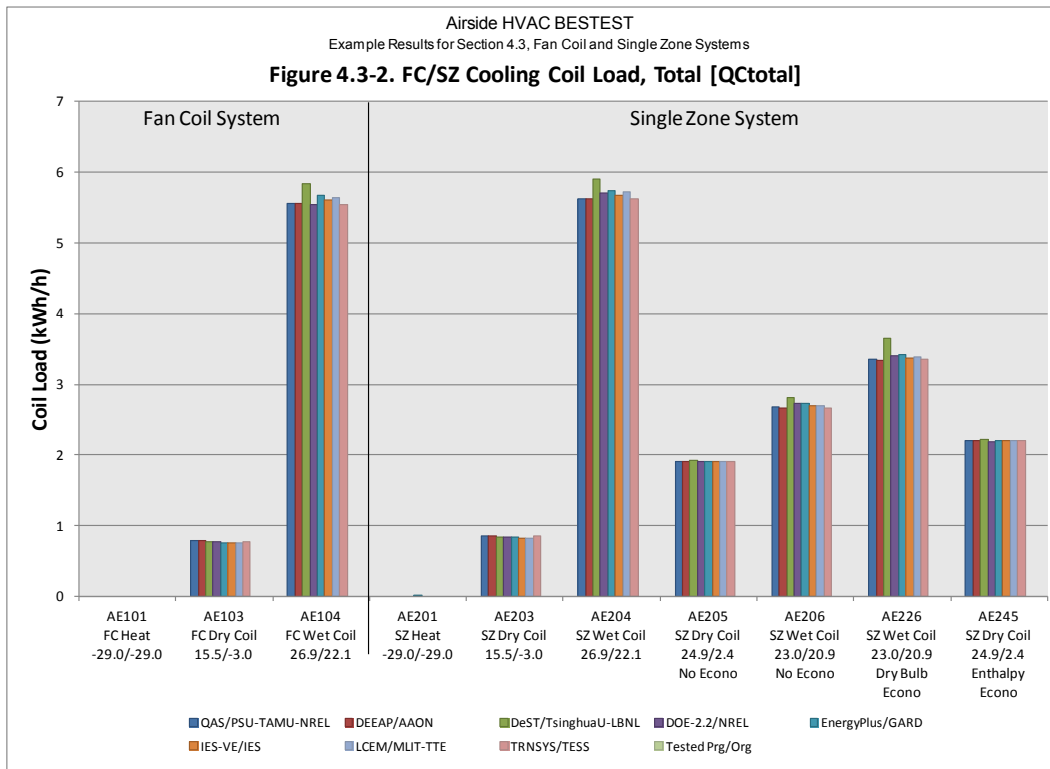
<sup>q</sup> TTE: Takasago Thermal Engineering, Japan

<sup>r</sup> TESS: Thermal Energy System Specialists, United States

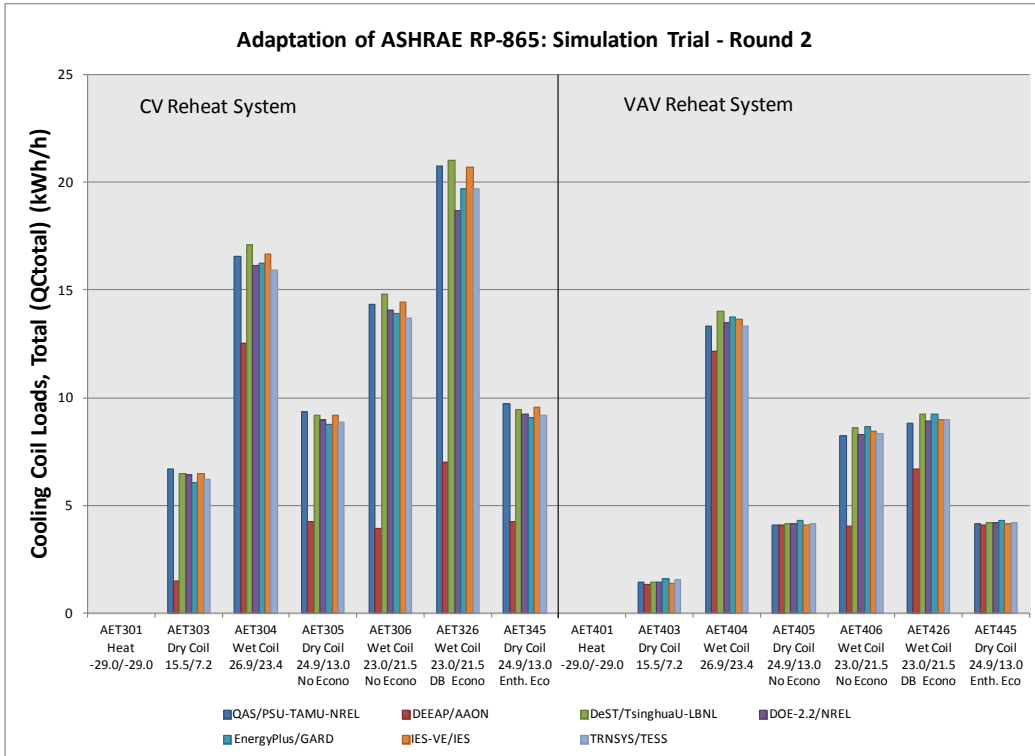
<sup>s</sup> UWM: University of Wisconsin – Madison, United States



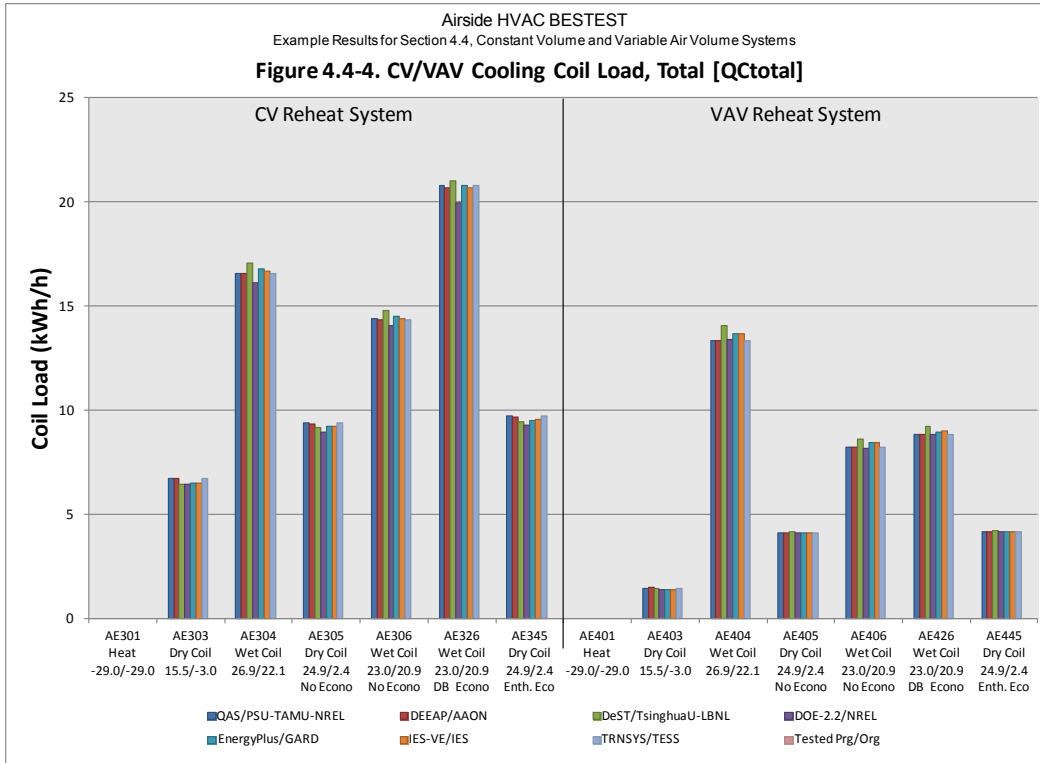
**Figure ES-1. Airside HVAC BESTEST—total coil load, FC and SZ, before BESTESTing** (Abbreviations along x-axis describe test cases, numeric values are ODB/OWB; see Part I for details.)



**Figure ES-2. Airside HVAC BESTEST—total coil load, FC and SZ, after BESTESTing** (Abbreviations along x-axis describe test cases, numeric values are ODB/ODP; see Part I for details.)



**Figure ES-3. Airside HVAC BESTEST—total coil load, CV and VAV, before BESTESTing** (Abbreviations along x-axis describe test cases, numeric values are ODB/OWB; see Part I for details.)



**Figure ES-4. Airside HVAC BESTEST—total coil load, CV and VAV, after BESTESTing** (Abbreviations along x-axis describe test cases, numeric values are ODB/ODP; see Part I for details.)

## Findings

A number of important technology advances occurred as a result of running the test cases:

- The QAS allowed diagnosis of errors and identification of potential areas for improvement that may have been missed in a comparative study of whole-building energy simulation programs versus each other, without a quasi-analytical solution benchmark.
- Of 26 found program errors or potential improvements, 17 were diagnosed and fixed, 2 are planned for investigation by the software authors, and 7 are unresolved at the time of this writing. Several of the found errors affected some individual results by >20%.
- Additionally, 17 input errors were found and corrected. These provided the basis for revealing a number of test specification ambiguities that were then clarified during the simulation trials and underscore the importance of simulation trials in vetting test specifications before they are included in a standard method of test.
- Based on this work, there are a number of recommended areas for further investigation with respect to developing additional validation test cases for airside HVAC equipment modeling. These are described in detail in Part III, Section 3.5.2.

The QAS provides an effective reference or benchmark against which software can be tested, while the example simulation results of Part IV indicate a reasonable range of disagreement for the current state-of-the-art simulation programs relative to the QAS. Based on results *after* several simulation trial iterations (“BESTESTing”) and resulting model improvements, all of the tested programs appear reliable for modeling these HVAC air distribution systems under the conditions tested, although use of constant system air density by some simulation programs causes modeling disagreements at extreme-low outdoor air temperatures for the specified CV system. Without additional test cases having realistic hourly varying weather data, we do not have a means to scale, relative to typical annual simulations, the effect of disagreements identified in these in-depth diagnostic steady-state analytical verification tests. For future work, we recommend developing an airside HVAC comparative test suite with annual hourly varying weather data, analogous to the HVAC BESTEST Volume 2<sup>3</sup> working-fluid side test suite.

## Closing Remarks

The work presented in this report and the work that has preceded it are important for two reasons:

- The methods have been successful at correcting software errors in advanced building energy simulation programs throughout the world.
- The methods are finding their way into industry by being adopted as the theoretical basis for formalized standard methods of test and software certification procedures; in this sense, the work may be thought of as pre-normative research.

ANSI/ASHRAE Standard 140<sup>1</sup> and the BESTEST reports that comprise the test suites contained therein are being referenced and used by a growing number of code promulgation authorities throughout the world. ASHRAE Standard 90.1,<sup>23</sup> which is ASHRAE’s consensus energy code for commercial buildings and for non-low-rise residential buildings, requires that software used for demonstrating performance compliance with Standard 90.1 be tested using ASHRAE Standard 140-2011.<sup>19</sup> Software used for calculating energy savings for purposes of the energy-efficient commercial building tax deductions in the United States<sup>24</sup> must be tested with Standard 140-2007.<sup>18</sup> As part of building energy performance assessments under the European Community’s Energy Performance Directive,<sup>25</sup> several countries are using software tools that have been checked with BESTEST. Further details of international use of BESTEST, along with growing evidence that the BESTEST procedures are becoming part of software developers’ normal in-house quality control efforts, are included elsewhere.<sup>8,26,27</sup>

Computer scientists universally accept the merits of software testing. A well-known rule of thumb is that in a typical programming project more than 50% of the total cost is expended in testing the program or system being developed.<sup>28</sup> Of this, about 20% of development time goes toward system testing.<sup>29</sup> Because new energy-related technologies are continually being introduced into the buildings market, there will always be a need for further development of simulation models, combined with a substantial program of testing and validation. Such an effort should contain all the elements of an overall validation methodology,<sup>8,13,14</sup> including:

- Analytical verification
- Comparative testing and diagnostics
- Empirical validation.

Future work should therefore:

- Continue to produce analytical verification tests
- Continue to develop a set of diagnostic comparative tests that emphasize the modeling issues important in large commercial buildings, including more tests for HVAC systems, and other mechanical equipment, including on-site power generation equipment
- Develop a sequentially ordered series of high-quality diagnostic data sets for empirical validation.

The work described herein represents continuing progress in the effort to develop carefully validated building energy simulation tools. Continued development and validation of whole-building energy simulation programs are two of the most important activities meriting the support of national energy research programs and are a beneficial area for international collaboration. The U.S. Department of Energy should consider future International Energy Agency collaborations for this essential research area.

Finally, the authors wish to acknowledge that the expertise available through ASHRAE SSPC 140 and other international participants, and their dedication to the simulation trials, were essential to the success of this project. Over the four-year field-trial effort, there were several revisions to the test specifications and subsequent re-executions of the computer simulations. This iterative process led to the refining of the new BESTEST cases, and the results of the tests led to improving and debugging the simulation models. This process underscores the importance of the software development industry participation in this project via the ASHRAE Standard 140 project committee (SSPC 140) and via our previous collaborations with IEA software testing and validation projects. Extensive field trials and the resulting enhancements to the tests were much more comprehensive and cost-effective with the voluntary participation of the national and international software-development industry experts. Their persistence throughout the project demonstrates their interest in, and the importance of, this work.

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## Introduction

A procedure for testing the ability of whole building energy simulation software to model heating, ventilating, and air-conditioning (HVAC) air distribution equipment was developed by the National Renewable Energy Laboratory (NREL), in collaboration with American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) Standing Standard Project Committee 140 (SSPC 140). SSPC 140 is the committee responsible for ASHRAE Standard 140, *Standard Method of Test for the Evaluation of Building Energy Analysis Computer Programs* (ANSI/ASHRAE 2014).

This work builds off ASHRAE Research Project 865 (RP 865) (Yuill and Haberl 2002). RP 865 developed a test specification, and two independently developed spreadsheet solutions intended as quasi-analytical solutions, for a number of typical HVAC configurations such as constant volume and variable air volume reheat systems. At the time RP 865 was developed, the scope for input descriptions in its test specification was limited to two prominent whole-building energy simulation programs. NREL led the collaborative effort to a) reconcile differences in the two analytical solutions into a single final quasi-analytical solution (QAS), b) rework the test specifications such that they would be unambiguous for the input structures of most whole-building energy simulation programs with time steps of one hour or less, and c) field test the specifications with a variety of different simulation programs and associated software development groups around the world, to ensure their suitability as a standard method of test that can be integrated into ASHRAE Standard 140. The collaboration included a number of software developer members of SSPC 140 along with other international software developers and participants.

## Background

Further discussion of adaptation of RP 865, and the importance of the HVAC air distribution system modeling problem, is included in the Executive Summary of this report (see “Background” there).

## Advantages of BESTEST Methodology

An advantage of the Building Energy Simulation Test and Diagnostic Method (BESTEST) methodology is that a program is examined over a broad range of parametric interactions based on a variety of output types, minimizing the possibility of concealing problems by compensating errors. Performance of the tests resulted in quality improvements, or identification of potential improvements, to all but one of the building energy simulation models used in the field trials. Some of the bugs that were found may well have been present for many years. The fact that they have just now been uncovered shows the power of BESTEST and suggests the importance of continuing to develop formalized validation and diagnostic methods. Only after coding bugs and input ambiguities have been eliminated can the assumptions and approximations in the algorithms be evaluated.

Checking a building energy simulation program for the first time with the airside HVAC equipment model test cases requires a few days for an experienced user, not including improvements to the software, if necessary. Subsequent program checks are faster because input files may be reused. Because the simulation programs have taken many years to produce, the new BESTEST cases provide a cost-effective way of testing them. As we continue to develop new test cases, we will adhere to the principle of parsimony so the entire suite of BESTEST cases may be implemented by users with a reasonable time commitment. Several software developers have automated the tests, further reducing the time to rerun them as an internal quality control check after code modifications.

Software developers, architects, engineers, and researchers can use these new BESTEST cases in a number of ways, including:

- Comparing predictions from building energy simulation programs to the QAS and simulation results in this report, where the QAS results constitute a reliable set of theoretical results given the underlying physical assumptions in the case definitions, and the simulation results represent a reasonable range of disagreement for the current state-of-the-art in simulation modeling.
- Comparing several building energy simulation programs to determine the degree of disagreement among them.
- Diagnosing the algorithmic sources of prediction differences.
- Checking a program against a previous version of itself after the internal code has been modified to ensure that only the intended changes actually resulted.
- Checking a program against itself after a single algorithmic change to understand the sensitivity between algorithms.

## Other BESTEST Procedures

As a BESTEST user, if you have not already tested your software with previously developed BESTEST procedures, we strongly recommend that you run all the building thermal fabric and mechanical equipment cases currently included in ANSI/ASHRAE Standard 140-2014 (ANSI/ASHRAE 2014):

- Class I tests (detailed diagnostic tests for simulation software capable of hourly or subhourly simulation time steps)
  - International Energy Agency (IEA) BESTEST, building thermal fabric comparative tests (Judkoff and Neymark 1995a)
  - IEA BESTEST, in-depth diagnostic ground-coupled slab-on-grade heat transfer analytical verification tests (Neymark and Judkoff, et al. 2008a)
  - HVAC BESTEST Volume 1, unitary cooling equipment analytical verification tests (Neymark and Judkoff 2002)
  - HVAC BESTEST Volume 2, unitary cooling equipment comparative tests (Neymark and Judkoff 2004)
  - HVAC BESTEST Fuel-Fired Furnace analytical verification and comparative tests (Purdy and Beausoleil-Morrison 2003)
- Class II tests (for all types of building load calculation methods regardless of time-step granularity)
  - Home Energy Rating Systems (HERS) BESTEST, building thermal fabric comparative tests (Judkoff and Neymark 1995b). These test cases have a more realistic base building than the Class I building thermal fabric test cases; however, their ability to diagnose sources of differences among results is not as robust (Neymark and Judkoff 1997).

Other BESTEST procedures published by NREL that we recommend include:

- IEA BESTEST multi-zone non-airflow in-depth diagnostic cases (Neymark and Judkoff, et al. 2008b). These test cases are currently under consideration for inclusion in Standard 140.
- BESTEST-EX thermal fabric test cases for existing homes (Judkoff et al. 2010, 2011). These include building physics tests, some of which update HERS BESTEST, under consideration as Standard 140 Class II tests, and calibration tests under consideration for inclusion as a Residential Energy Services Network (RESNET) Standard Method of Test (Judkoff et al. 2015).

## Final Report Structure

This report is divided into four parts. Part I is a user's manual that furnishes instructions on how to apply this BESTEST procedure. Part II describes the development, field-testing, and production of the QAS.

Part III describes the development, field testing, and production of simulation results data for the procedure. Part IV presents the QAS and simulation program example results in tables and graphs; these data can be used to compare results from other programs to the Part IV results.

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# 1.0 Part I: User's Manual: Procedure and Test Specification

## 1.1 General Description of the Test Cases

Twenty-four test cases are summarized in Tables B1 through B8 (see Appendix B). The test cases are designed to test the ability of building energy analysis tools to model heating, ventilating, and air conditioning (HVAC) air-distribution system equipment. These are steady-state analytical verification tests, where simulation results are compared to a quasi-analytical solution (QAS) (secondary mathematical truth standard), and may also be compared to other example simulation results. All results are provided in Part IV.

Further test case summary and background information is provided in the Introduction.

## 1.2 Accompanying Electronic Files

The following files are required for running the test cases described in Part I. Other accompanying files that support other parts of this report are described in the Electronic Media Contents with the front matter Table of Contents.

### 1.2.1 Weather Data Files

The following Typical Meteorological Year 2 (TMY2) format weather data files are included with the accompanying electronic media. These are applied as called out in the test specification:

- **AE101.TM2:** TMY2 weather data for -29.0°C/-29.0°C constant outdoor dry-bulb (ODB)/outdoor dew-point (ODP) temperatures
- **AE103.TM2:** TMY2 weather data for 15.5°C/-3.0°C constant ODB/ODP
- **AE104.TM2:** TMY2 weather data for 26.9°C/22.1°C constant ODB/ODP
- **AE105.TM2:** TMY2 weather data for 24.9°C/2.4°C constant ODB/ODP
- **AE106.TM2:** TMY2 weather data for 23.0°C/20.9°C constant ODB/ODP.

A summary of site weather data parameters is provided in Appendix A, Table A-2.

### 1.2.2 Output Report Template

For reporting output, use the following file provided with the accompanying electronic media:

- **Sec5-5out.xls.**

## 1.3 Modeling Rules

### 1.3.1 Time Convention

All references to time in this specification are to local standard time and assume that hour 1 = the interval from midnight to 1 A.M. Daylight savings time or holidays shall not be used for scheduling. TMY2 data are in hourly bins corresponding to local standard time.

### **1.3.2 Geometry Convention**

If the program being tested includes the thickness of walls in a three-dimensional definition of the building geometry, then wall, roof, and floor thicknesses shall be defined such that the interior air volume of the building model remains as specified (e.g., for the alternate specification zone geometry of Section 1.5.5.1.1.5.1,  $6 \times 8 \times 2.7 \text{ m} = 129.6 \text{ m}^3$ ). These thicknesses shall extend exterior to the defined internal volume.

### **1.3.3 Nonapplicable Inputs**

In some instances the specification includes input values that do not apply to the input structure of the program being tested. When this occurs, disregard the nonapplicable inputs and continue. Selected equivalent inputs are included in the test specification for those programs that may need them.

### **1.3.4 Consistent Modeling Methods**

Where options exist within a simulation program for modeling a specific thermal behavior, consistent modeling methods shall be used for all cases. For example, if a program gives a choice of methods for modeling system airflows, the same airflow modeling method is to be applied for all cases. For the purpose of generating the example results of Part IV, the ASHRAE SSPC 140 simulation-trial working group participants used the most detailed level of modeling that was allowed by their simulation programs and that was consistent with the level of detail provided in this test specification.

### **1.3.5 Equivalent Modeling Methods**

Where a program or specific model within a program does not allow direct input of specified values or where input of specified values causes instabilities in a program's calculations, modelers shall develop equivalent inputs that match the intent of the test specification as nearly as the software being tested allows. Such equivalent inputs shall be developed based on the data provided in the test specification, and such equivalent inputs shall have a mathematical, physical, or logical basis and shall be applied consistently throughout the test cases. *For the purpose of documenting equivalent modeling methods, many simulation-trial working group participants used the standard output report format of Standard 140 (ANSI/ASHRAE 140-2014, Annex A2).*

### **1.3.6 Simulation Initialization and Preconditioning**

If the program being tested allows for pre-conditioning (iterative simulation of an initial time period until temperatures or coil loads or both stabilize at initial values), that capability shall be used. If the program being tested allows and if applicable to the model, the simulation initialization process shall begin with zone air conditions that equal the outdoor air conditions.

### **1.3.7 Simulation Duration**

Use the weather data provided to run the simulation. Give outputs as required per Section 1.6.5.1.

### **1.3.8 Simulation Input Files**

All supporting data required for generating results with the tested software shall be saved, including:

- Input files



- Processed weather data
- Intermediate files containing calculations used for developing inputs
- A “Readme-software-name-yymmdd.pdf” file that briefly describes the contents of the above files according to their file type (i.e., their “.xyz” file extension).

### **1.3.9 Omitted Test Cases**

If a program being tested omits a test case, provide an explanation in the modeler report.

### **1.3.10 Modeler Reports**

The ASHRAE SSPC 140 simulation-trial working group participants submitted modeler reports along with their simulation results (see Part III, Section 3.9). Users developing modeler reports may consider using the structure of any of those modeler reports as a template.

## **1.4 Comparing Tested Program Output to the Quasi-Analytical Solution and Example Simulation Results**

Compare output with the QAS results; output may also be compared with other example simulation results or with other results that were generated using this test procedure. QAS and example simulation results are provided in Part IV. Information about how the QAS and example simulation results were produced is included in Parts II and III, respectively. For the convenience of users who wish to plot or tabulate their results along with the example results of Part IV, electronic versions of the example results have been included with the accompanying files Results-FCSZ.xlsm (for FC and SZ systems) and Results-CVVV.xlsm (for CV and VAV systems); see subfolder “PartIV-Files”.

### **1.4.1 Criteria for Determining Agreement between Results**

There are no formal criteria for when results agree or disagree. Determination of when results agree or disagree is left to the user. In making this determination, the user should consider:

- Magnitude of results for individual cases.
- Magnitude of difference in results between certain cases (e.g., Case AE326 – Case AE306).
- Same direction of sensitivity (positive or negative) for difference in results between certain cases (e.g., Case AE326 – Case AE306).
- If results are logically counterintuitive with respect to known or expected physical behavior.
- Availability of a mathematical truth standard (analytical solution) or secondary mathematical truth standard (quasi-analytical solution or verified numerical model results). *These terms are defined in ASHRAE Standard 140 (ANSI/ASHRAE 2014, Section 3.1).*
- Where a mathematical or secondary-mathematical truth standard is provided, the degree of disagreement that occurred for other simulation results versus the mathematical truth standard.
- Example simulation results do not represent a truth standard.

For any given case, a tested program may fall outside the range of example results without necessarily being incorrect. However, it is worthwhile to investigate the source of significant differences as the collective experience of the authors is that such differences often indicate problems with the software or its use, including, but not limited to:

- User input error, where the user misinterpreted or incorrectly entered one or more program inputs
- A problem with a particular algorithm in the program or its documentation

- One or more program algorithms used outside their intended range.

Also, for any given case, a program that yields values in the middle of the range established by the example results should not be perceived as better or worse than a program that yields values at the borders of the range.

### **1.4.2 Rules for Modifying Simulation Programs or Simulation Inputs**

Modifications to simulation programs or simulation inputs shall have a mathematical, physical, or logical basis and shall be applied consistently across tests. Such improvements must be documented in modeler reports. Arbitrary modification of a simulation program's input or internal code just for the purpose of more closely matching a given set of results shall be prohibited.

## **1.5 Test Specifications – Airside HVAC Equipment Test Cases**

**1.5.1 Intentionally left blank to facilitate renumbering for future Standard 140 adaptation.**

**1.5.2 Intentionally left blank to facilitate renumbering for future Standard 140 adaptation.**

### **1.5.3. Definitions, Abbreviations, and Acronyms for Part I**

#### **1.5.3.1 Definitions**

*Note to users: The definitions listed here are supplementary to the definitions of ASHRAE Standard 140-2014 (ANSI/ASHRAE 2014). See Standard 140-2014, Section 3.1 for additional terminology.*

**Apparatus dew point:** the effective coil surface temperature when there is dehumidification. On the psychrometric chart, this is the intersection of the condition line and the saturation curve, where the condition line is the line going through entering air conditions with slope defined by the sensible heat ratio (ratio of sensible heat transfer to total [sensible + latent] heat transfer for a process).

**Bypass factor:** the percentage of the distribution air that does not come into contact with the cooling coil; the remaining air is assumed to exit the coil at the average coil temperature (apparatus dew point). (see also *apparatus dew point*.)

**Cooling coil sensible load:** the sum of the rate of heat extraction required to:

- Cool the leaving moist air mass from the cooling coil entering air temperature to the cooling coil leaving air temperature
- Cool any to-be-condensed vapor from the cooling coil entering air temperature to the condensation temperature
- Cool any condensate from the condensation temperature to the leaving condensate temperature.

**Informative Note:** For example equations, see *2012 ASHRAE Handbook of HVAC Systems and Equipment* (ASHRAE 2012), Chapter 23, Equations 39 and 39a.

**Cooling coil latent load:** the rate of heat extraction required to condense the moisture in cooling coil entering air that becomes condensate. **Informative Note:** For example equation, see 2012 ASHRAE Handbook of HVAC Systems and Equipment (ASHRAE 2012), Chapter 23, Equation 38.

**Cooling coil total load:** the sum of *cooling coil sensible load* and *cooling coil latent load*. **Informative Note:** For example equations, see 2012 ASHRAE Handbook of HVAC Systems and Equipment (ASHRAE 2012), Chapter 23, Equations 36 and 37.

**Design nominal zone supply airflow rate:** the maximum zone supply volumetric airflow rate using air conditions at the supply fan inlet. **Informative Note:** The actual design zone supply air volumetric flow varies as supply fan heat and terminal reheat increase the specific volume of the air away from the supply fan inlet.

**Design system return airflow rate:** the volumetric return airflow rate calculated from the *design system supply airflow rate* minus the total *zone exhaust airflow rate(s)* for the defined zone(s). This is the volumetric airflow rate at which the return fan pressure rise is specified. **Informative Note:** The actual return fan volumetric flow varies with the return air mass flow and the specific volume of the air entering the return fan.

**Design system supply airflow rate:** the maximum system supply volumetric airflow rate at the supply fan inlet. This is the volumetric airflow rate at which the supply fan pressure rise is specified for the VAV system. **Informative Note:** In a variable air volume system, the supply fan volumetric flow varies to meet the zone terminal airflow requirements.

**Economizer:** a control system that conserves cooling energy by increasing the flow of outdoor air above minimum ventilation requirements when control logic indicates using more outdoor will reduce or eliminate cooling loads.

**Fan mechanical efficiency, or fan total efficiency ( $\eta_t$ ):**

$\eta_t = H_o / H_i$ , where:

$H_o = Q \times P_t \times K_p \times C$ ; this is the fan power output (causing airflow and pressure rise), W (hp), where:

$Q$  = fan airflow rate, m<sup>3</sup>/s (cfm)

$P_t$  = fan total pressure rise, Pa (in. wg); fan total pressure rise includes static pressure (from compression) and velocity pressure (from rate of motion)

$K_p$  = compressibility coefficient (dimensionless). **Informative Note:** For fan total pressure < 12 in. wg,  $K_p$  is usually greater than 0.99 and may be taken as unity (AMCA/ASHRAE 1985).

$C$  = Units conversion constant. **Informative Note:** For Système Internationale (SI) units,  $C = 1$ ; for inch-pound (I-P) units,  $C$  may be taken as approximately 1/6343.3 per the literature (AMCA/ASHRAE 2007).

$H_i$  = fan power input, W (hp). This is also called “power input to impeller” or “shaft power input”, and is the remaining mechanical power after subtracting fan motor and transmission drive (e.g., belt drive) power losses from the fan motor input power (see *motor efficiency* and *transmission drive efficiency*). **Informative Note:** This parameter is designated as  $W_{sh}$  in Standard 140-2014 (ANSI/ASHRAE 2014), informative Annex B15.

**Informative Note:** Fan total efficiency and related terminology are further described in the literature. (AMCA/ASHRAE 2007; McQuiston and Parker 1994)

**Heating coil load:** the rate of heat addition required to heat the moist air mass entering the heating coil from the heating coil entering air temperature to the heating coil leaving air temperature.

**Motor efficiency ( $\eta_m$ ):**

$\eta_m = H_m / H_e$ , where:

$H_m$  = usable motor shaft output power, W (hp)

$H_e$  = motor electric input power, W (hp). **Informative Note:** This parameter is designated as  $W$  in Standard 140-2014 (ANSI/ASHRAE 2014), informative Annex B15.

**Nominal zone supply airflow rate:** the zone supply volumetric airflow rate calculated using conditions of the air at the supply fan inlet. **Informative Note:** The actual zone supply air volumetric flow rate varies as supply fan heat and terminal reheat increase the specific volume of the air away from the supply fan inlet.

**Pre-heat coil load:** the rate of heat addition required to heat the moist air mass entering the pre-heat coil from the pre-heat coil entering air temperature to the pre-heat coil leaving air temperature.

**Reheat coil load:** for a given zone reheat coil, the rate of heat addition required to heat the given zone supply air moist air mass from the system supply air temperature to the zone supply air temperature for the given zone.

**System supply airflow rate:** the volumetric airflow rate measured at the supply fan inlet. **Informative Note:** As the temperature and humidity ratio of the air entering the supply fan change, the specific volume of that air changes. This means that the mass flow rate of supply air will change slightly from case to case.

**Transmission drive efficiency ( $\eta_d$ ):**

$\eta_d = H_i / H_m$ , where:

$H_i$  = fan power input, W (hp); see  $H_i$  under *fan mechanical efficiency*.

$H_m$  = usable motor shaft output power, W (hp).

**Informative Note:** Fan motor shaft power is typically transferred to the fan impeller using belts or direct drive.

**Zone exhaust airflow rate:** the volumetric airflow rate measured at the inlet of a given zone's exhaust fan. **Informative Note:** As the temperature and humidity ratio of the air entering an exhaust fan change, the specific volume of that air changes. This means that the mass flow rate of exhaust air from each zone will change from case to case.

**Zone latent load:** for the test cases of Section 1.5.5, the rate of heat addition to vaporize the water added to the zone at a temperature of  $0^\circ$  in the units system used (SI or I-P), plus the energy required to heat that added vapor from  $0^\circ$  to zone temperature. **Informative Note:** This definition may be expressed as shown in equation form below (see Part II, Section 2.2.1.8).

Zone latent load = [mass of moisture added to zone]  $\times$  [ $i_{0,\text{water vapor}} + C_{p\text{water vapor}} \times T_{\text{zone}}$ ], where:

mass of moisture added to zone (kg/s or lb/h)

$i_{0,\text{water vapor}}$  = specific enthalpy of water vapor at zero degrees (kJ/kg or Btu/lb)

$C_{p\text{water vapor}}$  = specific heat of water vapor (kJ/(kg·K) or Btu/(lb·°F))

$T_{\text{zone}}$  = zone air temperature (°C or °F).

In the above equation, “[ $i_{0,\text{water vapor}} + C_{p\text{water vapor}} \times T_{\text{zone}}$ ]” equals the enthalpy of water vapor ( $i_g$ ) at the given zone temperature based upon ideal gas laws. This approximates the real gas model values listed in the *2009 ASHRAE Handbook of Fundamentals* (ASHRAE 2009), Chapter 1, Table 3.

**Zone sensible cooling load:** for the test cases of Section 1.5.5, the rate at which sensible heat must be extracted from the zone to maintain the zone air temperature set point. *Informative Note:* This definition may be expressed as shown in equation form below (Yuill and Haberl 2002).

Zone sensible cooling load = [dry air mass flow rate]  $\times$  [ $(T_{\text{zone}} - T_{\text{supply}}) \times (C_{p\text{air}} + C_{p\text{water vapor}} \times W_{\text{supply}})$ ], where:

dry air mass flow rate (kg/s or lb/h)

$T_{\text{zone}}$  = zone air temperature (°C or °F)

$T_{\text{supply}}$  = zone supply air temperature (°C or °F); for the FC and SZ systems this is the same as the system supply air temperature

$C_{p\text{air}}$  = specific heat of dry air (kJ/(kg·K) or Btu/(lb·°F))

$C_{p\text{water vapor}}$  = specific heat of water vapor (kJ/(kg·K) or Btu/(lb·°F))

$W_{\text{supply}}$  = zone supply air humidity ratio ((kg or lb water vapor)/(kg or lb dry air)); for the FC and SZ systems, this is the same as the system supply air humidity ratio.

**Zone sensible heating load:** for the test cases of Section 1.5.5, the rate at which sensible heat must be added to the zone to maintain the zone air temperature set point. *Informative Note:* This definition may be expressed as shown in equation form below (based on Yuill and Haberl 2002).

Zone sensible heating load = [dry air mass flow rate]  $\times$  [ $(T_{\text{supply}} - T_{\text{zone}}) \times (C_{p\text{air}} + C_{p\text{water vapor}} \times W_{\text{supply}})$ ], where:

dry air mass flow rate (kg/s or lb/h)

$T_{\text{supply}}$  = zone supply air temperature (°C or °F); for the FC and SZ systems, this is the same as the system supply air temperature

$T_{\text{zone}}$  = zone air temperature (°C or °F)

$C_{p\text{air}}$  = specific heat of dry air (kJ/(kg·K) or Btu/(lb·°F))

$C_{p\text{water vapor}}$  = specific heat of water vapor (kJ/(kg·K) or Btu/(lb·°F))

$W_{\text{supply}}$  = zone supply air humidity ratio ((kg or lb water vapor)/(kg or lb dry air)); for the FC and SZ systems, this is the same as the system supply air humidity ratio.

**Zone supply air mass flow fraction:** the fraction of system supply air mass flow distributed to each zone.

### 1.5.3.2 Abbreviations and Acronyms

AMCA:	Air Movement and Control Association International, Inc.
ANSI:	American National Standards Institute
ASHRAE:	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BESTEST:	Building Energy Simulation Test and Diagnostic Method
BF:	bypass factor
CV or CV <sub>reheat</sub> :	constant volume system, see Section 1.5.5.3
FC:	fan coil system, see Section 1.5.5.1
HVAC:	heating, ventilating, and air-conditioning
I-P:	inch-pound
ODB:	outdoor dry-bulb temperature
ODP:	outdoor dew-point temperature
QAS:	quasi-analytical solution – see Part II
QC <sub>Sensible</sub> :	system cooling coil sensible load (kWh/h)
QC <sub>Latent</sub> :	system cooling coil latent load (kWh/h)
QC <sub>Total</sub> :	system cooling coil total load, QC <sub>Sensible</sub> + QC <sub>Latent</sub> (kWh/h)
QH:	system heating coil load (kWh/h); used for single zone (FC and SZ) system test cases
QH <sub>preheat</sub> :	system preheat coil load (kWh/h)
QH1 <sub>reheat</sub> :	reheat coil load: Zone 1 (kWh/h)
QH2 <sub>reheat</sub> :	reheat coil load: Zone 2 (kWh/h)
QZH <sub>Sensible</sub> :	zone sensible heating load (kWh/h); used for single zone (FC and SZ) system test cases
QZC <sub>Sensible</sub> :	zone sensible cooling load (kWh/h); used for single zone (FC and SZ) system test cases
QZ <sub>Latent</sub> :	zone latent load (kWh/h); used for single zone (FC and SZ) system test cases
QZH1 <sub>Sensible</sub> :	Zone 1 sensible heating load (kWh/h)
QZC1 <sub>Sensible</sub> :	Zone 1 sensible cooling load (kWh/h)
QZ1 <sub>Latent</sub> :	Zone 1 latent load (kWh/h)
QZH2 <sub>Sensible</sub> :	Zone 2 sensible heating load (kWh/h)
QZC2 <sub>Sensible</sub> :	Zone 2 sensible cooling load (kWh/h)
QZ2 <sub>Latent</sub> :	Zone 2 latent zone load (kWh/h)
R:	unit thermal resistance (m <sup>2</sup> ·K/W)
RH <sub>cco</sub> :	cooling coil leaving air relative humidity (%)
SI:	Système Internationale
SSPC 140:	Standing Standard Project Committee responsible for ANSI/ASHRAE Standard 140, <i>Standard Method of Test for the Evaluation of Building Energy Analysis Computer Programs</i>
SZ:	single-zone system, see Section 1.5.5.2
TM2 or TMY2:	Typical Meteorological Year 2
T <sub>zone</sub> :	dry-bulb temperature of the zone air (°C)
T <sub>supply</sub> :	dry-bulb temperature of the zone supply air (°C)
U:	unit thermal conductance (W/(m <sup>2</sup> ·K))
UA:	thermal conductance (W/K)
VAV:	variable air volume system, see Section 1.5.5.4
wg:	water gage
W <sub>zone</sub> :	humidity ratio of the zone air ((kg vapor)/(kg dry air))
W <sub>supply</sub> :	humidity ratio of the zone supply air ((kg vapor)/(kg dry air))

## Subscripts

cco:	system location, cooling coil outlet
hco:	system location, heating coil outlet
ma:	system location, mixed air (recirculated and outdoor) before coils and supply fan
oa:	system location, outdoor air inlet
pco:	system location, preheating coil outlet
ra:	system location, return air
rfi:	system location, return fan inlet
rfo:	system location, return fan outlet
sa:	system location, supply fan outlet (system supply air to zones)
z1:	Zone 1 air
z1s:	system location, supply air to Zone 1 after terminal reheat coil
z2:	Zone 2 air
z2s:	system location, supply air to Zone 2 after terminal reheat coil

### **1.5.4 Intentionally blank to facilitate renumbering for future Standard 140 adaptation.**

### **1.5.5 Airside HVAC Equipment Test Cases**

#### **1.5.5.1 Four Pipe Fan Coil (FC) System Cases (AE100 series)**

The ability to model a four pipe fan coil system shall be tested as described in this section. If the software being tested is capable of applying a variety of system models to address an FC system, the system model that is most similar to the FC system specified below shall be applied.

**Informative Note:** The user may test other possible modeling approaches (available system models) in this context, as appropriate to the software being tested.

##### **1.5.5.1.1 Case AE101: Base Case, High Heating 1**

Case AE101 shall be modeled as described in this section and its subsections. The system configuration shall be modeled as presented in the schematic diagram in Figure 1-1. System input parameters shall be as described in the following sections.

**Informative Note, Objective:** Test model treatment of a four-pipe fan coil air system with high sensible heating load and cold outdoor air.

**Informative Note, Method:** A fan coil air system with a constant volume supply fan and heating and cooling coils, conditions a single zone that has constant sensible and latent internal loads. The model is run at specified constant outdoor and indoor conditions. Resulting coil loads are compared with the QAS and with other example results (see Part IV).

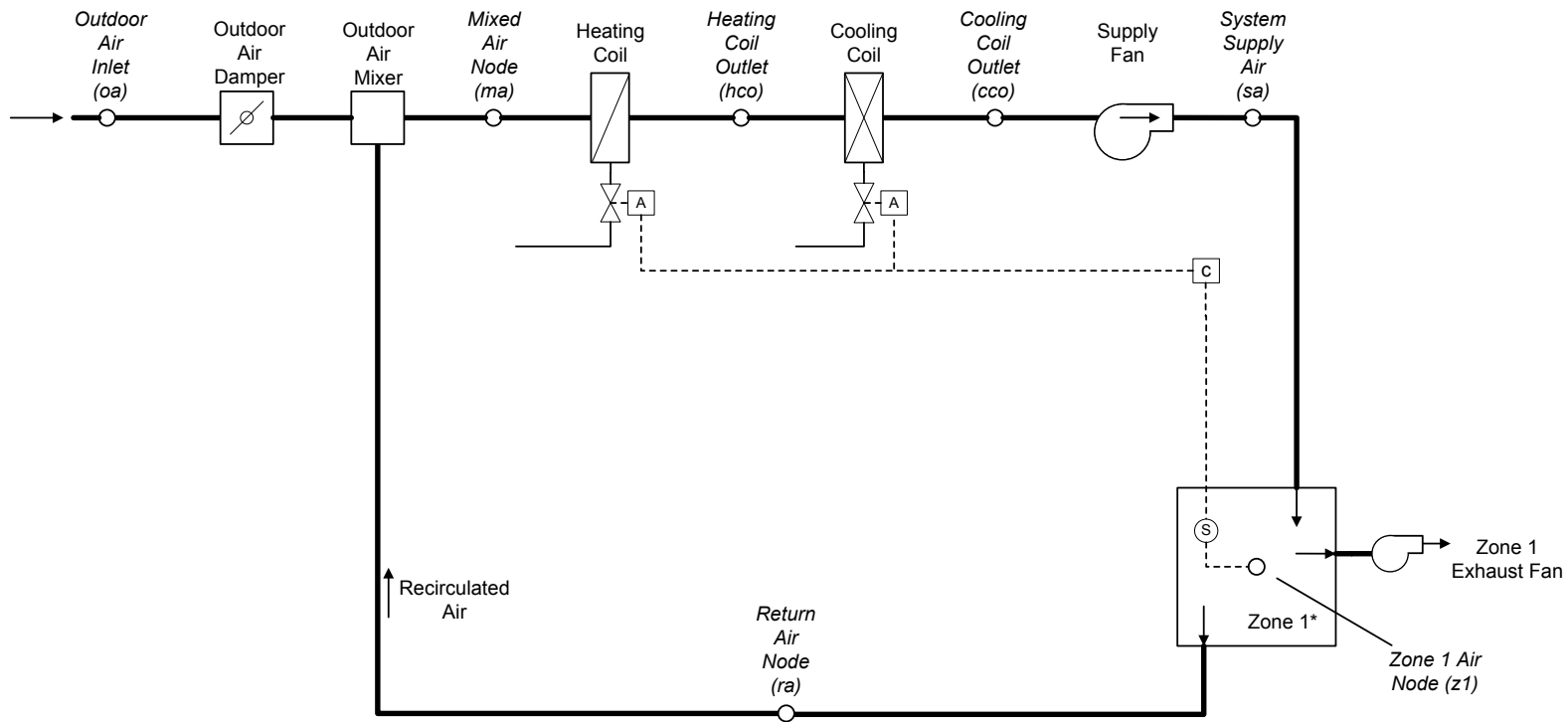


Diagram Abbreviations:  
 A = actuator; C = controller; S = sensor

\* Sensible and latent zone loads are specified for the base case in Section 1.5.5.1.1.4 and vary among the test cases. The zone thermostat senses only the zone air indoor dry-bulb temperature at the zone air node.

**Figure 1-1. Four pipe fan coil system schematic (FC)**

**Informative Note:** Valves indicated are for a typical hydronic system and are not explicitly required by the test specification. Coils can be of any type as long as they meet the operational requirements of the test specification.



**1.5.5.1.1.1 Fan Operation.** The system has a supply fan and an exhaust fan. There is no return fan.

**1.5.5.1.1.1.1 Supply Fan.** The supply fan characteristics shall be:

- a. The fan is located downstream of the heating and cooling coils, as specified in Figure 1-1.
- b. The supply fan total pressure rise = 2.0 inches wg (498 Pa).
- c. The fan provides a constant volume of supply air, measured at the fan inlet, as specified in Section 1.5.5.1.1.2.
- d. The fan operates continuously.
- e. Fan energy is transferred to the air that is being moved.
- f. Fan motor and transmission drive energy loss is not transferred to the moving air.
- g.  $(\text{fan heat to airstream}) = H_o \times (1 / (\text{fan mechanical efficiency}) - 1)$ , where:
  - $H_o$  is the fan output power required to meet the specified air flow requirements; see  $H_o$  under *fan mechanical efficiency* in Section 1.5.3.1.
  - Fan mechanical efficiency = 0.7, constant value; fan mechanical efficiency is defined in Section 1.5.3.1.
- h.  $(\text{fan motor input power}) = H_o / ((\text{fan mechanical efficiency}) \times (\text{transmission drive efficiency}) \times (\text{motor efficiency}))$ , where:
  - Transmission drive efficiency  $\leq 1.0$ ; transmission drive efficiency is defined in Section 1.5.3.1.
  - Motor efficiency  $\leq 1.0$ ; motor efficiency is defined in Section 1.5.3.1.

If the program being tested cannot model the transmission drive and motor losses as occurring outside the air stream, the fan, transmission drive, and motor efficiency losses shall be specified to have a combined efficiency of 0.7 and to occur in the air stream.

**Informative Note:** The air stream heat gain (fan heat) affects the test case outputs, while the motor and transmission drive input energy do not; therefore, the transmission and motor efficiency may be any value as long their heat is not imparted to the air stream, and the fan mechanical efficiency = 0.7.

**Informative Note:** The pressure rise and the efficiency of the fans in the fan-coil units are not realistic. The same values that are used for the larger systems (specified in later sections) are used here to simplify the task of those who use this specification.

**1.5.5.1.1.1.2 Exhaust Fan.** The zone exhaust fan shall maintain airflow as specified in Section 1.5.5.1.1.2.

**Informative Note:** Exhaust fan details are not defined explicitly as the exhaust fan's characteristics have no impact on the results of the test case.

**1.5.5.1.1.1.3 Programs with Pre-Determined Fan Modeling Assumptions.** Programs with pre-determined assumptions shall be permitted to apply those assumptions.

**Informative Note:** For example, a program may model the listed system supply airflow rate (see Section 1.5.5.1.1.2) using its default entering air conditions if they are not normally adjustable by a typical program user.

**1.5.5.1.1.2 Airflows.** System and zone airflows shall be as shown in Table 1-1 and the following subsections.

**Table 1-1. Case AE101 System and Zone Airflows**

<b>Input Parameter</b>	<b>SI Units</b>	<b>I-P Units</b>
System Supply Airflow Rate	283.17 L/s <sup>a</sup>	600 cfm <sup>a</sup>
Zone Exhaust Airflow Rate	94.39 L/s <sup>b</sup>	200 cfm <sup>b</sup>

<sup>a</sup> Volumetric airflow rate at the supply fan inlet conditions; see Section 1.5.5.1.1.2.1.

<sup>b</sup> Volumetric airflow rate at the exhaust fan inlet conditions; see Section 1.5.5.1.1.2.2.

**1.5.5.1.1.2.1 System Supply Air.** The system supply airflow rate shall be volumetrically constant and is measured at the supply fan inlet.

*Informative Note:* As the temperature and humidity ratio of the air entering the supply fan change, the specific volume of that air changes. This means that the mass flow rate of supply air, while constant for a given steady-state test case, will vary among the test cases based upon fan inlet conditions. The mass flow rate of the zone supply air is equal to that of the system supply air.

*Informative Note:* The QAS calculates the system mass flow rate from the volumetric flow using the local specific volume of air entering the supply fan. Results differences versus the QAS can be caused by differences in the method and assumptions a tested program uses to convert volumetric flow to mass flow. Example results for the QASs, including detailed outputs, e.g., mass flow rate, specific volume, enthalpy, etc., at specific system locations, are provided in Part IV; assumptions of the QAS for converting volumetric flows to mass flows are provided in Part II.

**1.5.5.1.1.2.2 Zone Exhaust Air.** The zone exhaust airflow rate shall be volumetrically constant and is measured at the exhaust fan inlet conditions.

*Informative Note:* The QASs (see Part II) calculate the exhaust air mass flow rate from the volumetric flow using the local specific volume of the air entering the exhaust fan (i.e., the zone air properties).

*Informative Note:* As the temperature and humidity ratio of the air entering the exhaust fan change, the specific volume of that air changes. This means that the mass flow rate of exhaust air, while constant for a given steady-state test case, will vary among the test cases based upon zone conditions.

**1.5.5.1.1.2.3 Outdoor Air.** The flow of outdoor air shall be introduced at a mass flow rate equal to the zone exhaust air mass flow rate. For programs that do not precisely apply the specified mass flow balance, introduction of outdoor air to replace the specified exhaust airflow (see Table 1-1), as typically applied by the program, shall be permitted.

**1.5.5.1.1.2.4 Frictionless Ducts, Coils, and Dampers.** Airflow through ducts, coils, and dampers shall be frictionless, such that there shall be no pressure drops through them. If the software being tested does not allow frictionless components, model the least amount of friction in these components that the software being tested allows.

*Informative Note:* Modeling of fan heat is as described previously, in Section 1.5.5.1.1.1.

**1.5.5.1.1.2.5 HVAC System Component Air Leakage and Heat Loss.** HVAC system components—including ducts, mixing boxes, dampers, fans, and coils—shall have no air leakage, and shall have no heat exchange (gains or losses) with their external surroundings. If the software being tested does not allow zero system air leakage or zero external heat gains or losses for HVAC system

components, model the least amount of air leakage and external heat exchange that the software being tested allows.

**Informative Note:** Modeling of exhaust and outdoor airflows is as described previously in Sections 1.5.5.1.1.2.2 and 1.5.5.1.1.2.3. Modeling of heating and cooling coils is as described below in Section 1.5.5.1.1.3.

**1.5.5.1.1.3 Operation of Heating and Cooling Coils.** The heating coil and the cooling coil shall be modeled with the following characteristics:

- a. All coils maintain set point precisely, without a throttling range. **Informative Note:** If the program being tested requires an input for throttling range, apply its minimum allowed value, and adjust the coil set point so that supply air is delivered at the specified temperature. This may require adjustment for each test case.
- b. All coils have adequate capacity to meet the coil loads.
- c. Heating and cooling coils modulate to meet the zone sensible load, delivering air at the temperature needed to maintain the zone thermostat set point.
- d. Cooling coil bypass factor (BF) = 0.

For building energy analysis programs not capable of directly modeling a cooling coil with a bypass factor of zero, a model shall be developed that applies the lowest bypass factor that the software being tested allows. Programs that do not have a bypass factor input shall apply a coil model that maximizes the cooling coil leaving air relative humidity when cooling coil latent load is present. For such programs, to approximate BF = 0 for each steady-state test case, variation of coil parameters among the test cases shall be allowed.

**Informative Note:** BF = 0 means that when the cooling coil leaving air temperature is less than the dew-point temperature of the air entering the cooling coil, the air will leave the coil at 100% relative humidity. If the cooling coil leaving air temperature is greater than the entering air dew-point temperature, no condensation will occur. When cooling coil latent load is present, the cooling coil leaving air should be 100% saturated. This is equivalent to having an apparatus dew point equal to the required coil leaving air temperature. To achieve saturation, programs with physical coil models should try to maximize the size of the coil and modulate the coil temperature or flow so the apparatus dew point is as close as possible to the required leaving air temperature while still delivering the proper amount of cooling. The QAS assumes that any condensate that forms is cooled from the entering air dew-point temperature to the coil leaving air temperature (see Part II, Section 2.2.1.26.1). Other assumptions regarding leaving condensate temperature may be reasonable.

**Informative Note:** For programs that do not accept a bypass factor input or do not accept BF near 0, modeling with an enlarged hydronic coil, with water temperature reset to meet the required supply air temperature, may be better than modeling with a direct expansion system coil unless the direct expansion coil model is capable of modulating continuously from 0 to 100% output.

**1.5.5.1.1.4 Zone Definition.** For programs that cannot specify zone loads directly and cannot define an adiabatic zone with negative sensible internal gains, skip the remainder of this section and model the zone by applying the alternative non-adiabatic zone specified in Section 1.5.5.1.1.5.

A single zone shall be modeled. The zone shall be defined by an ideal steady-state sensible heating load and a latent load as specified in Table 1-2:

**Table 1-2. Case AE101 Zone Loads**

	<b>SI Units</b>	<b>I-P Units</b>
Zone Sensible Heating Load <sup>a</sup>	2931 W	10000 Btu/hr
Zone Latent Load <sup>a,b</sup>	586.1 W	2000 Btu/hr

<sup>a</sup> Zone sensible heating load and zone latent load are defined in Section 1.5.3.1.

<sup>b</sup> Zone latent load is applied at the zone air temperature specified in Section 1.5.5.1.1.6.

Zone sensible heating load, as defined in Section 1.5.3.1, shall be modeled directly or as negative sensible internal gains within an adiabatic zone. Zone latent load, as defined in Section 1.5.3.1, is applied at the given zone air temperature. If the program being tested allows, intermediate output shall be checked to ensure the specified zone loads are modeled precisely. **Informative Note:** Latent internal gains can be modeled directly, or by specifying the number of occupants in a zone and the latent load per person.

**Informative Note:** Some programs may allow negative sensible internal gains, which can be applied to create a heating load. The physical process can be visualized as heat transfer associated with a zone with a refrigeration case that has a remote compressor.

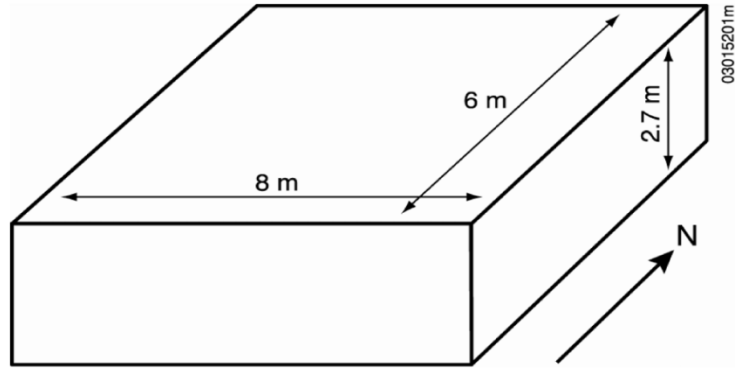
**Informative Note:** There are many approaches to achieving the zone load requirements. The preferred approach is to model the constant sensible and latent zone loads directly as loads on the system with no physical zone, or as internal gains to a zone with an adiabatic thermal envelope.

**1.5.5.1.1.5 Alternate Zone Specification.** If the program being tested was able to model this test case as specified in Section 1.5.5.1.1.4, skip this section and proceed to Section 1.5.5.1.1.6. For programs that require a non-adiabatic zone or that cannot model negative internal gains, the alternate zone specified in this section and its subsections shall be applied. When implementing this zone definition, zone loads in the model output shall be examined and input adjustments shall be permitted, such that modeled zone loads (internal gains plus zone envelope heat transfer) match those specified in Section 1.5.5.1.1.4.

**Informative Note:** Specified zone loads vary among the test cases.

**Informative Note:** The zone geometry, constructions, and default combined surface coefficients are similar to those developed in ASHRAE Standard 140-2014, Section 5.3.1 (ANSI/ASHRAE 2014), except insulation levels and internal gains assumptions are modified here.

**1.5.5.1.1.5.1 Zone Geometry.** The base zone geometry plan (see Figure 1-2) is a 6-m by 8-m zone with 48 m<sup>2</sup> floor area. The wall height is 2.7 m, and the zone air volume is 129.6 m<sup>3</sup>. The roof, floor, and four walls are specified with the exterior surface exposed to ambient air. **Informative Note:** This is as if the entire zone were suspended above the ground.



**Figure 1-2. Alternate Specification Zone Geometry**

**1.5.5.1.1.5.2 Zone Envelope Thermal Properties.** Table 1-3 lists the zone envelope construction and material properties in SI and I-P units, respectively.

Materials of the space shall have no thermal or moisture capacitance, and there is not moisture diffusion through them. If the software being tested requires inputs for thermal capacitance, moisture capacitance, or moisture diffusion, the minimum values the software being tested allows shall be applied.

**Informative Note:** When the alternate zone specification is modeled, the zone heat loss will contribute to the zone sensible load but does not exactly match the test case zone load specified in Section 1.5.5.1.1.4. Supplementary internal gains are specified in Section 1.5.5.1.1.5.3, so that total zone sensible load is as specified in Section 1.5.5.1.1.4.

**Table 1-3. Case AE101 Component Construction - Roof, Wall, and Floor**

Element	SI Units		I-P Units	
	R ((m <sup>2</sup> ·K)/W)	U (W/(m <sup>2</sup> ·K))	R ((h·ft <sup>2</sup> ·°F)/Btu)	U (Btu/(h·ft <sup>2</sup> ·°F))
Int Surf. Coef.	0.1206	8.2900	0.6849	1.4600
Insulation	2.7000	0.3704	15.3306	0.0652
Ext. Surf. Coef.	0.0328	30.4872	0.1862	5.3693
<b>Total air – air</b>	<b>2.8534</b>	<b>0.3505</b>	<b>16.2018</b>	<b>0.0617</b>
<b>Heat Loss Summary</b>				
Component	Area (m <sup>2</sup> )	UA (W/K)	Area <sup>a</sup> (ft <sup>2</sup> )	UA (Btu/(hr·°F))
Roof	48.0	16.822	516.67	31.890
Wall	75.6	26.494	813.75	50.226
Floor	48.0	16.822	516.67	31.890
Infiltration		0.000		0.000
<b>Total</b>	<b>123.6</b>	<b>43.316</b>	<b>1330.42</b>	<b>82.116</b>
<b>Supplementary Internal Gains</b>				
Sensible	82.9 W		283 Btu/hr	
Latent <sup>b</sup>	586.1 W		2000 Btu/hr	

<sup>a</sup> I-P areas converted from SI values.

<sup>b</sup> Supplementary latent internal gain is applied at the zone air temperature specified in Section 1.5.5.1.1.6.

**1.5.5.1.1.5.3 Supplementary Internal Gains.** To achieve the required test case load using the alternate zone specification, the zone supplementary internal gains listed in Table 1-3 shall be modeled. Supplementary latent internal gains are applied at the given zone air temperature. So that the modeled sensible and latent zone loads for the program being tested agree with the values specified in Section 1.5.5.1.1.4, adjustment of the supplementary internal gains shall be permitted. **Informative Note:** Latent internal gains can be modeled directly, or by specifying the number of occupants and the latent load per person.

**Informative Note:** Program treatment of the zone thermal envelope, particularly surface heat transfer, may vary from the assumptions used here. It is recommended that model output be checked to verify the correct zone loads are being modeled and, if not, that adjustments be made to the internal gains inputs so that exactly correct loads are modeled.

**1.5.5.1.1.5.4 Interior Combined Surface Coefficients.** Interior combined surface coefficients shall be  $8.29 \text{ W}/(\text{m}^2\cdot\text{K})$  ( $1.46 \text{ Btu}/(\text{h}\cdot\text{ft}^2\cdot^\circ\text{F})$ ) (ASHRAE 2005). This value shall be applied to the interior side of the exterior surfaces (walls, floor, and ceiling).

If the program being tested allows direct user input of combined interior surface coefficients, then skip the remainder of this paragraph. If the program allows direct user input of convective surface coefficients, but allows only automatically calculated surface infrared radiative exchange, the convective surface coefficient shall be  $8.29 \text{ W}/(\text{m}^2\cdot\text{K})$  ( $1.46 \text{ Btu}/(\text{h}\cdot\text{ft}^2\cdot^\circ\text{F})$ ) and the surface infrared emittance shall be 0 or as close to zero as the program allows. If the program does not allow direct user input of convective surface coefficients, the input infrared emittance shall be 0.9 and the convective surface coefficient that the program being tested automatically calculates shall be used.

**1.5.5.1.1.5.5 Exterior Combined Surface Coefficients.** Exterior combined surface coefficients shall be  $30.4872 \text{ W}/(\text{m}^2\cdot\text{K})$  ( $5.3694 \text{ Btu}/(\text{h}\cdot\text{ft}^2\cdot^\circ\text{F})$ ). This value shall be applied to all exterior surfaces, including the floor. **Informative Note:** This value corresponds with 4.3 m/s wind speed in the weather data for a rough (brick or rough plaster) surface (Judkoff and Neymark 1995; Walton 1983).

If the program being tested allows direct user input of combined exterior surface coefficients, then skip the remainder of this paragraph. If the program allows direct user input of convective surface coefficients but allows only automatically calculated surface infrared radiative exchange, then the convective surface coefficient shall be  $30.4872 \text{ W}/(\text{m}^2\cdot\text{K})$  ( $5.3694 \text{ Btu}/(\text{h}\cdot\text{ft}^2\cdot^\circ\text{F})$ ) and the surface infrared emittance shall be 0 or as close to zero as the program allows. If the program does not allow direct user input of convective surface coefficients, then the input infrared emittance shall be 0.9 and the convective surface coefficient that the program automatically calculates shall be used.

**1.5.5.1.1.6 Zone Temperature and Thermostat Set Point.** The zone air temperature shall be:

$$\text{Zone dry-bulb temperature} = 21.111^\circ\text{C} (70.0^\circ\text{F})$$

**1.5.5.1.1.6.1** There shall be no zone humidity control.

**Informative Note:** This means that the zone humidity will float in accordance with moisture in the outdoor air introduced by the system, zone latent load, and moisture removal by the mechanical system.

**1.5.5.1.1.6.2** The thermostat shall sense only the zone air temperature; the thermostat itself shall not sense any radiant heat transfer exchange with the interior surfaces.

**1.5.5.1.1.6.3** The controls for this system are ideal in that the equipment shall maintain the set point exactly when it is operating. There are no minimum ON or OFF time-duration requirements for the unit and no hysteresis control band (e.g., there is no ON at set point +  $x^{\circ}\text{C}$  or OFF at set point –  $y^{\circ}\text{C}$ ). If the software being tested requires input for these, the minimum values the software being tested allows shall be used. To eliminate temperature offset so that the modeled zone air temperature agrees exactly with the test case specification, adjustment of the modeled zone thermostat set point shall be permitted, and such adjustment shall be allowed to vary among test cases.

**1.5.5.1.1.6.4** The thermostat is nonproportional.

**Informative Note:** A nonproportional thermostat operates such that when the conditioned zone air temperature exceeds the thermostat cooling set point, the heat extraction rate is assumed to equal the maximum capacity of the cooling equipment corresponding to environmental conditions at the time of operation. A proportional thermostat throttles the heat difference between the zone set point temperature and the actual zone temperature. A proportional thermostat model can be made to approximate a nonproportional thermostat model by setting a very small throttling range (the minimum allowed by the software being tested).

**1.5.5.1.1.7 Ambient Conditions.** The ambient conditions specified in Table 1-4 shall be used.

**Table 1-4. Case AE101 Ambient Conditions**

	SI Units	I-P Units
Dry-Bulb Temperature	-29.0°C	-20.20°F
Dew-Point Temperature <sup>a</sup>	-29.0°C	-20.20°F
Solar Radiation	None	
Wind Speed	4.3m/s	9.62 mph
Atmospheric Pressure <sup>b</sup>	101.325 kPa	14.696 psia
Elevation (Sea Level) <sup>b</sup>	0 m	0 ft
Equivalent Ambient Moisture Indicators <sup>a</sup>		
Humidity Ratio	0.000260	0.000260
Wet-Bulb Temperature	-29.000°C	-20.200°F
Relative Humidity	100.000%	100.000%
Relative Humidity (TM2) <sup>c</sup>	100%	100%

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file indicates 0 m altitude, consistent with the given pressure. However, the weather file atmospheric pressure precision is limited to four digits by the weather file format such that 1013 millibars (101300 Pa) is listed in the weather file. The effect of modeling with 101325 Pa versus 101300 Pa is negligible (< 0.05% on cooling coil latent loads) as described in Appendix A, Table A-3.

<sup>c</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

If the tested program does not allow constant steady-state ambient conditions to be input directly, the TMY2-format weather data provided with the following file shall be applied:

**AE101.TM2**

These data are provided with the electronic files accompanying this report and are described in Appendix A.

**Informative Note:** Other cases call for different weather files as needed.

**Informative Note:** Some programs allow constant steady-state ambient conditions to be input directly for design-day calculations.

**1.5.5.1.1.8 Output Requirements.** The required outputs for this case shall be as specified in Section 1.6.5.1.

**1.5.5.1.2 Fan Coil System Parametric Variations**

**1.5.5.1.2.1 Case AE103: Low Cooling, Dry Coil 1**

**Informative Note, Objective.** Test model treatment of a four-pipe fan coil air system with low sensible cooling load and cool dry outdoor air.

**Informative Note, Method.** Use Case AE101 model with modifications. Dry outdoor air and low sensible and latent loads used in the case ensure sensible only cooling and isolate that portion of the cooling calculation. Compare cooling coil loads with the QAS and with other appropriate test case results.

**1.5.5.1.2.1.1 Input Specification.** This case shall be identical to Case AE101 (See Section 1.5.5.1.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-5; only values that change are shown in the table.

**Table 1-5. Case AE103 Input Parameters**

<b>Ambient Input Parameters</b>	<b>SI Units</b>	<b>I-P Units</b>
Ambient Dry-Bulb Temperature	15.5°C	59.90°F
Ambient Dew-Point Temperature <sup>a</sup>	-3.0°C	26.60°F
Equivalent Ambient Moisture Indicators <sup>a</sup>		
Ambient Humidity Ratio	0.002948	0.002948
Ambient Wet-Bulb Temperature	7.206°C	44.971°F
Ambient Relative Humidity	27.028 %	27.028 %
Ambient Relative Humidity (TM2) <sup>b</sup>	27%	27 %
<b>Zone Input Parameters</b>		
Zone Sensible Cooling Load <sup>c</sup>	1465 W	5000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.333°C	74°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format and is not as precise as the other equivalent moisture variables.



Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent load as defined in Section 1.5.3.1 is applied at the given zone air temperature.

- b. **Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

### AE103.TM2

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. **Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.1.1.5 shall apply changes to parameters as indicated in Table 1-6; only values that change are shown in the table. Check output and adjust modeled internal gains so total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-5, and zone latent cooling load (latent internal gains) is equivalent to that specified in Table 1-3 (Case AE101).

**Table 1-6. Case AE103 Alternate Zone Definition Input Parameters**

	SI Units	I-P Units
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W	567.8 (hr·ft <sup>2</sup> ·°F)/Btu
Supplementary Internal Gains – Sensible	1478.8 W	5045.8 Btu/hr

**Informative Note:** For cooling cases, a near-adiabatic zone is applied through use of insulation with high thermal resistance to facilitate precise system loading. Use of a near-adiabatic zone was not possible for the heating cases because some programs may not allow direct input of negative internal gains.

#### 1.5.5.1.2.2 Case AE104: High Cooling, Wet Coil 1

**Informative Note, Objective.** Test model treatment of a four-pipe fan coil air system with high sensible cooling load and warm, humid outdoor air.

**Informative Note, Method.** Use Case AE101 model with modifications. Humid ambient conditions and high sensible cooling load ensure a cold cooling coil with high latent load. Check cooling coil leaving air saturation to verify coil bypass factor is zero or near zero. Compare cooling coil load with the QAS and with other appropriate test case results.

**1.5.5.1.2.2.1 Input Specification.** This case shall be identical to Case AE101 (See Section 1.5.5.1.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-7; only values that change are shown in the table.

**Table 1-7. Case AE104 Input Parameters**

<b>Ambient Input Parameters</b>	<b>SI Units</b>	<b>I-P Units</b>
Ambient Dry-Bulb Temperature	26.9°C	80.42°F
Ambient Dew-Point Temperature <sup>a</sup>	22.1°C	71.78°F
Equivalent Ambient Moisture Indicators <sup>a</sup>		
Ambient Humidity Ratio	0.016849	0.016849
Ambient Wet-Bulb Temperature	23.441°C	74.194°F
Ambient Relative Humidity	75.023 %	75.023 %
Ambient Relative Humidity (TM2) <sup>b</sup>	75 %	75 %
<b>Zone Input Parameters</b>		
Zone Sensible Cooling Load <sup>c</sup>	2931 W	10000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.889°C	75°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent load as defined in Section 1.5.3.1 is applied at the given zone air temperature.

- b. Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

**AE104.TM2**

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.1.1.5, shall apply changes to parameters as indicated in Table 1-8; only values that change are shown in the table. Check output and adjust modeled internal gains so total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-7, and zone latent cooling load (latent internal gains) is equivalent to that specified in Table 1-3 (Case AE101).

**Table 1-8. Case AE104 Alternate Zone Definition Input Parameters**

	<b>SI Units</b>	<b>I-P Units</b>
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W	567.8 (hr·ft <sup>2</sup> ·°F)/Btu
Supplementary Internal Gains – Sensible	2925.6 W	9982.4 Btu/hr

### 1.5.5.2 Single Zone (SZ) Air System Cases (AE200 series).

The ability to model an SZ air system shall be tested as described in this section. If the software being tested is capable of applying a variety of system models to address an SZ system, the system model that is most similar to the SZ system specified below shall be applied. The selected system model shall be capable of economizer function.

**Informative Note:** The user may test other possible modeling approaches (available system models) in this context, as appropriate to the software being tested.

**Informative Note:** The progression of the first three test cases in this series (variation of zone loads, temperatures, and ambient conditions) follows the first three test cases in the AE100 (FC) series.

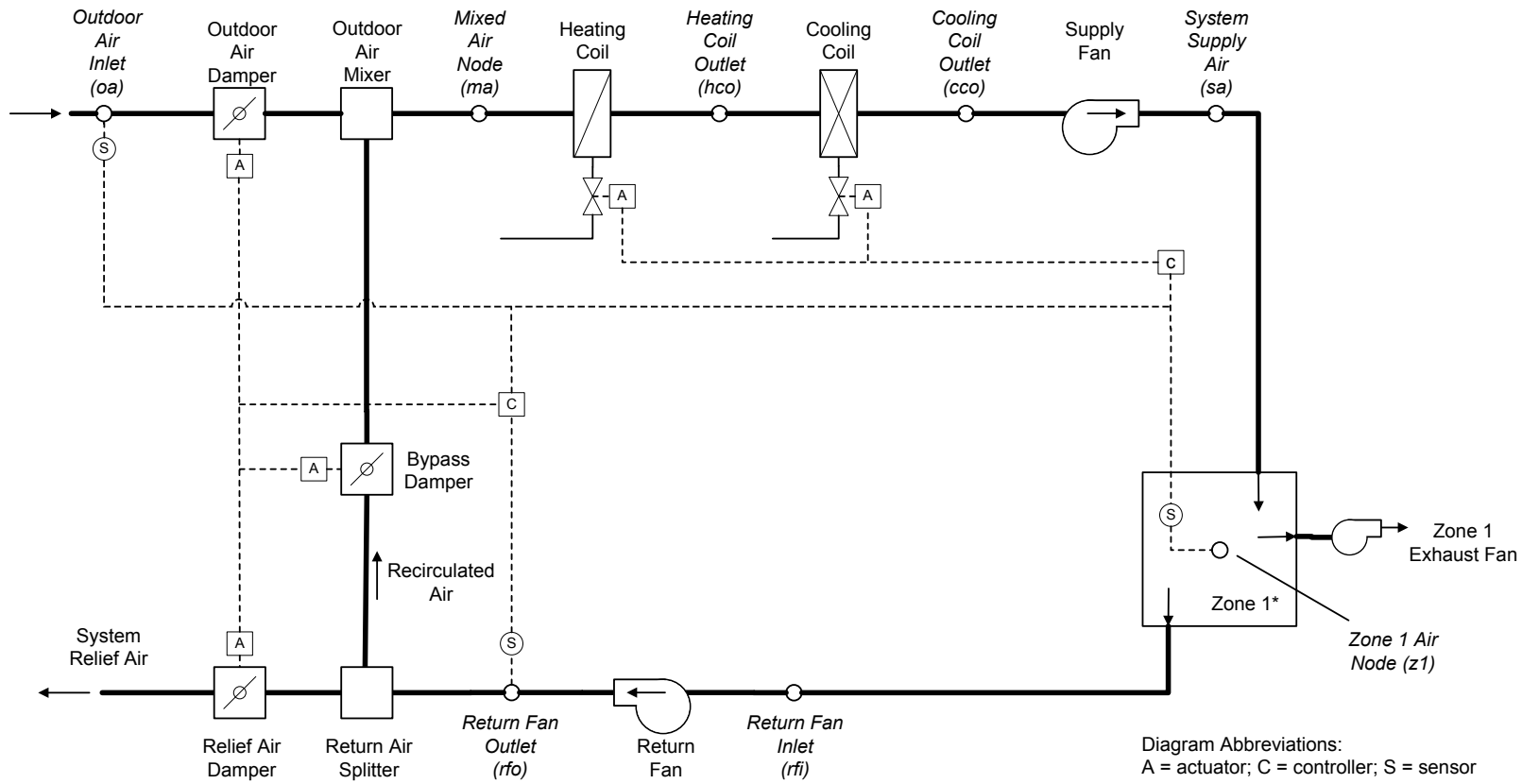
#### 1.5.5.2.1 Case AE201: Base Case, High Heating 1

Case AE201 shall be modeled as specified in this section and its subsections. The system configuration shall be modeled as presented in the schematic diagram of Figure 1-3. System input parameters shall be as described in the following sections. **Informative Note:** The test specifications for the heating and cooling coils, zone definition, alternate zone specifications, zone temperature and thermostat set points, and ambient conditions (Sections 1.5.5.2.1.3 through 1.5.5.2.1.7) are the same as those for the FC system (Sections 1.5.5.1.1.3 through 1.5.5.1.1.7).

**Informative Note, Objective.** Test model treatment of an SZ air system with high sensible heating load and cold exterior air.

**Informative Note, Method.** An SZ air system with supply fan, return fan, and heating and cooling coils, conditions a single zone that has constant sensible and latent zone loads. The model is run at specified constant outdoor and indoor conditions. Resulting coil loads are compared with the QAS, with other appropriate test case results (e.g., AE101), and with other example results (see Part IV).

**Informative Note:** In this base case, no economizer function is modeled. The economizer function is tested in later cases.



\* Sensible and latent zone loads are specified for the base case in Section 1.5.5.2.1.4, and vary among the test cases. The zone thermostat senses only the zone air indoor dry-bulb temperature at the zone air node.

**Figure 1-3. SZ system schematic**

**Informative Note:** Valves indicated are for a typical hydronic system and are not explicitly required by the test specification. Coils can be of any type as long as they meet the operational requirements of the test specification.

### 1.5.5.2.1.1 Fan Operation.

The system has a supply fan, return fan, and a zone exhaust fan.

**1.5.5.2.1.1.1 Supply Fan.** The supply fan characteristics shall be:

- a. The fan is located downstream of the heating and cooling coils, as specified in Figure 1-3.
- b. The supply fan total pressure rise = 2.0 inches wg (498 Pa).
- c. The fan provides a constant volume of supply air, measured at the fan inlet, as specified in Section 1.5.5.2.1.2.
- d. The fan operates continuously.
- e. Fan energy is transferred to the air that is being moved.
- f. Fan motor and transmission drive energy loss is not transferred to the moving air.
- g. (fan heat to airstream) =  $H_o \times (1 / (\text{fan mechanical efficiency}) - 1)$ , where
  - $H_o$  is the fan output power required to meet the specified air flow requirements; see  $H_o$  under *fan mechanical efficiency* in Section 1.5.3.1.
  - Fan mechanical efficiency = 0.7, constant value; fan mechanical efficiency is defined in Section 1.5.3.1.
- h. (fan motor input power) =  $H_o / ((\text{fan mechanical efficiency}) \times (\text{transmission drive efficiency}) \times (\text{motor efficiency}))$ , where:
  - Transmission drive efficiency  $\leq 1.0$ ; transmission drive efficiency is defined in Section 1.5.3.1.
  - Motor efficiency  $\leq 1.0$ ; motor efficiency is defined in Section 1.5.3.1.

If the program being tested cannot model the transmission drive and motor losses as occurring outside the air stream, the fan, transmission drive, and motor efficiency losses shall be specified to have a combined efficiency of 0.7 and to occur in the air stream.

**Informative Note:** The air stream heat gain (fan heat) affects the test case outputs, while the motor and transmission drive input energy do not; therefore, the transmission and motor efficiency may be any value as long their heat is not imparted to the air stream, and the fan mechanical efficiency = 0.7.

**Informative Note:** The supply fan characteristics are the same as for the FC system (see Section 1.5.5.1.1.1)

**1.5.5.2.1.1.2 Return Fan.** The return fan characteristics shall be:

- a. The fan is located downstream from the zone and before the return air splitter, as specified in Figure 1-3.
- b. The return fan total pressure rise = 1.0 inches wg (249 Pa) at 400 cfm (188.78 L/s).
- c. The return fan total pressure rise varies as:  
Pressure rise [inches wg] =  $(1.0 \text{ inches wg}) \times ((\text{actual return fan cfm}) / (400 \text{ cfm}))^2$ .
- d. The fan returns an air volume as specified in Section 1.5.5.2.1.2.
- e. The fan operates continuously.
- f. Fan energy is transferred to the air that is being moved.
- g. Fan motor and transmission drive energy loss is not transferred to the moving air.
- h. (fan heat to airstream) =  $H_o \times (1 / (\text{fan mechanical efficiency}) - 1)$ , where
  - $H_o$  is the fan output power required to meet the specified air flow requirements; see  $H_o$  under *fan mechanical efficiency* in Section 1.5.3.1

- Fan mechanical efficiency = 0.7, constant value; fan mechanical efficiency is defined in Section 1.5.3.1
- i. (fan motor input power) =  $H_o / ((\text{fan mechanical efficiency}) \times (\text{transmission drive efficiency}) \times (\text{motor efficiency}))$ , where:
- Transmission drive efficiency  $\leq 1.0$ ; transmission drive efficiency is defined in Section 1.5.3.1
  - Motor efficiency  $\leq 1.0$ ; motor efficiency is defined in Section 1.5.3.1.

If the program being tested cannot model the transmission drive and motor losses as occurring outside the air stream, the fan, transmission drive, and motor efficiency losses shall be specified to have a combined efficiency of 0.7 and to occur in the air stream.

**Informative Note:** The air stream heat gain (fan heat) affects the test case outputs while the motor and transmission drive input energy do not; therefore, the transmission and motor efficiency may be any value as long their heat is not imparted to the air stream, and the fan mechanical efficiency = 0.7.

**1.5.5.2.1.1.3 Exhaust Fan.** The zone exhaust fan shall maintain airflow as specified in Section 1.5.5.2.1.2.

**Informative Note:** Exhaust fan details are not defined explicitly as the exhaust fan’s characteristics have no impact on the results of the test case.

**1.5.5.2.1.1.4 Programs with Pre-Determined Fan Modeling Assumptions.** Programs with pre-determined fan modeling assumptions shall be permitted to apply those assumptions.

**Informative Note:** For example, a program may model the return air fan pressure increase using its default assumptions if they are not normally adjustable by a typical program user.

**1.5.5.2.1.2 Airflows.**

System and zone airflows shall be as shown in Table 1-9, and the following subsections.

**Table 1-9. Case AE201 System and Zone Airflows**

<b>Input Parameter</b>	<b>SI Units</b>	<b>I-P Units</b>
System Supply Airflow Rate	283.17 L/s <sup>a</sup>	600 cfm <sup>a</sup>
Design System Return Airflow Rate	188.78 L/s <sup>b</sup>	400 cfm <sup>b</sup>
Zone Exhaust Airflow Rate	94.39 L/s <sup>c</sup>	200 cfm <sup>c</sup>

<sup>a</sup> Volumetric airflow rate at the supply fan inlet conditions; see Section 1.5.5.2.1.2.1.

<sup>b</sup> This is the volumetric airflow rate upon which the return fan pressure rise (1 inch wg at 400 cfm) is based; see Section 1.5.5.2.1.2, items b and c. The actual return airflow rate shall be permitted to vary from the design airflow rate, and is defined in Section 1.5.5.2.1.2.3.

<sup>c</sup> Volumetric airflow rate at the exhaust fan inlet conditions; see Section 1.5.5.2.1.2.2.

**1.5.5.2.1.2.1 System Supply Air.** The system supply airflow rate shall be volumetrically constant and is measured at the supply fan inlet.

**Informative Note:** As the temperature and humidity ratio of the air entering the supply fan change, the specific volume of that air changes. This means that the mass flow rate of supply air, while

constant for a given steady-state test case, will vary among the test cases based upon fan inlet conditions. The mass flow rate of the zone supply air is equal to that of the system supply air.

**Informative Note:** The QAS calculates the system mass flow rate from the volumetric flow using the local specific volume of air entering the supply fan. Results differences versus the QAS can be caused by differences in the method and assumptions a tested program uses to convert volumetric flow to mass flow. Example results for the QAS, including detailed outputs, e.g., mass flow rate, specific volume, enthalpy, etc., at specific system locations, are provided in Part IV; assumptions of the QAS for converting volumetric flows to mass flows are provided in Part II.

**1.5.5.2.1.2.2 Zone Exhaust Air.** The zone exhaust airflow rate shall be volumetrically constant and is measured at the exhaust fan inlet conditions.

**Informative Note:** The QAS (see Part II) calculates the exhaust air mass flow rate from the volumetric flow using the local specific volume of the air entering the exhaust fan (i.e., the zone air properties).

**Informative Note:** As the temperature and humidity ratio of the air entering the exhaust fan change, the specific volume of that air changes. This means that the mass flow rate of exhaust air, while constant for a given steady-state test case, will vary among the test cases based upon zone conditions.

**1.5.5.2.1.2.3 Return Airflow.** The return airflow rate shall be the air volume, as measured at the return fan inlet, necessary to move an air mass equal to the zone supply mass flow minus the zone exhaust mass flow.

**Informative Note:** The return fan volumetric flow rate in the QAS (see Part II) is calculated from the return air mass flow using the specific volume of air entering the return fan. The return fan volumetric flow varies with the return air mass flow and the specific volume of the air entering the return fan.

**1.5.5.2.1.2.4 Outdoor Air.** The flow of outdoor air shall be introduced at a mass flow rate equal to the zone exhaust air mass flow rate. For programs that do not precisely apply the specified mass flow balance, introduction of outdoor air to replace the specified exhaust airflow (see Table 1-9), as typically applied by the program, shall be permitted.

**Informative Note:** Greater amounts of outdoor air may be required in other test cases applying economizer control logic.

**1.5.5.2.1.2.5 Frictionless Ducts, Coils, and Dampers.** Airflow through ducts, coils, and dampers shall be frictionless, such that there shall be no pressure drops through them. If the software being tested does not allow frictionless components, model the least amount of friction in these components that the software being tested allows.

**Informative Note:** Modeling of fan heat is as described previously, in Section 1.5.5.2.1.1.

**1.5.5.2.1.2.6 HVAC System Component Air Leakage and Heat Loss.** HVAC system components—including ducts, mixing boxes, dampers, fans, and coils—shall have no air leakage, and shall have no heat exchange (gains or losses) with their external surroundings. If the software being tested does not allow zero system air leakage or zero external heat gains or losses for HVAC system components, model the least amount of air leakage and external heat exchange that the software being tested allows.

**Informative Note:** Modeling of exhaust and outdoor airflows is as described previously in Sections 1.5.5.2.1.2.2 and 1.5.5.2.1.2.4. Modeling of heating and cooling coils is as described in Section 1.5.5.2.1.3.

**1.5.5.2.1.3 Operation of Heating and Cooling Coils.** The heating and cooling coils shall be modeled as specified for the FC system test base case in Section 1.5.5.1.1.3.

**1.5.5.2.1.4 Zone Definition.** For programs that cannot specify zone loads directly and cannot define an adiabatic zone with negative internal gains, skip the remainder of this section and model the zone by applying the alternative non-adiabatic zone specified in Section 1.5.5.2.1.5.

A single zone shall be modeled as specified for the FC system test base case in Section 1.5.5.1.1.4.

**1.5.5.2.1.5 Alternate Zone Specification.** If the program being tested was able to model this test case as specified in Section 1.5.5.2.1.4, skip this section and proceed to Section 1.5.5.2.1.6. For programs that require non-adiabatic zones or cannot model negative internal gains, the alternate zone specification for the FC system test base case in Section 1.5.5.1.1.5 shall be applied.

**1.5.5.2.1.6 Zone Temperature and Thermostat Set Point.** These shall be the same as for the FC system test base case as specified in Section 1.5.5.1.1.6.

**1.5.5.2.1.7 Ambient Conditions.** The ambient conditions shall be the same as for the FC system test base case as specified in Section 1.5.5.1.1.7.

**1.5.5.2.1.8 Output Requirements.** The required outputs for this case shall be as specified in Section 1.6.5.2.

## **1.5.5.2.2 Single Zone System Parametric Variations, No Economizer**

### **1.5.5.2.2.1 Case AE203: Low Cooling, Dry Coil 1**

**Informative Note, Objective.** Test model treatment of a single zone air system with low sensible cooling load and cool, dry outdoor air.

**Informative Note, Method.** Use the Case AE201 model with modifications. Use the Case AE103 zone and ambient conditions. Dry outdoor air and low sensible and latent loads used in the case ensure sensible only cooling and isolate that portion of the cooling calculation. Compare cooling coil loads with the QAS and with other appropriate test case results.

**1.5.5.2.2.1.1 Input Specification.** This case shall be identical to Case AE201 (see Section 1.5.5.2.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and temperature parameters shall be modeled as specified in Table 1-10; only values that change are shown in the table.



**Table 1-10. Case AE203 Input Parameters**

<b>Ambient Input Parameters</b>	<b>SI Units</b>	<b>I-P Units</b>
Ambient Dry-Bulb Temperature	15.5°C	59.90°F
Ambient Dew-Point Temperature <sup>a</sup>	-3.0°C	26.60°F
Equivalent Ambient Moisture Indicators <sup>a</sup>		
Ambient Humidity Ratio	0.002948	0.002948
Ambient Wet-Bulb Temperature	7.206°C	44.971°F
Ambient Relative Humidity	27.028 %	27.028 %
Ambient Relative Humidity (TM2) <sup>b</sup>	27%	27 %
<b>Zone Input Parameters</b>		
Zone Sensible Cooling Load <sup>c</sup>	1465 W	5000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.333°C	74°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent load as defined in Section 1.5.3.1 is applied at the given zone air temperature.

- b. Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

**AE103.TM2**

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.2.1.5, shall apply changes to parameters as indicated in Table 1-11; only values that change are shown in the table. Check output and adjust modeled internal gains so total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-10, and zone latent cooling load (latent internal gains) is equivalent to that specified in Table 1-3 (Case AE101).

**Table 1-11. Case AE203 Alternate Zone Definition Input Parameters**

	<b>SI Units</b>	<b>I-P Units</b>
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W	567.8 (hr·ft <sup>2</sup> ·°F)/Btu
Supplementary Internal Gains – Sensible	1478.8 W	5045.8 Btu/hr

**Informative Note:** For cooling cases a near-adiabatic zone is applied, through use of insulation with high thermal resistance, to facilitate precise system loading. Use of a near-adiabatic zone was not possible for the heating cases because some programs may not allow direct input of negative internal gains.

### 1.5.5.2.2.2 Case AE204: High Cooling, Wet Coil 1

**Informative Note, Objective.** Test model treatment of a single zone air system with high sensible cooling load and warm, humid outdoor air.

**Informative Note, Method.** Use Case AE201 model with modifications. Humid ambient conditions and high sensible cooling ensure a cold cooling coil with high latent loads. Check the cooling coil leaving air saturation to verify coil bypass factor is zero or near zero. Compare cooling coil loads with QAS and with other appropriate test case results.

**1.5.5.2.2.2.1 Input Specification.** This case shall be identical to Case AE201 (See Section 1.5.5.2.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-12; only values that change are shown in the table.

**Table 1-12. Case AE204 Input Parameters**

<b>Ambient Input Parameters</b>	<b>SI Units</b>	<b>I-P Units</b>
Ambient Dry-Bulb Temperature	26.9°C	80.42°F
Ambient Dew-Point Temperature <sup>a</sup>	22.1°C	71.78°F
Equivalent Ambient Moisture Indicators <sup>a</sup>		
Ambient Humidity Ratio	0.016849	0.016849
Ambient Wet-Bulb Temperature	23.441°C	74.194°F
Ambient Relative Humidity	75.023 %	75.023 %
Ambient Relative Humidity (TM2) <sup>b</sup>	75 %	75 %
<b>Zone Input Parameters</b>		
Zone Sensible Cooling Load <sup>c</sup>	2931 W	10000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.889°C	75°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent load as defined in Section 1.5.3.1 is applied at the given zone air temperature.

- b. **Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

#### AE104.TM2

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. **Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.2.1.5, shall apply changes to parameters as indicated in Table 1-13; only values that change are shown in the table. Check output and adjust modeled internal gains so total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-12, and zone latent cooling load (latent internal gains) is equivalent to that specified in Table 1-3 (Case AE101).

**Table 1-13. Case AE204 Alternate Zone Definition Input Parameters**

	<b>SI Units</b>	<b>I-P Units</b>
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W	567.8 (hr·ft <sup>2</sup> ·°F)/Btu
Supplementary Internal Gains – Sensible	2925.6 W	9982.4 Btu/hr

#### 1.5.5.2.2.3 Case AE205: Low Cooling, Dry Coil 1

**Informative Note, Objective.** Generate results for a single zone air system without enthalpy economizer, applying low sensible cooling load and warm, dry outdoor air, to compare with results for the same system with enthalpy economizer (Case AE245).

**Informative Note, Method.** Use the Case AE201 model with modifications. Case AE205 applies ambient conditions with a low wet-bulb temperature to ensure that when the economizer is specified in Case AE245 it will operate. Compare cooling coil loads with the QAS and with results for Case AE245 (which includes an enthalpy economizer system).

**1.5.5.2.2.3.1 Input Specification.** This case shall be identical to Case AE201 (see Section 1.5.5.2.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-14; only values that change are shown in the table.

**Table 1-14. Case AE205 Input Parameters**

<b>Ambient Input Parameters</b>	<b>SI Units</b>	<b>I-P Units</b>
Ambient Dry-Bulb Temperature	24.9°C	76.82°F
Ambient Dew-Point Temperature <sup>a</sup>	2.4°C	36.32°F
Equivalent Ambient Moisture Indicators <sup>a</sup>		
Ambient Humidity Ratio	0.004510	0.004510
Ambient Wet-Bulb Temperature	13.027°C	55.449°F
Ambient Relative Humidity	23.050 %	23.050 %
Ambient Relative Humidity (TM2) <sup>b</sup>	23 %	23 %
<b>Zone Input Parameters</b>		
Zone Sensible Cooling Load <sup>c</sup>	1465 W	5000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.333°C	74°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format, and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling

with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent load as defined in Section 1.5.3.1 is applied at the given zone air temperature.

- b. **Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

**AE105.TM2**

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. **Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.2.1.5, shall apply changes to parameters as indicated in Table 1-15; only values that change are shown in the table. Check output and adjust modeled internal gains so total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-14, and zone latent cooling load (latent internal gains) is equivalent to that specified in Table 1-3 (Case AE101).

**Table 1-15. Case AE205 Alternate Zone Definition Input Parameters**

	<b>SI Units</b>	<b>I-P Units</b>
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W	567.8 (hr·ft <sup>2</sup> ·°F)/Btu
Supplementary Internal Gains – Sensible	1462.7 W	4990.8 Btu/hr

**1.5.5.2.2.4 Case AE206: Low Cooling, Wet Coil 1**

**Informative Note, Objective.** Generate results for a single zone air system without dry-bulb economizer, applying low sensible cooling load and warm, humid outdoor air conditions, to compare with results for the same system with dry-bulb economizer (Case AE226).

**Informative Note, Method.** Use the Case AE201 model with modifications. Case AE206 applies an ambient dry-bulb temperature that ensures the economizer specified in Case AE226 will operate. Humid ambient conditions and high sensible cooling ensure a cold cooling coil with high latent loads. Check the cooling coil leaving air saturation to verify coil bypass factor is zero or near zero. Compare cooling coil loads with the QAS and with results for Case AE226 (which includes a comparative dry-bulb economizer system).

**1.5.5.2.2.4.1 Input Specification.** This case shall be identical to Case AE201 (see Section 1.5.5.2.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-16; only values that change are shown in the table.

**Table 1-16. Case AE206 Input Parameters**

<b>Ambient Input Parameters</b>	<b>SI Units</b>	<b>I-P Units</b>
Ambient Dry-Bulb Temperature	23.0°C	73.40°F
Ambient Dew-Point Temperature <sup>a</sup>	20.9°C	69.62°F
Equivalent Ambient Moisture Indicators <sup>a</sup>		
Ambient Humidity Ratio	0.015625	0.015625
Ambient Wet-Bulb Temperature	21.523°C	70.741°F
Ambient Relative Humidity	87.968 %	87.968 %
Ambient Relative Humidity (TM2) <sup>b</sup>	88 %	88 %
<b>Zone Input Parameters</b>		
Zone Sensible Cooling Load <sup>c</sup>	1465 W	5000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.333°C	74°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent load as defined in Section 1.5.3.1 is applied at the given zone air temperature.

- b. Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

**AE106.TM2**

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.2.1.5, shall apply changes to parameters as indicated in Table 1-17; only values that change are shown in the table. Check output and adjust modeled internal gains so total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-16, and zone latent cooling load (latent internal gains) is equivalent to that specified in Table 1-3 (Case AE101).

**Table 1-17. Case AE206 Alternate Zone Definition Input Parameters**

	<b>SI Units</b>	<b>I-P Units</b>
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W	567.8 (hr·ft <sup>2</sup> ·°F)/Btu
Supplementary Internal Gains – Sensible	1465.4 W	5000.0 Btu/hr

### 1.5.5.2.3 Single Zone System Parametric Variations, with Economizer

#### 1.5.5.2.3.1 Case AE226: Low Cooling, Wet Coil 1 with Return Air Comparative Dry-Bulb Economizer Outdoor Air Control

This test case shall be modeled with identical zone and system inputs as Case AE206 except for the existence of a return air comparative dry-bulb temperature economizer as described below.

**Informative Note, Objective.** Test model treatment of a single zone air system model with return air comparative dry-bulb temperature economizer at low sensible cooling load and warm, humid outdoor air.

**Informative Note, Method.** Use the Case AE206 model as defined in Section 1.5.5.2.2.4 and add the economizer as described here. Compare cooling coil loads with the QAS and with results for Case AE206.

**Informative Note:** For the given economizer control logic, at the given steady-state operating conditions some cooling loads are increased by varying amounts.

**1.5.5.2.3.1.1 Input Specification.** Case AE226 shall be identical to Case AE206 (see Section 1.5.5.2.2.4) except for the addition of a return air comparative dry-bulb temperature economizer outdoor air control as specified in Section 1.5.5.2.3.1.1.1.

**1.5.5.2.3.1.1.1 Return Air Comparative Dry-Bulb Economizer Outdoor Air Control.** When the economizer is ON, the outdoor air shall increase until either the cooling coil sensible load is eliminated, or the outdoor air is 100% of the supply airflow; if the economizer control cannot satisfy the cooling coil sensible load then the cooling coil shall operate, along with the economizer, to meet the remaining cooling coil sensible load. The air system dampers (see Figure 1-3) shall be adjusted using economizer control based on outdoor and return air dry-bulb temperature as specified below.

- Economizer = OFF IF  $ODB > RAT$  OR  $ODB \leq ODB_{\text{econo,min}}$ . In this configuration, the outdoor air mass flow rate shall equal the combined zone exhaust air mass flows.
  - Coil = ON IF  $T_{\text{zone}} >$  zone thermostat set point; see Section 1.5.5.1.1.3.c.
- Economizer = ON IF  $RAT \geq ODB > ODB_{\text{econo,min}}$ .
  - Coil = ON IF  $T_{\text{zone}} >$  zone thermostat set point; see Section 1.5.5.1.1.3.c. In this configuration the mixed air shall be 100% outdoor air.
  - Coil = OFF IF  $T_{\text{zone}} \leq$  zone thermostat set point; see Section 1.5.5.1.1.3.c. In this configuration the air system dampers shall modulate the flow of outdoor air to meet the zone thermostat set point, and the outdoor airflow shall be permitted to vary from 100% outdoor air to the minimum flow necessary to replace the zone exhaust air mass flow.

Where for:

- Economizer = ON, outdoor air shall be provided as needed up to 100% outdoor air, but not less than the minimum required outdoor airflow rate at any time.
- Economizer = OFF, outdoor air shall be provided at the minimum required outdoor airflow rate.

- Coil = ON, the cooling coil shall operate only as necessary to satisfy the cooling coil sensible load not compensated by the economizer.
- Coil = OFF, the cooling coil shall not operate.

And where:

- ODB: outdoor air dry-bulb temperature
- $ODB_{\text{econo,min}}$ : the outdoor dry-bulb temperature where at the minimum required outdoor airflow rate  $T_{\text{zone}} = \text{zone thermostat set point}$ .
- RAT: return air dry-bulb temperature, at the return fan outlet
- $T_{\text{zone}}$ : zone air dry-bulb temperature. As used above, it is the zone air temperature if the coil did not operate.

**Informative Note:** Because the cooling coil activates (cooling coil load occurs) when  $T_{\text{zone}}$  rises above the thermostat set point, an equivalent economizer control scheme is:

- Economizer = OFF AND Coil = ON IF cooling coil sensible load  $> 0$  and  $ODB > RAT$ .
- Economizer = ON AND Coil = ON IF cooling coil sensible load  $> 0$  AND  $ODB \leq RAT$  AND all cooling coil sensible load is NOT compensated by the economizer. (In this configuration the mixed air is 100% outdoor air.)
- Economizer = ON AND Coil = OFF IF cooling coil sensible load  $> 0$  AND  $ODB \leq RAT$  AND all cooling coil sensible load is compensated by the economizer. (In this configuration the air system dampers modulate so that outdoor airflow is between 100% outdoor air and the minimum flow necessary to replace the zone exhaust airflow.)
- Economizer = OFF AND Coil = OFF IF cooling coil sensible load = 0.

#### **1.5.5.2.3.2 Case AE245: Low Cooling, Dry Coil 1 with Return Air Comparative Enthalpy Economizer Outdoor Air Control**

This test case shall be modeled with identical zone and system inputs as Case AE205 except for the existence of a return air comparative enthalpy economizer as described below.

**Informative Note, Objective.** Test model treatment of a single zone air system with return air comparative enthalpy economizer with low sensible cooling load and warm, dry outdoor air.

**Informative Note, Method.** Use the Case AE205 model as defined in Section 1.5.5.2.2.3 and add the economizer as described here. Compare cooling coil loads with the QAS and with results for Case AE205.

**Informative Note:** For the given economizer control logic, at the given steady-state operating conditions some cooling loads are increased by varying amounts.

**1.5.5.2.3.2.1 Input Specification.** Case AE245 shall be identical to Case AE205 (see Section 1.5.5.2.2.3) except for the addition of a return air comparative enthalpy economizer outdoor air control as specified in Section 1.5.5.2.3.2.1.1.

**1.5.5.2.3.2.1.1 Return Air Comparative Enthalpy Economizer Outdoor Air Control.** When the economizer is ON, the outdoor air shall increase until either the cooling coil sensible load is eliminated or the outdoor air is 100% of the supply airflow; if the economizer control cannot satisfy the cooling coil sensible load, then the cooling coil shall operate, along with the economizer, to meet the remaining cooling coil sensible load. The air system dampers (see Figure 1-3) shall be adjusted using economizer control based on outdoor and return air enthalpy as specified below.

- Economizer = OFF IF  $OAE > RAE$  OR  $ODB \leq ODB_{\text{econo,min}}$ . In this configuration the outdoor air mass flow rate shall equal the combined zone exhaust air mass flows.
  - Coil = ON IF  $T_{\text{zone}} >$  zone thermostat set point; see Section 1.5.5.1.1.3.c.
- Economizer = ON IF  $OAE \leq RAE$  AND  $ODB > ODB_{\text{econo,min}}$ .
  - Coil = ON IF  $T_{\text{zone}} >$  zone thermostat set point; see Section 1.5.5.1.1.3.c. In this configuration the mixed air shall be 100% outdoor air.
  - Coil = OFF IF  $T_{\text{zone}} \leq$  zone thermostat set point; see Section 1.5.5.1.1.3.c. In this configuration the air system dampers shall modulate the flow of outdoor air to meet the zone thermostat set point, and the outdoor airflow shall be permitted to vary from 100% outdoor air to the minimum flow necessary to replace the zone exhaust air mass flow.

Where for:

- Economizer = ON, outdoor air shall be provided as needed up to 100% outdoor air, but not less than the minimum required outdoor airflow rate at any time.
- Economizer = OFF, outdoor air shall be provided at the minimum required outdoor airflow rate.
- Coil = ON, the cooling coil shall operate only as necessary to satisfy the cooling coil sensible load not compensated by the economizer.
- Coil = OFF, the cooling coil shall not operate.

And where:

- OAE: Outdoor air enthalpy
- $ODB_{\text{econo,min}}$ : the outdoor dry-bulb temperature where at the minimum required outdoor airflow rate  $T_{\text{zone}} =$  zone thermostat set point.
- RAE: Return air enthalpy at the return fan outlet.
- $T_{\text{zone}}$ : zone air dry-bulb temperature. As used above, it is the zone air temperature if the coil did not operate.

**Informative Note:** Because the cooling coil activates (cooling coil load occurs) when  $T_{\text{zone}}$  rises above the thermostat set point, an equivalent economizer control scheme is:

- Economizer = OFF AND Coil = ON IF cooling coil sensible load  $> 0$  and  $OAE > RAE$ .
- Economizer = ON AND Coil = ON IF cooling coil sensible load  $> 0$  AND  $OAE \leq RAE$  AND all cooling coil sensible load is NOT compensated by the economizer. (In this configuration the mixed air is 100% outdoor air.)



- Economizer = ON AND Coil = OFF IF cooling coil sensible load  $> 0$  AND  $OAE \leq RAE$  AND all cooling coil sensible load is compensated by the economizer. (In this configuration the air system dampers modulate so that outdoor air is between 100% outdoor air and the minimum flow necessary to replace the zone exhaust airflow.)
- Economizer = OFF AND Coil = OFF IF cooling coil sensible load = 0.

### 1.5.5.3 Constant Volume Terminal Reheat ( $CV_{reheat}$ ) System Cases (AE300 series)

The ability to model a constant volume terminal reheat ( $CV_{reheat}$ ) air system serving multiple zones shall be tested as described in this section. If the software being tested is capable of applying a variety of system models to address a  $CV_{reheat}$  system, the system model that is most similar to the system specified below shall be applied. The selected system model shall be capable of economizer function.

**Informative Note:** The user may test other possible modeling approaches (available system models) in this context, as appropriate to the software being tested.

**Informative Note:** The progression of these test cases follows the AE200 series (SZ system) tests. The  $CV_{reheat}$  system serves two zones.

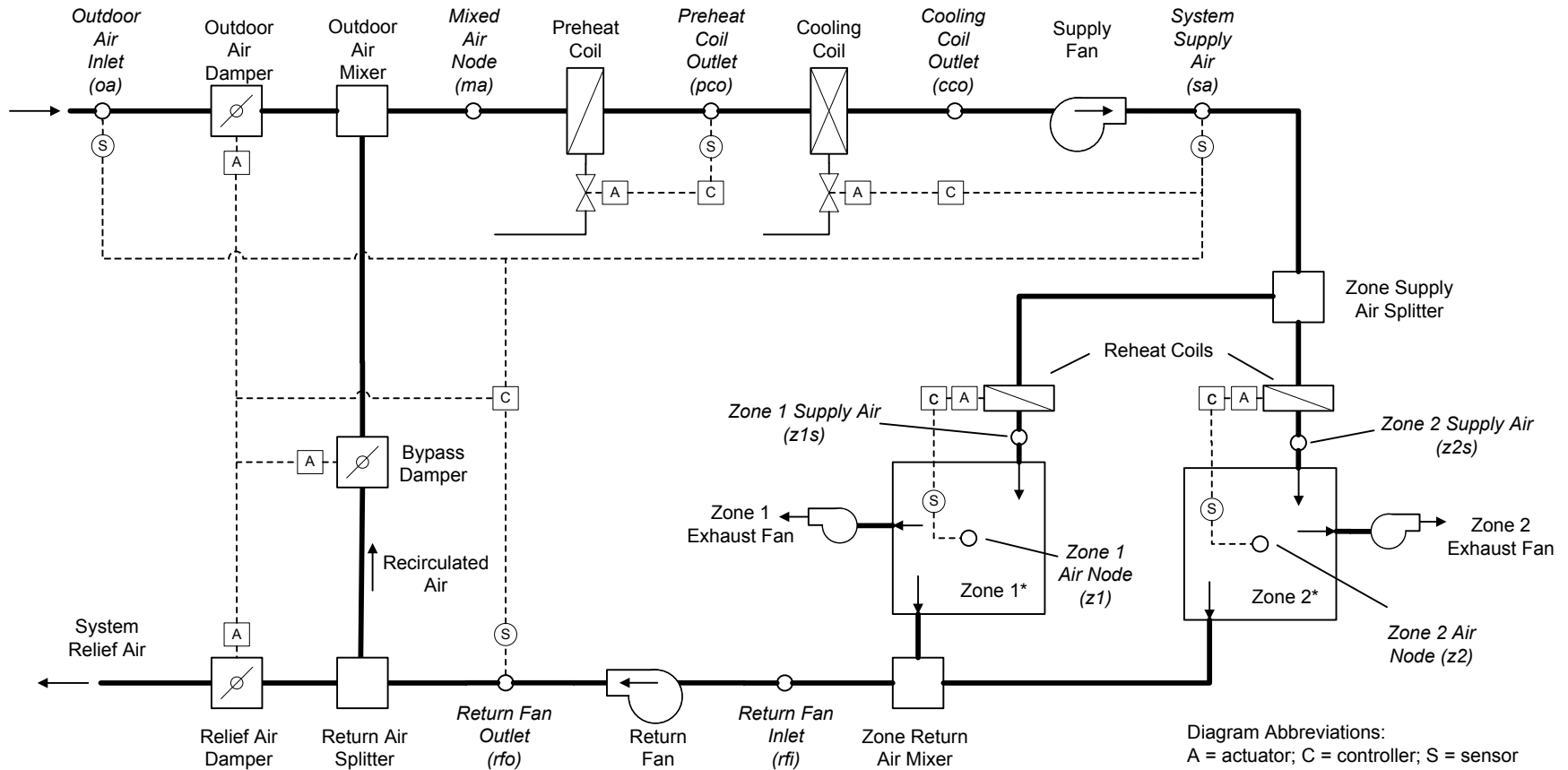
#### 1.5.5.3.1 Case AE301: Base Case, High Heating 1

Case AE301 shall be modeled as described in this section and its subsections. The system configuration shall be modeled as presented in the schematic diagram in Figure 1-4. System input parameters shall be as described in the following sections.

**Informative Note, Objective:** Test model treatment of a constant volume terminal-reheat system with high sensible heating load and cold outdoor air.

**Informative Note, Method:** A constant volume terminal reheat air system conditions two zones that have constant sensible and latent internal loads. The system consists of a constant volume air system with supply and return fans, pre-heat and cooling coils, and terminal reheat coils. The cooling coil provides cooling as needed to maintain the supply air temperature set point, and the reheat coils provide heating to maintain each zone temperature at its set point. The pre-heat coil operates as needed to prevent system supply air temperature from being too low. The model is run at specified constant outdoor and indoor conditions. Resulting coil loads are compared with the QAS and with other example results (see Part IV).

**Informative Note:** In this base case, no economizer function is modeled; the economizer function is tested in later cases.



\* Sensible and latent zone loads are specified for the base case in Section 1.5.5.3.1.4 and vary among the test cases. The zone thermostats sense only the zone air indoor dry-bulb temperature at their respective zone air nodes.

**Figure 1-4. Constant Volume Terminal Reheat system schematic ( $CV_{reheat}$ )**

**Informative Note:** Valves indicated are for a typical hydronic system and are not explicitly required by the test specification. Coils can be of any type as long as they meet the operational requirements of the test specification.

### 1.5.5.3.1.1 Fan Operation.

The system has a supply fan, a return fan, and individual zone exhaust fans.

**1.5.5.3.1.1.1 Supply Fan.** The supply fan characteristics shall be:

- a. The fan is located downstream of the heating and cooling coils, as specified in Figure 1-4.
- b. The supply fan total pressure rise = 2.0 inches wg (498 Pa).
- c. The fan provides a constant volume of supply air, measured at the fan inlet, as specified in Section 1.5.5.3.1.2.
- d. The fan operates continuously.
- e. Fan energy is transferred to the air that is being moved.
- f. Fan motor and transmission drive energy loss is not transferred to the moving air.
- g. (fan heat to airstream) =  $H_o \times (1 / (\text{fan mechanical efficiency}) - 1)$ , where
  - $H_o$  is the fan output power required to meet the specified air flow requirements; see  $H_o$  under *fan mechanical efficiency* in Section 1.5.3.1
  - Fan mechanical efficiency = 0.7, constant value; fan mechanical efficiency is defined in Section 1.5.3.1.
- h. (fan motor input power) =  $H_o / ((\text{fan mechanical efficiency}) \times (\text{transmission drive efficiency}) \times (\text{motor efficiency}))$ , where:
  - Transmission drive efficiency  $\leq 1.0$ ; transmission drive efficiency is defined in Section 1.5.3.1.
  - Motor efficiency  $\leq 1.0$ ; motor efficiency is defined in Section 1.5.3.1.

If the program being tested cannot model the transmission drive and motor losses as occurring outside the air stream, the fan, transmission drive, and motor efficiency losses shall be specified to have a combined efficiency of 0.7 and to occur in the air stream.

**Informative Note:** The air stream heat gain (fan heat) affects the test case outputs, while the motor and transmission drive input energy do not; therefore, the transmission and motor efficiency may be any value as long their heat is not imparted to the air stream, and the fan mechanical efficiency = 0.7.

**1.5.5.3.1.1.2 Return Fan.** The return fan characteristics shall be:

- a. The fan is located downstream of the zone return air mixer, and before the return air splitter, as specified in Figure 1-4.
- b. The return fan total pressure rise = 1.0 inches wg (249 Pa) at 800 cfm (377.56 L/s).
- c. The return fan total pressure rise varies as:  
Pressure rise [inches wg] = (1.0 inches wg)  $\times ((\text{actual return fan cfm}) / (800 \text{ cfm}))^2$
- d. The fan returns an air volume as specified in Section 1.5.5.3.1.2.
- e. The fan operates continuously.
- f. Fan energy is transferred to the air that is being moved.
- g. Fan motor and transmission drive energy loss is not transferred to the moving air.
- h. (fan heat to airstream) =  $H_o \times (1 / (\text{fan mechanical efficiency}) - 1)$ , where
  - $H_o$  is the fan output power required to meet the specified air flow requirements; see  $H_o$  under *fan mechanical efficiency* in Section 1.5.3.1
  - Fan mechanical efficiency = 0.7, constant value; fan mechanical efficiency is defined in Section 1.5.3.1.

- i. (fan motor input power) =  $H_o / ((\text{fan mechanical efficiency}) \times (\text{transmission drive efficiency}) \times (\text{motor efficiency}))$ , where:
- Transmission drive efficiency  $\leq 1.0$ ; transmission drive efficiency is defined in Section 1.5.3.1
  - Motor efficiency  $\leq 1.0$ ; motor efficiency is defined in Section 1.5.3.1.

If the program being tested cannot model the transmission drive and motor losses as occurring outside the air stream, the fan, transmission drive, and motor efficiency losses shall be specified to have a combined efficiency of 0.7 and to occur in the air stream.

**Informative Note:** The air stream heat gain (fan heat) affects the test case outputs, while the motor and transmission drive input energy do not; therefore, the transmission and motor efficiency may be any value as long their heat is not imparted to the air stream, and the fan mechanical efficiency = 0.7.

**1.5.5.3.1.1.3 Exhaust Fan.** The zone exhaust fan shall maintain airflow as specified in Section 1.5.5.3.1.2.

**Informative Note:** Exhaust fan details are not defined explicitly as the exhaust fan’s characteristics have no impact on the results of the test case.

**1.5.5.3.1.1.4 Programs with Pre-Determined Fan Modeling Assumptions.** Programs with pre-determined assumptions shall be permitted to apply those assumptions.

**Informative Note:** For example, a program may model the listed system supply airflow rate (see Section 1.5.5.3.1.2) using its default entering air conditions, if they are not normally adjustable by a typical program user.

**1.5.5.3.1.2 Airflows.**

System and zone airflows shall be as shown in Table 1-18 and the following subsections.

**Table 1-18. Case AE301 System and Zone Air Flows**

<b>System Airflows</b>	<b>SI Units</b>		<b>I-P Units</b>	
System Supply Airflow Rate	613.53 L/s <sup>a</sup>		1300 cfm <sup>a</sup>	
Design System Return Airflow Rate	377.56 L/s <sup>b</sup>		800 cfm <sup>b</sup>	
<b>Zone Airflows</b>	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
Nominal Zone Supply Airflow Rate	283.17 L/s <sup>c</sup>	330.36 L/s <sup>c</sup>	600 cfm <sup>c</sup>	700 cfm <sup>c</sup>
Zone Supply Air Mass Flow Fraction	0.4615 <sup>d</sup>	0.5385 <sup>c</sup>	0.4615 <sup>d</sup>	0.5385 <sup>c</sup>
Zone Exhaust Airflow Rate	94.39 L/s <sup>f</sup>	141.58 L/s <sup>f</sup>	200 cfm <sup>f</sup>	300 cfm <sup>f</sup>

<sup>a</sup> Volumetric airflow rate at the supply fan inlet conditions; see Section 1.5.5.3.1.2.1.

<sup>b</sup> This is the volumetric return airflow rate calculated from the *system supply airflow rate* minus the total *zone exhaust airflow rates*. It is the volumetric airflow rate upon which the return fan pressure rise (1 inch wg at 800 cfm) is based; see Section 1.5.5.3.1.1.2, items b and c. The actual return airflow rate shall be permitted to vary from the design airflow rate and is defined in Section 1.5.5.3.1.2.4.

<sup>c</sup> The nominal zone supply airflow rate is the volume of air delivered to each zone using air conditions at the supply fan inlet. Actual zone supply volumetric airflow rates shall be permitted to vary from the nominal volumetric airflow rates, and are defined by the zone supply air mass flow fractions and the corresponding local zone supply air properties.

<sup>d</sup> **Informative Note:** Calculated from 600 cfm / 1300 cfm; also see informative note accompanying Section 1.5.5.3.1.2.2.

<sup>e</sup> **Informative Note:** Calculated from 700 cfm / 1300 cfm; also see informative note accompanying Section 1.5.5.3.1.2.2.

<sup>f</sup> Volumetric airflow rate at exhaust fan inlet conditions; see Section 1.5.5.3.1.2.3.

**1.5.5.3.1.2.1 System Supply Air.** The system supply airflow rate shall be volumetrically constant and is measured at the supply fan inlet.

**Informative Note:** As the temperature and humidity ratio of the air entering the supply fan change, the specific volume of that air changes. This means that the mass flow rate of system supply air, while constant for a given steady-state test case, will vary among the test cases based upon fan inlet conditions. The combined mass flow rate of the zone air supplies is equal to that of the system supply air.

**Informative Note:** The QAS calculates the system mass flow rate from the volumetric flow using the local specific volume of air entering the supply fan. Results differences versus the QAS can be caused by differences in the method and assumptions a tested program uses to convert volumetric flow to mass flow. Example results for the QAS, including detailed outputs, e.g., mass flow rate, specific volume, enthalpy, etc., at specific system locations, are provided in Part IV; assumptions of the QAS for converting volumetric flows to mass flows are provided in Part II.

**1.5.5.3.1.2.2 Zone Supply Air.** The zone supply air mass flow rates are defined by the nominal zone supply volumetric flow rates and the supply air specific volume at the supply fan inlet conditions.

**Informative Note:** The actual zone supply air volumetric flow will vary as supply fan heat and terminal reheat increase the specific volume of the supply air away from the supply fan inlet conditions.

**Informative Note:** Listed zone supply air mass flow fractions (see Table 1-18) are the fraction of system supply air mass flow distributed to each zone. While the resulting zone volumetric airflow rates may vary from nominal values as noted above, the mass flow fractions remain constant.

**1.5.5.3.1.2.3 Zone Exhaust Air.** The zone exhaust airflow rates shall be volumetrically constant and are measured at the exhaust fan inlet conditions.

**Informative Note:** The QAS (see Part II) calculates the exhaust air mass flow rate from the volumetric flow using the local specific volume of the air entering the exhaust fan (i.e., the zone air properties).

**Informative Note:** As the temperature and humidity ratio of the air entering the exhaust fan change, the specific volume of that air changes. This means that the mass flow rate of exhaust air, while constant for a given steady-state test case, will vary among the test cases based upon zone conditions.

**1.5.5.3.1.2.4 Return Air.** The system return airflow rate shall be the air volume, as measured at the return fan inlet, necessary to move an air mass equal to the zone supply mass flows minus the zone exhaust mass flows.

**Informative Note:** The return fan volumetric flow rate in the QAS (see Part II) is calculated from the return air mass flow using the specific volume of air entering the return fan. The return fan volumetric flow varies with the return air mass flow and the specific volume of the air entering the return fan.

**1.5.5.3.1.2.5 Outdoor Air.** The flow of outdoor air shall be introduced at a mass flow rate equal to the sum of the zone exhaust air mass flow rates. For programs that do not precisely apply the specified mass flow balance, introduction of outdoor air to replace the specified exhaust airflow (see Table 1-18), as typically applied by the program, shall be permitted. *Informative Note:* Greater amounts of outdoor air may be required in other test cases applying economizer control logic.

**1.5.5.3.1.2.6 Frictionless Ducts, Coils, and Dampers.** Airflow through ducts, coils, and dampers shall be frictionless, such that there shall be no pressure drops through them. If the software being tested does not allow frictionless components, model the least amount of friction in these components that the software being tested allows.

*Informative Note:* Modeling of fan heat is as described previously, in Section 1.5.5.3.1.1.

**1.5.5.3.1.2.7 HVAC System Component Air Leakage and Heat Loss.** HVAC system components—including ducts, mixing boxes, dampers, fans, and coils—shall have no air leakage, and shall have no heat exchange (gains or losses) with their external surroundings. If the software being tested does not allow zero system air leakage or zero external heat gains or losses for HVAC system components, model the least amount of air leakage and external heat exchange that the software being tested allows.

*Informative Note:* Modeling of exhaust and outdoor airflows is as described previously in Sections 1.5.5.3.1.2.3 and 1.5.5.3.1.2.5. Modeling of heating and cooling coils is as described below in Section 1.5.5.3.1.3.

**1.5.5.3.1.3 Operation of Pre-heating, Cooling and Reheat Coils.** The pre-heating coil, the cooling coil, and the reheat coils shall be modeled with the following characteristics:

- a. All coils maintain set points precisely, without a throttling range. *Informative Note:* If the program being tested requires an input for throttling range, apply its minimum allowed value and adjust the coil set point so that supply air is delivered at the specified temperature. This may require adjustment for each test case.
- b. All coils have adequate capacity to meet the coil loads.
- c. Preheat coil set point = 45°F [7.22°C].  
The preheat coil activates when the temperature of the mixed air (passing through the “Preheat Coil Outlet” node of Figure 1-4) is below 45°F [7.22°C]. When activated, the preheat coil modulates to maintain the coil leaving air temperature at 45°F [7.22°C].
- d. Cooling coil set point = 55°F [12.78°C].  
The cooling coil activates when the temperature of the air leaving the supply fan (passing through the “System Supply Air” node of Figure 1-4) is above 55°F [12.78°C]. When activated, the cooling coil modulates to maintain the temperature of the air leaving the supply fan at 55°F [12.78°C].
- e. Zone reheat coils modulate to meet the zone sensible loads, delivering air at the temperature (above the system supply air temperature) needed to maintain the zone thermostat set point.
- f. Cooling coil bypass factor (BF) = 0.  
For building energy analysis programs not capable of directly modeling a cooling coil with a bypass factor of zero, a model shall be developed that applies the lowest bypass factor that the software being tested allows. Programs that do not have a bypass factor input shall apply a coil model that maximizes the cooling coil leaving air relative humidity when cooling coil latent load is present. For such programs, to approximate BF = 0 for each steady-state test case, variation of coil parameters among the test cases shall be allowed.

**Informative Note:** BF = 0 means that when the cooling coil leaving air temperature is less than the dew-point temperature of the air entering the cooling coil, the air will leave the coil at 100% relative humidity. If the cooling coil leaving air temperature is greater than the entering air dew-point temperature, no condensation will occur. When cooling coil latent load is present the cooling coil leaving air should be 100% saturated. This is equivalent to having an apparatus dew point equal to the required coil leaving air temperature (55°F [12.78°C]). To achieve saturation, programs with physical coil models should try to maximize the size of the coil and modulate the coil temperature or flow so the apparatus dew point is as close as possible to the required leaving air temperature while still delivering the proper amount of cooling. The QAS assumes that any condensate that forms is cooled from the entering air dew-point temperature to the coil leaving air temperature (see Part II, Section 2.2.1.26.1). Other assumptions regarding leaving condensate temperature may be reasonable.

**Informative Note:** For programs that do not accept a bypass factor input or do not accept BF near 0, modeling with an enlarged hydronic coil, with water temperature reset to meet the required supply air temperature, may be better than modeling with a direct expansion system coil unless the direct expansion coil model is capable of modulating continuously from 0 to 100% output.

**1.5.5.3.1.4 Zone Definition.** For programs that cannot specify zone loads directly and cannot define an adiabatic zone with negative internal gains, skip the remainder of this section and model the zones by applying the alternative non-adiabatic zone specified in Section 1.5.5.3.1.5.

Two unattached zones shall be modeled. Each zone shall conform to the specification in Section 1.5.5.1.1.4, except the ideal steady-state zone sensible heating and latent loads, as defined in Section 1.5.3.1, shall be modeled as specified in Table 1-19. Zone latent loads are applied at the given zone air temperature.

**Table 1-19. Case AE301 Zone Loads**

Input Parameter	SI Units		I-P Units	
	Zone 1	Zone 2	Zone 1	Zone 2
Zone Sensible Heating Load <sup>a</sup>	2931 W	2345 W	10000 Btu/hr	8000 Btu/hr
Zone Latent Load <sup>a,b</sup>	586.1 W	879.2 W	2000 Btu/hr	3000 Btu/hr

<sup>a</sup> Zone sensible heating load and zone latent load are defined in Section 1.5.3.1

<sup>b</sup> Zone latent loads are applied at the respective zone air temperatures specified in Section 1.5.5.3.1.6.

**1.5.5.3.1.5 Alternate Zone Specifications.** If the program being tested was able to model this test case as specified in Section 1.5.5.3.1.4, skip this section and proceed to Section 1.5.5.3.1.6. For programs that require non-adiabatic zones or cannot model negative internal gains, apply the alternate zone specification of Section 1.5.5.1.1.5 for each of the two unattached zones, except that the supplementary internal gains of Table 1-20 shall be applied. When implementing this zone definition, zone loads in the model output shall be examined and input adjustments shall be permitted, such that modeled zone loads (internal gains plus zone envelope heat transfer) match those specified in Section 1.5.5.3.1.4.

**Table 1-20. Case AE301 Supplementary Internal Gains for Alternate Load Specification**

Input Parameter	SI Units		I-P Units	
	Zone 1 (W)	Zone 2 (W)	Zone 1 (Btu/hr)	Zone 2 (Btu/hr)
Supplementary Internal Gains				
Sensible	82.9	735.8	282.8	2510.8
Latent <sup>a</sup>	586.1	879.2	2000.0	3000.0

<sup>a</sup> Supplementary latent internal gains are applied at the respective zone air temperatures specified in Section 1.5.5.3.1.6.

**1.5.5.3.1.6 Zone Temperatures and Thermostat Set Points.** The zone temperature shall be modeled as specified in Table 1-21.

**Table 1-21. Case AE301 Zone Air Temperature**

Input Parameter	SI Units		I-P Units	
	Zone 1	Zone 2	Zone 1	Zone 2
Zone Dry-Bulb Temperature	21.111°C	22.222°C	70.0°F	72.0°F

The thermostat specifications of Sections 1.5.5.1.1.6.1 through 1.5.5.1.1.6.4 shall apply.

**1.5.5.3.1.7 Ambient Conditions.** The ambient conditions specified in Table 1-22 shall be used.

**Table 1-22. Case AE301 Ambient Conditions**

	SI Units	I-P Units
Dry-Bulb Temperature	-29.0°C	-20.20°F
Dew-Point Temperature <sup>a</sup>	-29.0°C	-20.20°F
Solar Radiation	None	
Wind Speed	4.3 m/s	9.62 mph
Atmospheric Pressure <sup>b</sup>	101.325 kPa	14.696 psia
Elevation (Sea Level) <sup>b</sup>	0 m	0 ft
Equivalent Ambient Moisture Indicators <sup>a</sup>		
Humidity Ratio	0.000260	0.000260
Wet-Bulb Temperature	-29.000°C	-20.200°F
Relative Humidity	100.000%	100.000%
Relative Humidity (TM2) <sup>c</sup>	100%	100%

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file indicates 0 m altitude, consistent with the given pressure. However, the weather file atmospheric pressure precision is limited to four digits by the weather file format such that 1013 millibars (101300 Pa) is listed in the weather file. The effect of modeling with 101325 Pa versus 101300 Pa is negligible (< 0.05% on cooling coil latent loads) as described in Appendix A, Table A-3.

<sup>c</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.



If the tested program does not allow constant steady-state ambient conditions to be input directly, the TMY2-format weather data provided with the following file shall be applied:

**AE101.TM2**

These data are provided with the electronic files accompanying this report and are described in normative Appendix A.

**Informative Note:** The instructions for Section 1.5.5.3.1.7 are the same as those in Section 1.5.5.1.1.7 (for Case AE101).

**Informative Note:** Other cases call for different weather files as needed.

**Informative Note:** Some programs allow constant steady-state ambient conditions to be input directly for design-day calculations.

**1.5.5.3.1.8 Output Requirements.** The required outputs for this case shall be applied as specified in Section 1.6.5.3.

**1.5.5.3.2 Constant Volume Terminal Reheat System Parametric Variations, No Economizer**

**1.5.5.3.2.1 Case AE303: Low Cooling, Dry Coil 1**

**Informative Note, Objective.** Test model treatment of a constant volume terminal reheat system with low sensible cooling load and cool, dry outdoor air.

**Informative Note, Method.** Use Case AE301 model with modifications. Dry outdoor air and low sensible and latent loads used in the case ensure sensible only cooling and isolate that portion of the cooling calculation. Compare coil loads with the QAS and with other appropriate test case results.

**1.5.5.3.2.1.1 Input Specification.** This case shall be identical to Case AE301 (See Section 1.5.5.3.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-23; only values that change are shown in the table.

**Table 1-23. Case AE303 Input Parameters**

<b>Ambient Input Parameters</b>	<b>SI Units</b>		<b>I-P Units</b>	
Dry-Bulb Temperature	15.5°C		59.90°F	
Dew-Point Temperature <sup>a</sup>	-3.0°C		26.60°F	
Equivalent Ambient Moisture Indicators <sup>a</sup>				
Humidity Ratio	0.002948		0.002948	
Wet-Bulb Temperature	7.206°C		44.971°F	
Relative Humidity	27.028 %		27.028 %	
Relative Humidity (TM2) <sup>b</sup>	27 %		27 %	
<b>Zone Input Parameters</b>	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
Zone Sensible Cooling Load <sup>c</sup>	1465 W	2345 W	5000 Btu/hr	8000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.333°C	24.444°C	74°F	76°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485

(Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent loads as defined in Section 1.5.3.1 are applied at the given zone air temperature.

- b. **Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

### AE103.TM2

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. **Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.3.1.5, shall apply changes to parameters as indicated in Table 1-24; only values that change are shown in the table. Check output and adjust modeled internal gains so that for each zone: total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-23, and latent cooling load (latent internal gains) is equivalent to that specified in Table 1-20 (Case AE301).

**Table 1-24. Case AE303 Alternate Zone Definition Input Parameters**

Input Parameter	SI Units		I-P Units	
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W		567.8 (hr·ft <sup>2</sup> ·°F)/Btu	
Supplementary	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
Internal Gains – Sensible	1478.8 W	2359.9 W	5045.8 Btu/hr	8052.3 Btu/hr

**Informative Note:** For cooling cases, a near-adiabatic zone is applied, through use of insulation with high thermal resistance, to facilitate precise system loading. Use of a near-adiabatic zone was not possible for the heating cases because some programs may not allow direct input of negative internal gains.

#### 1.5.5.3.2.2 Case AE304: High Cooling, Wet Coil 1

**Informative Note, Objective.** Test model treatment of a constant volume terminal reheat system with high sensible cooling load and warm, humid outdoor air.

**Informative Note, Method.** Use Case AE301 model with modifications. Humid ambient conditions and high sensible cooling load ensure a cold cooling coil with high latent loads. Check cooling coil leaving air saturation to verify coil bypass factor is zero or near zero. Compare coil loads with the QAS and with other appropriate test case results.

**1.5.5.3.2.2.1 Input Specification.** This case shall be identical to Case AE301 (See Section 1.5.5.3.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-25; only values that change are shown in the table.

**Table 1-25. Case AE304 Input Parameters**

<b>Ambient Input Parameters</b>	<b>SI Units</b>		<b>I-P Units</b>	
Dry-Bulb Temperature	26.9°C		80.42°F	
Dew-Point Temperature <sup>a</sup>	22.1°C		71.78°F	
Equivalent Ambient Moisture Indicators <sup>a</sup>				
Humidity Ratio	0.016849		0.016849	
Wet-Bulb Temperature	23.441°C		74.194°F	
Relative Humidity	75.023 %		75.023 %	
Relative Humidity (TM2) <sup>b</sup>	75 %		75 %	
<b>Zone Input Parameters</b>	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
Zone Sensible Cooling Load <sup>c</sup>	2931 W	3517 W	10000 Btu/hr	12000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.889°C	25.000°C	75°F	77°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format, and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent loads as defined in Section 1.5.3.1 are applied at the given zone air temperature.

- b. Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

**AE104.TM2**

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.3.1.5, shall apply changes to parameters as indicated in Table 1-26; only values that change are shown in the table. Check output and adjust modeled internal gains so that for each zone: total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-25, and latent cooling load (latent internal gains) is equivalent to that specified in Table 1-20 (Case AE301).

**Table 1-26. Case AE304 Alternate Zone Definition Input Parameters**

<b>Input Parameter</b>	<b>SI Units</b>		<b>I-P Units</b>	
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W		567.8 (hr·ft <sup>2</sup> ·°F)/Btu	
Supplementary Internal Gains – Sensible	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
	2925.6 W	3513.6 W	9982.4 Btu/hr	11988.9 Btu/hr

**1.5.5.3.2.3 Case AE305: Low Cooling, Dry Coil 1**

**Informative Note, Objective.** Generate results for a constant volume terminal reheat system without enthalpy economizer, applying low sensible cooling load and warm, dry outdoor air, to compare with results for the same system with enthalpy economizer (Case AE345).

**Informative Note, Method.** Use the Case AE301 model with modifications. Case AE305 applies ambient conditions with low wet-bulb temperature to ensure that when the economizer is specified in Case AE345 it will operate. Compare coil loads with the QAS and with results for Case AE345 (which includes an enthalpy economizer system).

**1.5.5.3.2.3.1 Input Specification.** This case shall be identical to Case AE301 (see Section 1.5.5.3.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-27; only values that change are shown in the table.

**Table 1-27. Case AE305 Input Parameters**

<b>Ambient Input Parameters</b>	<b>SI Units</b>		<b>I-P Units</b>	
Dry-Bulb Temperature	24.9°C		76.82°F	
Dew-Point Temperature <sup>a</sup>	2.4°C		36.32°F	
Equivalent Ambient Moisture Indicators <sup>a</sup>				
Humidity Ratio	0.004510		0.004510	
Wet-Bulb Temperature	13.027°C		55.449°F	
Relative Humidity	23.050 %		23.050 %	
Relative Humidity (TM2) <sup>b</sup>	23 %		23 %	
<b>Zone Input Parameters</b>	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
Zone Sensible Cooling Load <sup>c</sup>	1465 W	2345 W	5000 Btu/hr	8000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.333°C	24.444°C	74°F	76°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent loads as defined in Section 1.5.3.1 are applied at the given zone air temperature.

- b. **Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

**AE105.TM2**

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. **Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.3.1.5, shall apply changes to parameters as indicated in Table 1-28; only values that change are shown in the table. Check output and adjust modeled internal gains so for each zone: total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-27, and latent cooling load (latent internal gains) is equivalent to that specified in Table 1-20 (Case AE301).

**Table 1-28. Case AE305 Alternate Zone Definition Input Parameters**

Input Parameter	SI Units		I-P Units	
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W		567.8 (hr·ft <sup>2</sup> ·°F)/Btu	
Supplementary Internal Gains – Sensible	Zone 1	Zone 2	Zone 1	Zone 2
	1462.7 W	2343.8 W	4990.8 Btu/hr	7997.3 Btu/hr

**1.5.5.3.2.4 Case AE306: Low Cooling, Wet Coil 1**

**Informative Note, Objective.** Generate results for a constant volume terminal reheat system without dry-bulb economizer, applying low sensible cooling load at warm, humid outdoor air conditions, to compare with results for the same system with dry-bulb economizer (Case AE326).

**Informative Note, Method.** Use the Case AE301 model with modifications. Case AE306 applies an ambient dry-bulb temperature that ensures the economizer specified in Case AE326 will operate. Humid ambient conditions and high sensible cooling ensure a cold cooling coil with high latent loads. Check the cooling coil leaving air saturation to verify coil bypass factor is zero or near zero. Compare coil loads with the QAS and with results for Case AE326 (which includes a comparative dry-bulb economizer system).

**1.5.5.3.2.4.1 Input Specification.** This case shall be identical to Case AE301 (see Section 1.5.5.3.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-29; only values that change are shown in the table.

**Table 1-29. Case AE306 Input Parameters**

Ambient Input Parameters	SI Units		I-P Units	
Dry-Bulb Temperature	23.0°C		73.40°F	
Dew-Point Temperature <sup>a</sup>	20.9°C		69.62°F	
Equivalent Ambient Moisture Indicators <sup>a</sup>				
Humidity Ratio	0.015625		0.015625	
Wet-Bulb Temperature	21.523°C		70.741°F	
Relative Humidity	87.968 %		87.968 %	
Relative Humidity (TM2) <sup>b</sup>	88 %		88 %	
Zone Input Parameters	Zone 1	Zone 2	Zone 1	Zone 2
Zone Sensible Cooling Load <sup>c</sup>	1465 W	2345 W	5000 Btu/hr	8000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.333°C	24.444°C	74°F	76°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent loads as defined in Section 1.5.3.1 are applied at the given zone air temperature.

- b. **Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

## AE106.TM2

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. **Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.3.1.5 shall apply changes to parameters as indicated in Table 1-30; only values that change are shown in the table. Check output and adjust modeled internal gains so that for each zone: total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-29, and latent cooling load (latent internal gains) is equivalent to that specified in Table 1-20 (Case AE301).

**Table 1-30. Case AE306 Alternate Zone Definition Input Parameters**

Input Parameter	SI Units		I-P Units	
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W		567.8 (hr·ft <sup>2</sup> ·°F)/Btu	
Supplementary Internal Gains -- Sensible	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
	1465.4 W	2347.0 W	5000.0 Btu/hr	8008.4 Btu/hr

### 1.5.5.3.3 Constant Volume Terminal Reheat System Parametric Variations, With Economizer

#### 1.5.5.3.3.1 Case AE326: Low Cooling, Wet Coil 1 with Return Air Comparative Dry-Bulb Economizer Outdoor Air Control

This test case shall be modeled with identical zone and system inputs as Case AE306 except for the existence of a return air comparative dry-bulb temperature economizer as described below.

**Informative Note, Objective.** Test model treatment of constant volume terminal reheat system with return air comparative dry-bulb temperature economizer with low sensible cooling load and warm, humid outdoor air.

**Informative Note, Method.** Use the Case AE306 model as defined in Section 1.5.5.3.2.4 and add the economizer as described here. Compare coil loads with the QAS and with results for Case AE306.

**Informative Note:** For the given economizer control logic, at the given steady-state operating conditions some cooling loads are increased by varying amounts.

**1.5.5.3.3.1.1 Input Specification.** Case AE326 shall be identical to Case AE306 (see Section 1.5.5.3.2.4) except for the addition of a return air comparative dry-bulb temperature economizer outdoor air control, as specified in Section 1.5.5.3.3.1.1.1.

**1.5.5.3.3.1.1.1 Return Air Comparative Dry-Bulb Economizer Outdoor Air Control.** When the economizer is ON, the outdoor air shall increase until either the cooling coil sensible load is eliminated, or the outdoor air is 100% of the supply airflow; if the economizer control cannot satisfy the cooling coil sensible load then the cooling coil shall operate, along with the economizer, to meet the remaining cooling coil sensible load. The air system dampers (see Figure 1-4) shall be adjusted using economizer control based on outdoor and return air dry-bulb temperature as specified below.

- Economizer = OFF IF  $ODB > RAT$  OR  $ODB \leq ODB_{\text{econo,min}}$ . In this configuration the outdoor air mass flow rate shall equal the combined zone exhaust air mass flows.

- Coil = ON IF  $SAT > 55^{\circ}F$  [ $12.78^{\circ}C$ ]; see Section 1.5.5.3.1.3.d.
- Economizer = ON IF  $RAT \geq ODB > ODB_{\text{econo,min.}}$ .
  - Coil = ON IF  $SAT > 55^{\circ}F$  [ $12.78^{\circ}C$ ]; see Section 1.5.5.3.1.3.d. In this configuration the mixed air shall be 100% outdoor air. **Informative Note:** For the test cases, this condition occurs when  $RAT \geq ODB > SFEAT$ , where SFEAT is the supply fan *entering* air temperature required to meet the system supply fan leaving air temperature set point.
  - Coil = OFF IF  $SAT \leq 55^{\circ}F$  [ $12.78^{\circ}C$ ]; see Section 1.5.5.3.1.3.d. In this configuration the air system dampers shall modulate the flow of outdoor air to meet the SAT set point, and the outdoor airflow shall be permitted to vary from 100% outdoor air to the minimum flow necessary to replace the combined zone exhaust air mass flows. **Informative Note:** For the test cases, this condition occurs when  $ODB \leq SFEAT$ .

Where for:

- Economizer = ON, outdoor air shall be provided as needed up to 100% outdoor air, but not less than the minimum required outdoor airflow rate at any time.
- Economizer = OFF, outdoor air shall be provided at the minimum required outdoor airflow rate.
- Coil = ON, the cooling coil shall operate only as necessary to satisfy the cooling coil sensible load not compensated by the economizer.
- Coil = OFF, the cooling coil shall not operate.

And where:

- ODB: outdoor air dry-bulb temperature
- $ODB_{\text{econo,min.}}$ : the outdoor dry-bulb temperature where at the minimum required outdoor airflow rate mixed air temperature (i.e., temperature of air passing through the mixed air node in Figure 1-4) = SAT set point minus the supply fan temperature rise.
- RAT: return air dry-bulb temperature, at the return fan outlet
- SAT: supply air dry-bulb temperature, at the supply air node (sa), see Figure 1-4. As used above, it is the supply air temperature if the coil did not operate.

**Informative Note:** Because the cooling coil activates (cooling coil load occurs) when supply fan leaving air temperature rises above  $55^{\circ}F$ , an equivalent economizer control scheme is:

- Economizer = OFF AND Coil = ON IF cooling coil sensible load  $> 0$  and  $ODB > RAT$ . In this configuration the outdoor airflow rate equals the combined zone exhaust airflows.
- Economizer = ON AND Coil = ON IF cooling coil sensible load  $> 0$  AND  $ODB \leq RAT$  AND all cooling coil sensible load is NOT compensated by the economizer. (In this configuration mixed air is 100% outdoor air.)
- Economizer = ON AND Coil = OFF IF cooling coil sensible load  $> 0$  AND  $ODB \leq RAT$  AND all cooling coil sensible load is compensated by the economizer. (In this configuration the air system dampers modulate so that outdoor airflow is between 100% outdoor air and the minimum flow necessary to replace the combined zone exhaust airflows.)

- Economizer = OFF AND Coil = OFF IF cooling coil sensible load = 0.

### 1.5.5.3.3.2 Case AE345: Low Cooling, Dry Coil 1 with Return Air Comparative Enthalpy Economizer Outdoor Air Control

This test case shall be modeled with identical zone and system inputs as Case AE305 except for the existence of a return air comparative enthalpy economizer as described below.

**Informative Note, Objective.** Test model treatment of a constant volume terminal reheat system with return air comparative enthalpy economizer at low sensible cooling load and warm, dry outdoor air.

**Informative Note, Method.** Use the Case AE305 model as defined in Section 1.5.5.3.2.3 and add the economizer as described here. Compare coil loads with the QAS and with results for Case AE305.

**Informative Note:** For the given economizer control logic, at the given steady-state operating conditions some cooling loads are increased by varying amounts.

**1.5.5.3.3.2.1 Input Specification.** Case AE345 shall be identical to Case AE305 (see Section 1.5.5.3.2.3) except for the addition of a return air comparative enthalpy economizer outdoor air control as specified in Section 1.5.5.3.3.2.1.1.

**1.5.5.3.3.2.1.1 Return Air Comparative Enthalpy Economizer Outdoor Air Control.** When the economizer is ON, the outdoor air shall increase until either the cooling coil sensible load is eliminated, or the outdoor air is 100% of the supply airflow; if the economizer control cannot satisfy the cooling coil sensible load then the cooling coil shall operate, along with the economizer, to meet the remaining cooling coil sensible load. The air system dampers (see Figure 1-4) shall be adjusted using economizer control based on outdoor and return air enthalpy as specified below.

- Economizer = OFF IF  $OAE > RAE$  OR  $ODB \leq ODB_{\text{econo,min}}$ . In this configuration the outdoor air mass flow rate shall equal the combined zone exhaust air mass flows.
  - Coil = ON IF  $SAT > 55^{\circ}\text{F}$  [ $12.78^{\circ}\text{C}$ ]; see Section 1.5.5.3.1.3.d.
- Economizer = ON IF  $OAE \leq RAE$  AND  $ODB > ODB_{\text{econo,min}}$ .
  - Coil = ON IF  $SAT > 55^{\circ}\text{F}$  [ $12.78^{\circ}\text{C}$ ]; see Section 1.5.5.3.1.3.d. In this configuration the mixed air shall be 100% outdoor air. **Informative Note:** For the test cases, this condition occurs when coincident  $ODB > SFEAT$ , where SFEAT is the supply fan entering air temperature required to meet the system supply fan leaving air temperature set point.
  - Coil = OFF IF  $SAT \leq 55^{\circ}\text{F}$  [ $12.78^{\circ}\text{C}$ ]; see Section 1.5.5.3.1.3.d. In this configuration the air system dampers shall modulate the flow of outdoor air to meet the SAT set point, and the outdoor airflow shall be permitted to vary from 100% outdoor air to the minimum flow necessary to replace the combined zone exhaust air mass flows. **Informative Note:** For the test cases this condition occurs when coincident  $ODB \leq SFEAT$ .

Where for:

- Economizer = ON, outdoor air shall be provided as needed up to 100% outdoor air, but not less than the minimum required outdoor airflow rate at any time.



- Economizer = OFF, outdoor air shall be provided at the minimum required outdoor airflow rate.
- Coil = ON, The cooling coil shall operate only as necessary to satisfy the cooling coil sensible load not compensated by the economizer.
- Coil = OFF, The cooling coil shall not operate.

And where:

- OAE: Outdoor air enthalpy
- $ODB_{\text{econo,min}}$ : the outdoor dry-bulb temperature where at the minimum required outdoor airflow rate mixed air temperature (i.e., temperature of air passing through the mixed air node in Figure 1-4) = SAT set point minus supply fan temperature rise.
- RAE: Return air enthalpy, at the return fan outlet.
- SAT: Supply air dry-bulb temperature, at the supply air node (sa), see Figure 1-4. As used above, it is the supply air temperature if the coil did not operate.

**Informative Note:** Because the cooling coil activates (cooling coil load occurs) when supply fan leaving air temperature rises above 55°F, an equivalent economizer control scheme is:

- Economizer = OFF AND Coil = ON IF cooling coil sensible load  $> 0$  and  $OAE > RAE$ .
- Economizer = ON AND Coil = ON IF cooling coil sensible load  $> 0$  AND  $OAE \leq RAE$  AND all cooling coil sensible load is NOT compensated by the economizer. (In this configuration mixed air is 100% outdoor air.)
- Economizer = ON AND Coil = OFF IF cooling coil sensible load  $> 0$  AND  $OAE \leq RAE$  AND all cooling coil sensible load is compensated by the economizer. (In this configuration the air system dampers modulate so that outdoor air is between 100% outdoor air and the minimum flow necessary to replace the combined zone exhaust airflows.)
- Economizer = OFF AND Coil = OFF IF cooling coil sensible load = 0.

#### 1.5.5.4 Variable Air Volume (VAV) System Cases (AE400 series).

The ability to model a variable air volume (VAV) air system with zone reheat serving multiple zones shall be tested as described in this section. If the software being tested is capable of applying a variety of system models to address a variable air volume system, the system model that is most similar to the system specified below shall be applied. The selected system model shall be capable of economizer function.

**Informative Note:** The user may test other possible modeling approaches (available system models) in this context, as appropriate to the software being tested.

**Informative Note:** The progression of these test cases follows the AE300 series ( $CV_{\text{reheat}}$  system) tests.

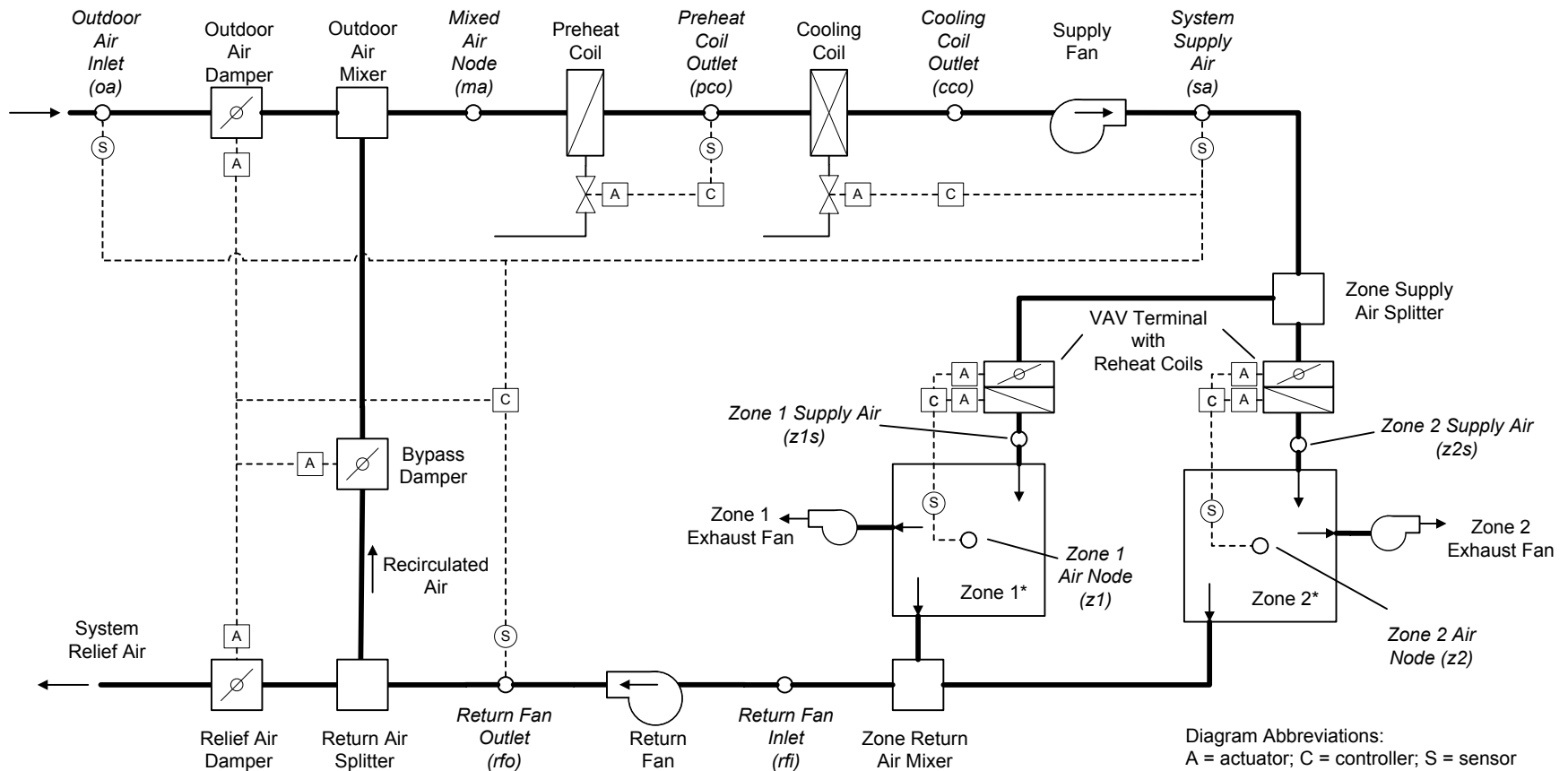
##### 1.5.5.4.1 Case AE401: Base Case, High Heating 1

Case AE401 shall be modeled as specified in this section and its subsections. The system configuration shall be modeled as presented in the schematic diagram of Figure 1-5. System input parameters shall be as described in the following sections. **Informative Note:** The test specifications for the heating, cooling, and reheat coils; zone definition; alternate zone specifications; zone temperature and thermostat set points; and ambient conditions (Sections 1.5.5.4.1.4 through 1.5.5.4.1.8) are the same as those for the  $CV_{\text{reheat}}$  system (Sections 1.5.5.3.1.3 through 1.5.5.3.1.7).

**Informative Note, Objective.** Test model treatment of a variable air volume terminal-reheat system with high sensible heating load and cold outdoor air.

**Informative Note, Method.** A VAV air system with variable air volume terminals and reheat coils conditions two zones that have constant sensible and latent internal loads. The system has variable speed supply and return fans, pre-heat and cooling coils, and zone variable air volume terminals with reheat coils. The cooling coil provides cooling as needed to maintain the supply air temperature set point, and the reheat coils provide heating to maintain each zone temperature at its set point. The pre-heat coils will operate as needed to prevent supply air temperature from going too far below the supply air temperature set point. Zone VAV terminals modulate to provide only the necessary airflow needed to maintain the zone temperature set point, but not below the zone exhaust airflow rate. The model is run at specified constant outdoor and indoor conditions. Resulting coil loads are compared with the QAS and with other example results (see Part IV).

**Informative Note:** In this base case, no economizer function is modeled; economizer function is tested in later cases.



\* Sensible and latent zone loads are specified for the base case in Section 1.5.5.4.1.5, and vary among the test cases. The zone thermostats sense only the zone air indoor dry-bulb temperature at their respective zone air nodes.

**Figure 1-5. Variable Air Volume Air System schematic (VAV)**

**Informative Note:** Valves indicated are for a typical hydronic system and are not explicitly required by the test specification. Coils can be of any type as long as they meet the operational requirements of the test specification.

**1.5.5.4.1.1 Fan Operation.** The system has a supply fan, return fan, and individual zone exhaust fans.

**1.5.5.4.1.1.1 Supply Fan.** The supply fan characteristics shall be:

- a. The fan is located downstream of the heating and cooling coils, as specified in Figure 1-5.
- b. The supply fan total pressure rise = 2.0 inches wg (498 Pa) at 1300 cfm (613.53 L/s).
- c. The supply fan total pressure rise varies as:  
$$\text{Pressure rise (inches wg)} = 2.0 \text{ inches wg} \times ((\text{actual supply fan cfm}) / (1300 \text{ cfm}))^2$$
- d. The fan modulates to provide the airflow established by the zone VAV terminals, as specified in Section 1.5.5.4.1.3.
- e. The fan operates continuously.
- f. Fan energy is transferred to the air that is being moved.
- g. Fan motor and transmission drive energy loss is not transferred to the moving air.
- h.  $(\text{fan heat to airstream}) = H_o \times (1 / (\text{fan mechanical efficiency}) - 1)$ , where
  - $H_o$  is the fan output power required to meet the specified air flow requirements; see  $H_o$  under *fan mechanical efficiency* in Section 1.5.3.1
  - Fan mechanical efficiency = 0.7, constant value; fan mechanical efficiency is defined in Section 1.5.3.1.
- i.  $(\text{fan motor input power}) = H_o / ((\text{fan mechanical efficiency}) \times (\text{transmission drive efficiency}) \times (\text{motor efficiency}))$ , where:
  - Transmission drive efficiency  $\leq 1.0$ ; transmission drive efficiency is defined in Section 1.5.3.1
  - Motor efficiency  $\leq 1.0$ ; motor efficiency is defined in Section 1.5.3.1.

If the program being tested cannot model the transmission drive and motor losses as occurring outside the air stream, the fan, transmission drive, and motor efficiency losses shall be specified to have a combined efficiency of 0.7 and to occur in the air stream.

**Informative Note:** The air stream heat gain (fan heat) affects the test case outputs, while the motor and transmission drive input energy do not; therefore, the transmission and motor efficiency may be any value as long their heat is not imparted to the air stream, and the fan mechanical efficiency = 0.7.

**1.5.5.4.1.1.2 Return Fan.** The return fan characteristics shall be:

- a. The fan is located downstream of the zone return air mixer, and before the return air splitter, as specified in Figure 1-5.
- b. The return fan total pressure rise = 1.0 inches wg (249 Pa) at 800 cfm (377.56 L/s).
- c. The return fan total pressure rise varies as:  
$$\text{Pressure rise (inches wg)} = 1.0 \text{ inches wg} \times ((\text{actual return fan cfm}) / (800 \text{ cfm}))^2$$
- d. The fan modulates to return an air volume as specified in Section 1.5.5.4.1.2.4.
- e. The fan operates continuously.
- f. Fan energy is transferred to the air that is being moved.
- g. Fan motor and transmission drive energy loss is not transferred to the moving air.
- h.  $(\text{fan heat to airstream}) = H_o \times (1 / (\text{fan mechanical efficiency}) - 1)$ , where
  - $H_o$  is the fan output power required to meet the specified air flow requirements; see  $H_o$  under *fan mechanical efficiency* in Section 1.5.3.1
  - Fan mechanical efficiency = 0.7, constant value; fan mechanical efficiency is defined in Section 1.5.3.1.

- i. (fan motor input power) =  $H_o / ((\text{fan mechanical efficiency}) \times (\text{transmission drive efficiency}) \times (\text{motor efficiency}))$ , where:
- Transmission drive efficiency  $\leq 1.0$ ; transmission drive efficiency is defined in Section 1.5.3.1
  - Motor efficiency  $\leq 1.0$ ; motor efficiency is defined in Section 1.5.3.1.

If the program being tested cannot model the transmission drive and motor losses as occurring outside the air stream, the fan, transmission drive, and motor efficiency losses shall be specified to have a combined efficiency of 0.7 and to occur in the air stream.

**Informative Note:** The air stream heat gain (fan heat) affects the test case outputs, while the motor and transmission drive input energy do not; therefore, the transmission and motor efficiency may be any value as long their heat is not imparted to the air stream, and the fan mechanical efficiency = 0.7.

**1.5.5.4.1.1.3 Exhaust Fan.** The zone exhaust fan shall maintain airflow as specified in section 1.5.5.4.1.2.

**Informative Note:** Exhaust fan details are not defined explicitly as the exhaust fan’s characteristics have no impact on the results of the test case.

**1.5.5.4.1.1.4 Programs with Pre-Determined Fan Modeling Assumptions.** Programs with pre-determined assumptions shall be permitted to apply those assumptions.

**Informative Note:** For example, a program may model the listed system supply airflow rate (see Section 1.5.5.4.1.2) using its default entering air conditions if they are not normally adjustable by a typical program user.

**1.5.5.4.1.2 Airflows.** System and zone airflows shall be as shown in Table 1-31 and the following subsections.

**Table 1-31 Case AE401 System Airflow Rates**

<b>System Airflows</b>	<b>SI Units</b>		<b>I-P Units</b>	
Design System Supply Airflow Rate	613.53 L/s <sup>a</sup>		1300 cfm <sup>a</sup>	
Design System Return Airflow Rate	377.56 L/s <sup>b</sup>		800 cfm <sup>b</sup>	
<b>Zone Airflows</b>	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
Design Nominal Zone Supply Airflow Rate	283.17 L/s <sup>c</sup>	330.36 L/s <sup>c</sup>	600 cfm <sup>c</sup>	700 cfm <sup>c</sup>
Design Zone Minimum Airflow Rate	94.39 L/s <sup>d</sup>	141.58 L/s <sup>d</sup>	200 cfm <sup>d</sup>	300 cfm <sup>d</sup>
Design Zone Minimum Airflow Fraction (% of Design Nominal Zone Supply Air)	33% <sup>d,e</sup>	43% <sup>d,f</sup>	33% <sup>d,e</sup>	43% <sup>d,f</sup>
Zone Exhaust Airflow Rate	94.39 L/s <sup>g</sup>	141.58 L/s <sup>g</sup>	200 cfm <sup>g</sup>	300 cfm <sup>g</sup>

<sup>a</sup> This is the volumetric airflow rate upon which the supply fan pressure rise (2 inches wg at 1300 cfm) is based; see Section 1.5.5.4.1.1.1, items b and c. The actual system supply airflow rate shall be permitted to vary from the design system supply airflow rate and is defined in Section 1.5.5.4.1.2.1.

<sup>b</sup> This is the volumetric return airflow rate calculated from the *design system supply airflow* rate minus the total *zone exhaust airflow* rates. It is the volumetric airflow rate upon which the return fan pressure rise (1 inch wg at 800 cfm) is based; see Section 1.5.5.4.1.1.2, items b and c. The actual return airflow rate shall be permitted to vary from the design airflow rate and is defined in Section 1.5.5.4.1.2.4.

<sup>c</sup> The *design nominal zone supply airflow rate* is the maximum volume of air entering the zone at the outlet of the VAV terminal, using air conditions at the supply fan inlet. Any design zone supply airflow rate shall be

permitted as long as zone sensible loads are satisfied. Actual zone supply volumetric airflow rates shall be permitted to vary according to zone VAV terminal requirements (see Section 1.5.5.4.1.3). **Informative Note:** At design conditions, the actual zone supply air volumetric flow varies as supply fan heat increases the specific volume of the air away from the supply fan inlet. Use of the values in this table will ensure adequate capacity for the specified test cases.

<sup>d</sup> **Informative Note:** Design zone minimum airflow rate and minimum airflow fraction are approximate equivalent inputs provided for those programs that may need them. The actual zone minimum airflow rate equals the zone exhaust air mass flow rate. Programs that require volumetric inputs for the zone minimum airflow and do not calculate volumetric airflows based upon system specific local conditions (e.g., zone exhaust at zone air conditions, supply air at supply fan entering air conditions) may use the listed values. Because zone conditions and supply fan entering air temperature vary among the test cases, the actual minimum zone air mass flow rates and volumetric flow rates, as calculated by the QAS, also vary among the test cases.

<sup>e</sup> **Informative Note:** Zone (VAV terminal) minimum airflow fraction; listed approximate value is calculated from 200 cfm / 600 cfm. A more precise value can be obtained from the ratio of the mass flows.

<sup>f</sup> **Informative Note:** Zone (VAV terminal) minimum airflow fraction; listed approximate value is calculated from 300 cfm / 700 cfm. A more precise value can be obtained from the ratio of the mass flows.

<sup>g</sup> Volumetric airflow rate at exhaust fan inlet conditions; see Section 1.5.5.4.1.2.3.

**Informative Note:** The design system supply and return, and design zone supply airflow rates are the same as the designated constant volume airflow rates of Case AE301. For the VAV system cases, the actual system supply and return airflow rates, and the zone supply airflow rate, vary in response to the operation of VAV terminals.

**1.5.5.4.1.2.1 System Supply Air.** The system supply airflow rate shall deliver the airflow required to satisfy the combined VAV terminal airflow requirements of zones 1 and 2; see Section 1.5.5.4.1.3 for VAV terminal operation requirements.

**Informative Note:** The system supply mass flow rate is equal to the combined mass flow rate of the zone air supplies.

**Informative Note:** As the temperature and humidity ratio of the air entering the supply fan change, the specific volume of that air changes.

**Informative Note:** The QAS calculates the system mass flow rate from the volumetric flow using the local specific volume of air entering the supply fan. Results differences versus the QAS can be caused by differences in the method and assumptions a tested program uses to convert volumetric flow to mass flow. Example results for the QAS, including detailed outputs, e.g., mass flow rate, specific volume, enthalpy, etc., at specific system locations, are provided in Part IV; assumptions of the QAS for converting volumetric flows to mass flows are provided in Part II.

**1.5.5.4.1.2.2 Zone Supply Air.** The zone supply airflow rates shall be those required to meet the VAV terminal airflow requirements, see Section 1.5.5.4.1.3, items b, c, d, and f.

**1.5.5.4.1.2.3 Zone Exhaust Air.** The zone exhaust airflow rates shall be volumetrically constant and are measured at the exhaust fan inlet conditions.

**Informative Note:** The QAS (see Part II) calculates the exhaust air mass flow rate from the volumetric flow using the local specific volume of the air entering the exhaust fan (i.e., the zone air properties).

**Informative Note:** As the temperature and humidity ratio of the air entering the exhaust fan change, the specific volume of that air changes. This means that the mass flow rate of exhaust air, while constant for a given steady-state test case, will vary among the test cases based upon zone conditions.

**1.5.5.4.1.2.4 Return Air.** The system return airflow rate shall be the air volume, as measured at the return fan inlet, necessary to move an air mass equal to the zone supply mass flows minus the zone exhaust mass flows.

*Informative Note:* When the VAV terminals are at minimum position, defined as that needed to offset zone exhaust (Section 1.5.5.4.1.3, item c), there will be zero return airflow.

*Informative Note:* The return fan volumetric flow rate in the QAS (see Part II) is calculated from the return air mass flow using the specific volume of air entering the return fan. The return fan volumetric flow varies with the return air mass flow and the specific volume of the air entering the return fan.

**1.5.5.4.1.2.5 Outdoor Air.** The flow of outdoor air shall be introduced at a mass flow rate equal to the sum of the zone exhaust air mass flow rates. For programs that do not precisely apply the specified mass flow balance, introduction of outdoor air to replace the specified exhaust airflow (see Table 1-31) as typically applied by the program shall be permitted. *Informative Note:* Greater amounts of outdoor air may be required in other test cases applying economizer control logic.

**1.5.5.4.1.2.6 Frictionless Ducts, Coils, and Dampers.** Airflow through ducts, coils, and dampers shall be frictionless, such that there shall be no pressure drops through them. If the software being tested does not allow frictionless components, model the least amount of friction in these components that the software being tested allows.

*Informative Note:* Modeling of fan heat is as described previously, in Section 1.5.5.4.1.1.

**1.5.5.4.1.2.7 HVAC System Component Air Leakage and Heat Loss.** HVAC system components—including ducts, mixing boxes, dampers, fans, and coils—shall have no air leakage, and shall have no heat exchange (gains or losses) with their external surroundings. If the software being tested does not allow zero system air leakage or zero external heat gains or losses for HVAC system components, model the least amount of air leakage and external heat exchange that the software being tested allows.

*Informative Note:* Modeling of exhaust and outdoor airflows is as described previously in Sections 1.5.5.4.1.2.3 and 1.5.5.4.1.2.5. Modeling of heating and cooling coils is as described below in Section 1.5.5.4.1.4.

**1.5.5.4.1.3 Operation of VAV Terminals.** Zone VAV terminals shall operate as follows:

- a. Zone VAV terminals are controlled by zone thermostats and maintain zone set points precisely without a throttling range or dead band.
- b. Zone VAV terminals shall have adequate airflow capacity and reheat coil heating capacity to meet the space conditioning requirements of the test case.
- c. The VAV terminal dampers shall reduce the zone supply airflows to the zone minimum required airflow rates, unless zone space cooling requires higher flows.
- d. The minimum required zone airflow rates are those required to replace the respective zone exhaust fan airflows. I.e., the VAV terminal minimum airflow rate shall equal the zone exhaust fan mass flow rate. *Informative Note:* Because the exhaust mass flow rates change with zone (exhaust fan inlet) conditions, the VAV terminal minimum mass flow rate will also change.
- e. If the minimum zone supply airflow causes a zone to drop below the zone temperature set point, the reheat coil shall modulate to maintain the zone temperature set point. At all other conditions, the reheat coil shall be off.

- f. If the minimum zone supply airflow is not adequate to keep a zone temperature from rising above set point, the VAV terminal damper shall open to increase the flow of supply air to the zone as required to maintain zone space cooling set points. **Informative Note:** If input values are needed for a maximum or design VAV terminal flow rate, the Design Nominal Zone Supply Airflow Rates provided in Table 1-31 may be used; at the design condition, the VAV terminal dampers are in the fully open position. The current test cases do not require operation at or above 100% design conditions.

**1.5.5.4.1.4 Operation of Pre-heating, Cooling and Reheat Coils.** The pre-heating coil, the cooling coil, and the reheat coils shall be modeled as specified for the CV<sub>reheat</sub> system test base case in Section 1.5.5.3.1.3.

**1.5.5.4.1.5 Zone Definition.** For programs that cannot specify zone loads directly and cannot define an adiabatic zone with negative internal gains, skip the remainder of this section and model the zone by applying the alternative non-adiabatic zone specified in Section 1.5.5.4.1.6.

Two unattached zones shall be modeled as specified for the CV<sub>reheat</sub> system test base case in Section 1.5.5.3.1.4.

**1.5.5.4.1.6 Alternate Zone Specifications.** If the program being tested was able to model this test case as specified in Section 1.5.5.4.1.5, skip this section and proceed to Section 1.5.5.4.1.7.

For programs that require non-adiabatic zones or cannot model negative internal gains, the alternate zone specification for the CV<sub>reheat</sub> system test base case in Section 1.5.5.3.1.5 shall be applied.

**1.5.5.4.1.7 Zone Temperatures and Thermostat Set Points.** These shall be the same as for the CV<sub>reheat</sub> system test base case as specified in Section 1.5.5.3.1.6.

**1.5.5.4.1.8 Ambient Conditions.** The ambient conditions shall be the same as for the CV<sub>reheat</sub> system test base case as specified in Section 1.5.5.3.1.7.

**1.5.5.4.1.9 Output Requirements.** The required outputs for this case shall be as specified in Section 1.6.5.4.

#### **1.5.5.4.2 VAV System Parametric Variations, No Economizer**

##### **1.5.5.4.2.1 Case AE403: Low Cooling, Dry Coil 1**

**Informative Note, Objective.** Test model treatment of a variable air volume terminal reheat system with low sensible cooling load and cool, dry outdoor air.

**Informative Note, Method.** Use the Case AE401 model with modifications. Dry outdoor air and low sensible and latent loads used in the case ensure sensible only cooling and isolate that portion of the cooling calculation. Compare coil loads with the QAS and with other appropriate test case results.

**1.5.5.4.2.1.1 Input Specification.** This case shall be identical to Case AE401 (see Section 1.5.5.4.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-32; only values that change are shown in the table.



**Table 1-32. Case AE403 Input Parameters**

<b>Ambient Input Parameters</b>	<b>SI Units</b>		<b>I-P Units</b>	
Dry-Bulb Temperature	15.5°C		59.90°F	
Dew-Point Temperature <sup>a</sup>	-3.0°C		26.60°F	
Equivalent Ambient Moisture Indicators <sup>a</sup>				
Humidity Ratio	0.002948		0.002948	
Wet-Bulb Temperature	7.206°C		44.971°F	
Relative Humidity	27.028 %		27.028 %	
Relative Humidity (TM2) <sup>b</sup>	27 %		27 %	
<b>Zone Input Parameters</b>	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
Zone Sensible Cooling Load <sup>c</sup>	1465 W	2345 W	5000 Btu/hr	8000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.333°C	24.444°C	74°F	76°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format, and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent loads as defined in Section 1.5.3.1 are applied at the given zone air temperature.

- b. Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

**AE103.TM2**

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.4.1.6, shall apply changes to parameters as indicated in Table 1-33; only values that change are shown in the table. Check output and adjust modeled internal gains so for each zone: total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-32, and latent cooling load (latent internal gains) is equivalent to that specified in Table 1-20 (Case AE301).

**Table 1-33. Case AE403 Alternate Zone Definition Input Parameters**

<b>Input Parameter</b>	<b>SI Units</b>		<b>I-P Units</b>	
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W		567.8 (hr·ft <sup>2</sup> ·°F)/Btu	
Supplementary Internal Gains – Sensible	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
	1478.8 W	2359.9 W	5045.8 Btu/hr	8052.3 Btu/hr

**Informative Note:** For cooling cases a near-adiabatic zone is applied, through use of insulation with high thermal resistance, to facilitate precise system loading. Use of a near-adiabatic zone was not possible for the heating cases because some programs may not allow direct input of negative internal gains.

### 1.5.5.4.2.2 Case AE404: High Cooling, Wet Coil 1

**Informative Note, Objective.** Test model treatment of a variable air volume terminal reheat system with high sensible cooling load and warm, humid outdoor air.

**Informative Note, Method.** Use Case AE401 model with modifications. Humid ambient conditions and high sensible cooling ensure a cold cooling coil with high latent loads. Check cooling coil leaving air saturation to verify coil bypass factor is zero or near zero. Compare coil loads with the QAS and with other appropriate test case results.

**1.5.5.4.2.2.1 Input Specification.** This case shall be identical to Case AE401 (See Section 1.5.5.4.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-34; only values that change are shown in the table.

**Table 1-34. Case AE404 Input Parameters**

<b>Ambient Input Parameters</b>	<b>SI Units</b>		<b>I-P Units</b>	
Dry-Bulb Temperature	26.9°C		80.42°F	
Dew-Point Temperature <sup>a</sup>	22.1°C		71.78°F	
Equivalent Ambient Moisture Indicators <sup>a</sup>				
Humidity Ratio	0.016849		0.016849	
Wet-Bulb Temperature	23.441°C		74.194°F	
Relative Humidity	75.023 %		75.023 %	
Relative Humidity (TM2) <sup>b</sup>	75 %		75 %	
<b>Zone Input Parameters</b>	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
Zone Sensible Cooling Load <sup>c</sup>	2931 W	3517 W	10000 Btu/hr	12000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.889°C	25.000°C	75°F	77°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format, and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent loads as defined in Section 1.5.3.1 are applied at the given zone air temperature.

- b. **Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

#### **AE104.TM2**

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. **Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.4.1.6, shall apply changes to parameters as indicated in Table 1-35; only values that change are shown in the table. Check output and adjust modeled internal gains so that for each zone: total zone sensible cooling load (envelope sensible heat gains plus all sensible internal

gains) is equivalent to that specified in Table 1-34, and latent cooling load (latent internal gains) is equivalent to that specified in Table 1-20 (Case AE301).

**Table 1-35. Case AE404 Alternate Zone Definition Input Parameters**

Input Parameter	SI Units		I-P Units	
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W		567.8 (hr·ft <sup>2</sup> ·°F)/Btu	
Supplementary Internal Gains – Sensible	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
	2925.6 W	3513.6 W	9982.4 Btu/hr	11988.9 Btu/hr

**1.5.5.4.2.3 Case AE405: Low Cooling, Dry Coil 1**

**Informative Note, Objective.** Generate results for a variable air volume terminal reheat system without enthalpy economizer, applying low sensible cooling load and warm, dry outdoor air, to compare with results for the same system with enthalpy economizer (Case AE445).

**Informative Note, Method.** Use Case AE401 model with modifications. Case AE405 applies ambient conditions with low wet-bulb temperature to ensure that when the economizer is specified in Case AE445 it will operate. Compare coil loads with the QAS and with results for Case AE445 (which includes an enthalpy economizer system).

**1.5.5.4.2.3.1 Input Specification.** This case shall be identical to Case AE401 (See Section 1.5.5.4.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-36; only values that change are shown in the table.

**Table 1-36. Case AE405 Input Parameters**

Ambient Input Parameters	SI Units		I-P Units	
Dry-Bulb Temperature	24.9°C		76.82°F	
Dew-Point Temperature <sup>a</sup>	2.4°C		36.32°F	
Equivalent Ambient Moisture Indicators <sup>a</sup>				
Humidity Ratio	0.004510		0.004510	
Wet-Bulb Temperature	13.027°C		55.449°F	
Relative Humidity	23.050 %		23.050 %	
Relative Humidity (TM2) <sup>b</sup>	23 %		23 %	
Zone Input Parameters	Zone 1	Zone 2	Zone 1	Zone 2
Zone Sensible Cooling Load <sup>c</sup>	1465 W	2345 W	5000 Btu/hr	8000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.333°C	24.444°C	74°F	76°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent loads as defined in Section 1.5.3.1 are applied at the given zone air temperature.

- b. **Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

**AE105.TM2**

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. **Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.4.1.6, shall apply changes to parameters as indicated in Table 1-37; only values that change are shown in the table. Check output and adjust modeled internal gains so that for each zone: total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-36, and latent cooling load (latent internal gains) is equivalent to that specified in Table 1-20 (Case AE301).

**Table 1-37. Case AE405 Alternate Zone Definition Input Parameters**

Input Parameter	SI Units		I-P Units	
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W		567.8 (hr·ft <sup>2</sup> ·°F)/Btu	
Supplementary	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
Internal Gains – Sensible	1462.7 W	2343.8 W	4990.8 Btu/hr	7997.3 Btu/hr

**1.5.5.4.2.4 Case AE406: Low Cooling, Wet Coil 1**

*Informative Note, Objective.* Generate results for a variable air volume terminal reheat system without dry-bulb economizer, applying low sensible cooling load at warm, humid outdoor air conditions, to compare with results for the same system with dry-bulb economizer (Case AE426).

*Informative Note, Method.* Use Case AE401 model with modifications. Case AE406 applies an ambient dry-bulb temperature that ensures the economizer specified in Case AE426 will operate. Humid ambient conditions and high sensible cooling ensure a cold cooling coil with high latent loads. Check the cooling coil leaving air saturation to verify coil bypass factor is zero or near zero. Compare coil loads with the QAS and with results for Case AE426 (which includes a comparative dry-bulb economizer system).

**1.5.5.4.2.4.1 Input Specification.** This case shall be identical to Case AE401 (See Section 1.5.5.4.1) except for changes as follow.

- a. **Zone and ambient parameters.** Zone and ambient parameters shall be modeled as specified in Table 1-38; only values that change are shown in the table.

**Table 1-38. Case AE406 Input Parameters**

<b>Ambient Input Parameters</b>	<b>SI Units</b>		<b>I-P Units</b>	
Dry-Bulb Temperature	23.0°C		73.40°F	
Dew-Point Temperature <sup>a</sup>	20.9°C		69.62°F	
Equivalent Ambient Moisture Indicators <sup>a</sup>				
Humidity Ratio	0.015625		0.015625	
Wet-Bulb Temperature	21.523°C		70.741°F	
Relative Humidity	87.968 %		87.968 %	
Relative Humidity (TM2) <sup>b</sup>	88 %		88 %	
<b>Zone Input Parameters</b>	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
Zone Sensible Cooling Load <sup>c</sup>	1465 W	2345 W	5000 Btu/hr	8000 Btu/hr
Zone Dry-Bulb Temperature <sup>d</sup>	23.333°C	24.444°C	74°F	76°F

<sup>a</sup> **Informative Note:** Dew-point temperature is exact. The equivalent ambient moisture variables are calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

<sup>b</sup> **Informative Note:** The weather data file relative humidity precision is limited to integer percent values by the weather file format, and is not as precise as the other equivalent moisture variables. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in Appendix A, Tables A-4 and A-5.

<sup>c</sup> Zone sensible cooling load is defined in Section 1.5.3.1.

<sup>d</sup> Zone latent loads as defined in Section 1.5.3.1 are applied at the given zone air temperature.

- b. Weather data.** If the tested program does not allow constant steady-state ambient conditions to be input directly, use the TMY2-format weather data provided with the file:

**AE106.TM2**

These data are provided with the electronic files accompanying this report and are described in Appendix A.

- c. Alternate zone definition parameters.** Programs using the alternate zone definition described in Section 1.5.5.4.1.6 shall apply changes to parameters as indicated in Table 1-39; only values that change are shown in the table. Check output and adjust modeled internal gains so that for each zone: total zone sensible cooling load (envelope sensible heat gains plus all sensible internal gains) is equivalent to that specified in Table 1-38, and latent cooling load (latent internal gains) is equivalent to that specified in Table 1-20 (Case AE301).

**Table 1-39. Case AE406 Alternate Zone Definition Input Parameters**

<b>Input Parameter</b>	<b>SI Units</b>		<b>I-P Units</b>	
Insulation Resistance	100.0 (m <sup>2</sup> ·K)/W		567.8 (hr·ft <sup>2</sup> ·°F)/Btu	
Supplementary Internal Gains – Sensible	<b>Zone 1</b>	<b>Zone 2</b>	<b>Zone 1</b>	<b>Zone 2</b>
	1465.4 W	2347.0 W	5000.0 Btu/hr	8008.4 Btu/hr

### **1.5.5.4.3 Variable Air Volume Terminal Reheat System Parametric Variations, With Economizer**

#### **1.5.5.4.3.1 Case AE426: Low Cooling, Wet Coil 1 with Return Air Comparative Dry-Bulb Economizer Outdoor Air Control**

This test case shall be modeled with identical zone and system inputs as Case AE406, except for the existence of a return air comparative dry-bulb economizer as described in Section 1.5.5.3.3.1.1.1.

**Informative Note, Objective.** Test model treatment of a variable air volume terminal reheat system with return air comparative dry-bulb economizer with low sensible cooling load and warm, humid outdoor air.

**Informative Note, Method.** Use the Case AE406 model as defined in Section 1.5.5.4.2.4 and add the economizer as described in Section 1.5.5.3.3.1.1.1. Compare coil loads with the QAS and with results for Case AE406.

**Informative Note:** For the given economizer control logic, at the given steady-state operating conditions some cooling loads are increased by varying amounts.

**1.5.5.4.3.1.1 Input Specification.** Case AE426 shall be identical to Case AE406 (see Section 1.5.5.4.2.4) except for the addition of a return air comparative dry-bulb temperature economizer outdoor air control as specified in Section 1.5.5.3.3.1.1.1. **Informative Note:** The operation of the VAV system supply air fan is independent of the operation of the economizer. The VAV system supply air fan continues to modulate the supply airflow as needed to satisfy zone loads.

#### **1.5.5.4.3.2 Case AE445: Low Cooling, Dry Coil 1 with Return Air Comparative Enthalpy Economizer Outdoor Air Control**

This test case shall be modeled with identical zone and system inputs as Case AE405 except for the existence of a return air comparative enthalpy economizer as described in Section 1.5.5.3.3.2.1.1.

**Informative Note, Objective.** Test model treatment of a variable air volume terminal reheat system with return air comparative enthalpy economizer with low sensible cooling load at warm, dry outdoor air conditions.

**Informative Note, Method.** Use the Case AE405 model as defined in Section 1.5.5.3.2.3 and add the economizer as described in Section 1.5.5.3.3.2.1.1. Compare coil loads with the QAS and with results for Case AE405.

**Informative Note:** For the given economizer control logic, at the given steady-state operating conditions some cooling loads are increased by varying amounts.

**1.5.5.4.3.2.1 Input Specification.** Case AE445 shall be identical to Case AE405 (see Section 1.5.5.4.2.3) except for the addition of a return air comparative enthalpy economizer outdoor air control as specified in Section 1.5.5.3.3.2.1.1. **Informative Note:** The operation of the VAV system supply air fan is independent of the operation of the economizer. The VAV system supply air fan continues to modulate the supply airflow as needed to satisfy zone loads.

## 1.6 Output Requirements

**Sections 1.6.1 through 1.6.4 intentionally left blank to facilitate renumbering for future Standard 140 adaptation.**

### 1.6.5 Output Requirements for Air Handling Equipment Tests of Section 1.5.5

Outputs that are not direct program outputs (i.e., hand calculations) shall be noted in the output spreadsheet and modeler report using S140outNotes.txt (under report Block B, Alternative Modeling Methods). [This file is available with ASHRAE Standard 140-2014 (ANSI/ASHRAE 2014).]

#### 1.6.5.1 Output Requirements for AE100 Series Cases

Output results specified in this section shall be reported in the output spreadsheet (“Sec5-5out.xls”), using the nomenclature and units specified here. The simulation shall be run until the final hour output agrees with the previous hour output, and the values specified in Sections 1.6.5.1.1, 1.6.5.1.2, and 1.6.5.1.3 shall be reported for the last hour of the simulation.

##### 1.6.5.1.1 Zone Load Outputs – Last Hour of Simulation

- a. Zone Sensible Heating Load ( $QZH_{\text{Sensible}}$  [kWh/h])
- b. Zone Sensible Cooling Load ( $QZC_{\text{Sensible}}$  [kWh/h])
- c. Zone Latent Load ( $QZ_{\text{Latent}}$  [kWh/h]).

Zone loads are defined in Section 1.5.3.1. All zone loads shall be entered as positive values ( $\geq 0$ ). When a zone sensible cooling load occurs, the zone sensible heating load shall be blank, and vice versa.

##### 1.6.5.1.2 Heating and Cooling Coil Outputs – Last Hour of Simulation

- a. Heating Coil Load ( $QH$  [kWh/h])
- b. Cooling Coil Sensible Load ( $QC_{\text{Sensible}}$  [kWh/h])
- c. Cooling Coil Latent Load ( $QC_{\text{Latent}}$  [kWh/h])
- d. Cooling Coil Total Load ( $QC_{\text{Total}} = QC_{\text{Sensible}} + QC_{\text{Latent}}$  [kWh/h])
- e. Cooling Coil Leaving Air Relative Humidity ( $RH_{\text{cco}}$  [%], as  $0\% \leq RH_{\text{cco}} \leq 100\%$ ).

Coil loads are defined in Section 1.5.3.1. All coil loads shall be entered as positive values ( $\geq 0$ ).

**1.6.5.1.3 Detailed Outputs – Last Hour of Simulation.** Provide the dry-bulb temperature ( $^{\circ}\text{C}$ ), humidity ratio (g/gda), specific volume (L/kgda), enthalpy (J/gda), and mass flow rate (kgda/s), for the following locations as indicated by *italicized labels* in Figure 1-1:

- a. Outdoor air inlet (oa)
- b. Mixed air (ma)
- c. Heating coil outlet (hco)
- d. Cooling coil outlet (cco)
- e. Zone supply air (sa)
- f. Zone air (z1), except mass flow rate
- g. Return air (ra).

### 1.6.5.2 Output Requirements for AE200 Series Cases

Output results specified in this section shall be reported in the output spreadsheet (“Sec5-5out.xls”) using the nomenclature and units specified here. The simulation shall be run until the final hour output agrees with the previous hour output, and the values specified in Sections 1.6.5.2.1, 1.6.5.2.2, and 1.6.5.2.3 shall be reported for the last hour of the simulation.

#### 1.6.5.2.1 Zone Load Outputs – Last Hour of Simulation

The requirements of Section 1.6.5.1.1 shall be applied.

#### 1.6.5.2.2 Heating and Cooling Coil Outputs – Last Hour of Simulation

The requirements of Section 1.6.5.1.2 shall be applied.

**1.6.5.2.3 Detailed Outputs – Last Hour of Simulation.** Provide the dry-bulb temperature (°C), humidity ratio (g/gda), specific volume (L/kgda), enthalpy (J/gda), and mass flow rate (kgda/s) for the following locations as indicated by *italicized labels* in Figure 1-3:

- a. Outdoor air inlet (oa)
- b. Mixed air (ma)
- c. Heating coil outlet (hco)
- d. Cooling coil outlet (cco)
- e. Zone supply air (sa)
- f. Zone air (z1), except mass flow rate
- g. Return fan inlet (rfi)
- h. Return fan outlet (rfo).

### 1.6.5.3 Output Requirements for AE300 Series Cases

Output results specified in this section shall be reported in the output spreadsheet (“Sec5-5out.xls”), using the nomenclature and units specified here. The simulation shall be run until the final hour output agrees with the previous hour output, and the values specified in Sections 1.6.5.3.1, 1.6.5.3.2, and 1.6.5.3.3 shall be reported for the last hour of the simulation.

#### 1.6.5.3.1 Zone Load Outputs – Last Hour of Simulation

- a. Zone 1 Sensible Heating Load ( $QZH1_{Sensible}$  [kWh/h])
- b. Zone 1 Sensible Cooling Load ( $QZC1_{Sensible}$  [kWh/h])
- c. Zone 1 Latent Load ( $QZ1_{Latent}$  [kWh/h])
- d. Zone 2 Sensible Heating Load ( $QZH2_{Sensible}$  [kWh/h])
- e. Zone 2 Sensible Cooling Load ( $QZC2_{Sensible}$  [kWh/h])
- f. Zone 2 Latent Load ( $QZ2_{Latent}$  [kWh/h]).

Zone loads are defined in Section 1.5.3.1. All zone loads shall be entered as positive values ( $\geq 0$ ). When a zone sensible cooling load occurs, the zone sensible heating load shall be blank, and vice versa.

#### 1.6.5.3.2 Heating and Cooling Coil Outputs – Last Hour of Simulation

- a. Pre-heat Coil Load ( $QH_{Preheat}$  [kWh/h])
- b. Cooling Coil Sensible Load ( $QC_{Sensible}$  [kWh/h])
- c. Cooling Coil Latent Load ( $QC_{Latent}$  [kWh/h])
- d. Cooling Coil Total Load ( $QC_{total} = QC_{Sensible} + QC_{Latent}$  [kWh/h])



- e. Cooling Coil Leaving Air Relative Humidity ( $RH_{cco}$  [%], as  $0\% \leq RH_{cco} \leq 100\%$ )
- f. Zone 1 Reheat Coil Load ( $QH1_{Reheat}$  [kWh/h])
- g. Zone 2 Reheat Coil Load ( $QH2_{Reheat}$  [kWh/h]).

Coil loads are defined in Section 1.5.3.1. All coil loads shall be entered as positive values ( $\geq 0$ ).

**1.6.5.3.3 Detailed Outputs – Last Hour of Simulation.** Provide the dry-bulb temperature ( $^{\circ}C$ ), humidity ratio (g/gda), specific volume (L/kgda), enthalpy (J/gda), and mass flow rate (kgda/s) for the following locations, as indicated by *italicized labels* in Figure 1-4:

- a. Outside air (oa)
- b. Mixed air (ma)
- c. Pre-heat coil outlet (pco)
- d. Cooling coil outlet (cco)
- e. System supply air (sa)
- f. Zone 1 supply air, from reheat coil outlet (z1s)
- g. Zone 2 supply air, from reheat coil outlet (z2s)
- h. Zone 1 (z1), except mass flow rate
- i. Zone 2 (z2), except mass flow rate
- j. Return fan inlet (rfi)
- k. Return fan outlet (rfo).

#### 1.6.5.4 Output Requirements for AE400 Series Cases

Output results specified in this section shall be reported in the output spreadsheet (“Sec5-5out.xls”), using the nomenclature and units specified here. The simulation shall be run until the final hour output agrees with the previous hour output, and the values specified in Sections 1.6.5.4.1, 1.6.5.4.2, and 1.6.5.4.3 shall be reported for the last hour of the simulation.

##### 1.6.5.4.1 Zone Load Outputs – Last Hour of Simulation

The requirements of Section 1.6.5.3.1 shall be applied.

##### 1.6.5.4.2 Heating and Cooling Coil Outputs – Last Hour of Simulation

The requirements of Section 1.6.5.3.2 shall be applied.

**1.6.5.4.3 Detailed Outputs – Last Hour of Simulation.** Provide the dry-bulb temperature ( $^{\circ}C$ ), humidity ratio (g/gda), specific volume (L/kgda), enthalpy (J/gda), and mass flow rate (kgda/s) for the following locations, as indicated by *italicized labels* in Figure 1-5:

- a. Outside air (oa)
- b. Mixed air (ma)
- c. Pre-heat coil outlet (pco)
- d. Cooling coil outlet (cco)
- e. System supply air (sa)
- f. Zone 1 supply air, from VAV terminal reheat coil outlet (z1s)
- g. Zone 2 supply air, from VAV terminal reheat coil outlet (z2s)
- h. Zone 1 (z1), except mass flow rate
- i. Zone 2 (z2), except mass flow rate
- j. Return fan inlet (rfi)
- k. Return fan outlet (rfo).

## Appendix A. TMY2 Weather Data for Airside HVAC Equipment Tests

For programs that do not allow direct entry of ambient conditions as specified in Section 1.5.5, the full-year weather data on the electronic media provided with this report shall be used for performing the tests of Section 1.5.5 as assigned in Table A-1. Weather file and site characteristics are summarized in Table A-2. These weather data files are based on Miami.TM2, but have a number of data elements set to constant values and solar radiation off, as specified in Table A-2. The weather data also include many data elements set to 0 or approximate lower limits, and other data elements set to neutral (non-extreme) constant values, as specified in Table A-2.

TMY2 weather data format description is originally available in Marion and Urban (1995). This information may be more easily obtained from ASHRAE Standard 140-2014 (ANSI/ASHRAE 2014), Annex A1, Section A1.6, or HVAC BESTEST Volume 2 (Neymark and Judkoff 2004), Appendix A.

**Table A-1. Weather Data for Air Handling Equipment Tests**

<b>Data Files</b>	<b>Applicable Cases/Output</b>	<b>Sections</b>
AE101.TM2	AE101, AE201, AE301, AE401	1.5.5.1.1, 1.5.5.2.1, 1.5.5.3.1, 1.5.5.4.1
AE103.TM2	AE103, AE203, AE303, AE403	1.5.5.1.2.1, 1.5.5.2.2.1, 1.5.5.3.2.1, 1.5.5.4.2.1
AE104.TM2	AE104, AE204, AE304, AE404	1.5.5.1.2.2, 1.5.5.2.2.2, 1.5.5.3.2.2, 1.5.5.4.2.2
AE105.TM2	AE205, AE245, AE305, AE345, AE405, AE445	1.5.5.2.2.3, 1.5.5.2.3.2, 1.5.5.3.2.3, 1.5.5.3.3.2, 1.5.5.4.2.3, 1.5.5.4.3.2
AE106.TM2	AE206, AE226, AE306, AE326, AE406, AE426	1.5.5.2.2.4, 1.5.5.2.3.1, 1.5.5.3.2.4, 1.5.5.3.3.1, 1.5.5.4.2.4, 1.5.5.4.3.1

**Table A-2. Weather File and Site Summary for Air Handling Equipment Tests**

Weather Type	Artificial Conditions				
Weather Format	TMY2				
Latitude	25.8° North				
Longitude (local site)	80.3° West				
Altitude	0 m (0 ft) <sup>a</sup>				
Time Zone (Standard Meridian Longitude)	5 (75° West)				
Site	Flat, unobstructed, located exactly at weather station				
Atmospheric Pressure (constant)	1013 millibars (14.692 psia) <sup>a</sup>				
Wind Speed (constant)	4.3 m/s (9.62 miles/h) <sup>b</sup>				
Global Horizontal Solar Radiation Annual Total	0 kWh/m <sup>2</sup> (0 kBtu/ft <sup>2</sup> )				
Direct Normal Solar Radiation Annual Total	0 kWh/m <sup>2</sup> (0 kBtu/ft <sup>2</sup> )				
Diffuse Horizontal Solar Radiation Annual Total	0 kWh/m <sup>2</sup> (0 kBtu/ft <sup>2</sup> )				
Illuminance	0 kWh/m <sup>2</sup> (0 kBtu/ft <sup>2</sup> )				
Total and Opaque Sky Cover	10 tenths <sup>c</sup>				
Visibility	20 km <sup>d</sup>				
Ceiling Height	2000 m <sup>d</sup>				
Aerosol Optical Depth	0.1 broadband turbidity <sup>d</sup>				
Present Weather:	No rain, hail, etc.				
Precipitable water	0 mm				
Snow Depth	0 cm, with ≥ 88 days since last snowfall.				
<b>Quantities That Vary between Data Sets</b>					
Weather File Name	<b>AE101.TM2</b>	<b>AE103.TM2</b>	<b>AE104.TM2</b>	<b>AE105.TM2</b>	<b>AE106.TM2</b>
Dry-Bulb Temperature	-29.0°C (-20.20°F)	15.5°C (59.90°F)	26.9°C (80.42°F)	24.9°C (76.82°F)	23.0°C (73.40°F)
Dew-Point Temperature	-29.0°C (-20.20°F)	-3.0°C (26.60°F)	22.1°C (71.78°F)	2.4°C (36.32°F)	20.9°C (69.62°F)
Relative Humidity <sup>e</sup>	100%	27%	75%	23%	88%

<sup>a</sup> **Informative Note:** The weather data file indicates 0 m altitude, which corresponds with atmospheric pressure of 101325 Pa (14.696 psia) specified in Section 1.5.5.1.1.7. However, the weather file atmospheric pressure precision is limited to four digits by the weather file format such that 1013 millibars (101300 Pa = 14.692 psia) is listed in the weather files. The effect of modeling with 101325 Pa versus 101300 Pa is negligible (< 0.05% on cooling coil latent loads) as shown in informative Table A-3.

<sup>b</sup> **Informative Note:** Miami.TM2 annual average wind speed = 4.34 m/s, but TM2 weather data allow input only to the nearest 0.1 m/s.

<sup>c</sup> **Informative Note:** Total and opaque sky cover of 10 tenths implies the entire sky dome is covered by clouds; this setting is intended to reduce exterior infrared radiation exchange.

<sup>d</sup> **Informative Note:** Rough annual average for Miami and Denver.

<sup>e</sup> **Informative Note:** The weather file relative humidity precision is limited to integer percent values by the weather file format. Among the test cases, weather file relative humidities are different from relative humidities specified in various tables of Section 1.5.5 by < 0.22%. The effects of modeling with weather file relative humidities versus specified relative humidities or versus weather file (specified) dew-point temperatures, are small (< 0.3% on cooling coil latent loads), as shown in informative Tables A-4 and A-5.

**Informative Note, Dew-Point Temperature, Relative Humidity, and Atmospheric Pressure.** The outdoor dew-point temperature shown in Table A-2 is exact. Weather file relative humidity and atmospheric pressure have reduced precision because of the precision limits of the weather file format. To develop estimates of the effect of these precision limits, the QAS was modified to allow variable atmospheric pressure and to accept relative humidity as an input variable. The differences in QAS results with atmospheric pressure of 101325 Pa versus 101300 Pa are negligible (< 0.05% on cooling coil latent loads), as shown in informative Table A-3. The differences in QAS results with weather file relative humidities versus relative humidities specified in various tables of Section 1.5.5 are small (< 0.3% on cooling coil latent loads), as shown in informative Table A-4. The differences in QAS results from applying weather file relative humidities versus weather file (and specified) dew-point temperatures are also small (< 0.14% on cooling coil latent loads), as shown in informative Table A-5.

**Table A-3. (Informative) QAS Predicted Humidity Ratio (W) and Coil Loads for Atmospheric Pressure of 101300 Pa versus 101325 Pa**

Case	101325 Pa				101300 Pa				101300 Pa to 101325 Pa (%)			
	W (kg/kg)	Coil Loads (kWh)			W (kg/kg)	Coil Loads (kWh)			W	Coil Loads		
		Sens	Lat	Total		Sens	Lat	Total		Sens	Lat	Total
201	0.0002589	-8.360	0.000	-8.360	0.0002590	-8.359	0.000	-8.359	0.025%	-0.017%		-0.017%
203	0.0029359	0.856	0.000	0.856	0.0029366	0.856	0.000	0.856	0.025%	0.026%		0.026%
204	0.0167736	3.574	2.054	5.628	0.0167778	3.574	2.055	5.629	0.025%	-0.002%	0.021%	0.006%
205	0.0044909	1.916	0.000	1.916	0.0044920	1.916	0.000	1.916	0.025%	-0.002%		-0.002%
206	0.0155555	1.706	0.970	2.677	0.0155595	1.706	0.971	2.677	0.025%	0.001%	0.028%	0.011%
226	0.0155555	1.562	1.800	3.362	0.0155595	1.562	1.801	3.363	0.025%	0.002%	0.047%	0.026%
245	0.0044909	2.208	0.000	2.208	0.0044920	2.208	0.000	2.208	0.025%	-0.006%		-0.006%

**Table A-4. (Informative) QAS Predicted Humidity Ratio (W) and Coil Loads for Weather file Relative Humidity (RH) versus Specified RH**

Case	Specified RH				Weather file RH				Weather RH to Specified RH (%)			
	W (kg/kg)	Coil Loads (kWh)			W (kg/kg)	Coil Loads (kWh)			W	Coil Loads		
		Sens	Lat	Total		Sens	Lat	Total		Sens	Lat	Total
201	0.0002589	-8.360	0.000	-8.360	0.0002589	-8.360	0.000	-8.360	0.000%	0.000%		0.000%
203	0.0029355	0.856	0.000	0.856	0.0029325	0.856	0.000	0.856	-0.104%	0.000%		0.000%
204	0.0167716	3.574	2.054	5.628	0.0167663	3.574	2.052	5.626	-0.031%	-0.001%	-0.069%	-0.026%
205	0.0044894	1.916	0.000	1.916	0.0044796	1.916	0.000	1.916	-0.218%	0.000%		0.000%
206	0.0155547	1.706	0.970	2.677	0.0155605	1.706	0.972	2.678	0.037%	0.001%	0.161%	0.059%
226	0.0155547	1.562	1.799	3.361	0.0155605	1.562	1.804	3.366	0.037%	0.002%	0.265%	0.143%
245	0.0044894	2.208	0.000	2.208	0.0044796	2.208	0.000	2.208	-0.218%	0.000%		0.000%

**Table A-5. (Informative) QAS Predicted Humidity Ratio (W) and Coil Loads for Weather File Relative Humidity (RH) versus Weather File (Specified) Dew-Point Temperature (DP)**

Case	Weather File (Specified) DP				Weather File RH				Specified RH to Specified DP (%)			
	W (kg/kg)	Coil Loads (kWh)			W (kg/kg)	Coil Loads (kWh)			W	Coil Loads		
		Sens	Lat	Total		Sens	Lat	Total		Sens	Lat	Total
201	0.0002589	-8.360	0.000	-8.360	0.0002589	-8.360	0.000	-8.360	0.000%	0.000%		0.000%
203	0.0029359	0.856	0.000	0.856	0.0029325	0.856	0.000	0.856	-0.115%	0.000%		0.000%
204	0.0167736	3.574	2.054	5.628	0.0167663	3.574	2.052	5.626	-0.043%	-0.001%	-0.095%	-0.035%
205	0.0044909	1.916	0.000	1.916	0.0044796	1.916	0.000	1.916	-0.250%	0.000%		0.000%
206	0.0155555	1.706	0.970	2.677	0.0155605	1.706	0.972	2.678	0.032%	0.000%	0.138%	0.050%
226	0.0155555	1.562	1.800	3.362	0.0155605	1.562	1.804	3.366	0.032%	0.001%	0.227%	0.123%
245	0.0044909	2.208	0.000	2.208	0.0044796	2.208	0.000	2.208	-0.250%	0.000%		0.000%

## Appendix B. Tabular Summary of Test Cases

Tables B-1 through B-8 summarize the airside HVAC equipment test cases of Section 1.5.5 in I-P and SI units; these tables are provided for background information only. Use the test specification beginning in Section 1.5.5 to develop simulation input files.

### Nomenclature

Abbreviations and symbols used in Tables B-1 through B-8 are listed below.

DB     dry-bulb temperature  
 DP     dew-point temperature  
 Num.   zone number (1 or 2)

### Summary Tables

**Table B-1. Section 1.5.5.1 Fan Coil Case Descriptions – I-P Units**

Case	Ambient		Zone	Zone Loads (Btu/h) <sup>a</sup>			Coil Load Type	Section Number
	DB	DP	DB	Sensible		Latent		
	(°F)	(°F)	(°F)	Heating	Cooling			
No Economizer								
AE101	-20.20	-20.20	70	10000	0	2000	High Heating	Section 1.5.5.1.1
AE103	59.90	26.60	74	0	5000	2000	Low Cooling, dry coil	Section 1.5.5.1.2.1
AE104	80.42	71.78	75	0	10000	2000	High Cooling, wet coil	Section 1.5.5.1.2.2

<sup>a</sup> Zone loads do not include outdoor air ventilation loads introduced by the mechanical equipment.

**Table B-2. Section 1.5.5.1 Fan Coil Case Descriptions – SI Units**

Case	Ambient		Zone	Zone Loads (W)			Coil Load Type	Section Number
	DB	DP	DB	Sensible		Latent		
	(°C)	(°C)	(°C)	Heating	Cooling			
No Economizer								
AE101	-29.0	-29.0	21.111	2931	0	586.1	High Heating	Section 1.5.5.1.1
AE103	15.5	-3.0	23.333	0	1465	586.1	Low Cooling, dry coil	Section 1.5.5.1.2.1
AE104	26.9	22.1	23.889	0	2931	586.1	High Cooling, wet coil	Section 1.5.5.1.2.2

<sup>a</sup> Zone loads do not include outdoor air ventilation loads introduced by the mechanical equipment.

**Table B-3. Section 1.5.5.2 Single Zone Case Descriptions – I-P Units**

Case	Ambient		Zone	Zone Loads (Btu/h) <sup>a</sup>			Coil Load Type	Section Number
	DB	DP	DB	Sensible		Latent		
	(°F)	(°F)	(°F)	Heating	Cooling			
No Economizer								
AE201	-20.20	-20.20	70	10000	0	2000	High Heating	Section 1.5.5.2.1
AE203	59.90	26.60	74	0	5000	2000	Low Cooling, dry coil	Section 1.5.5.2.2.1
AE204	80.42	71.78	75	0	10000	2000	High Cooling, wet coil	Section 1.5.5.2.2.2
AE205	76.82	36.32	74	0	5000	2000	Low Cooling, dry coil	Section 1.5.5.2.2.3
AE206	73.40	69.62	74	0	5000	2000	Low Cooling, wet coil	Section 1.5.5.2.2.4
Return Air Comparative Dry-Bulb Economizer Outdoor Air Control								
AE226	73.40	69.62	74	0	5000	2000	Low Cooling, wet coil	Section 1.5.5.2.3.1. AE206 with dry-bulb economizer.
Return Air Comparative Enthalpy Economizer Outdoor Air Control								
AE245	76.82	36.32	74	0	5000	2000	Low Cooling, dry coil	Section 1.5.5.2.3.2. AE205 with enthalpy economizer.

<sup>a</sup> Zone loads do not include outdoor air ventilation loads introduced by the mechanical equipment.

**Table B-4. Section 1.5.5.2 Single Zone Case Descriptions – SI Units**

Case	Ambient		Zone	Zone Loads (W)			Coil Load Type	Section Number
	DB	DP	DB	Sensible		Latent		
	(°C)	(°C)	(°C)	Heating	Cooling			
No Economizer								
AE201	-29.0	-29.0	21.111	2931	0	586.1	High Heating	Section 1.5.5.2.1
AE203	15.5	-3.0	23.333	0	1465	586.1	Low Cooling, dry coil	Section 1.5.5.2.2.1
AE204	26.9	22.1	23.889	0	2931	586.1	High Cooling, wet coil	Section 1.5.5.2.2.2
AE205	24.9	2.4	23.333	0	1465	586.1	Low Cooling, dry coil	Section 1.5.5.2.2.3
AE206	23.0	20.9	23.333	0	1465	586.1	Low Cooling, wet coil	Section 1.5.5.2.2.4
Return Air Comparative Dry-Bulb Economizer Outdoor Air Control								
AE226	23.0	20.9	23.333	0	1465	586.1	Low Cooling, wet coil	Section 1.5.5.2.3.1. AE206 with dry-bulb economizer.
Return Air Comparative Enthalpy Economizer Outdoor Air Control								
AE245	24.9	2.4	23.333	0	1465	586.1	Low Cooling, dry coil	Section 1.5.5.2.3.2. AE205 with enthalpy economizer.

<sup>a</sup> Zone loads do not include outdoor air ventilation loads introduced by the mechanical equipment.

**Table B-5. Section 1.5.5.3 Constant Volume Terminal Reheat Case Descriptions – I-P Units**

Case	Ambient		Zone	Zone	Zone Loads (Btu/h) <sup>a</sup>			Coil Load Type	Section Number
	DB	DP	Num.	DB	Sensible		Latent		
	(°F)	(°F)		(°F)	Heating	Cooling			
No Economizer									
AE301	-20.20	-20.20	1	70	10000	0	2000	High Heating	Section 1.5.5.3.1
			2	72	8000	0	3000		
AE303	59.90	26.60	1	74	0	5000	2000	Low Cooling, dry coil	Section 1.5.5.3.2.1
			2	76	0	8000	3000		
AE304	80.42	71.78	1	75	0	10000	2000	High Cooling, wet coil	Section 1.5.5.3.2.2
			2	77	0	12000	3000		
AE305	76.82	36.32	1	74	0	5000	2000	Low Cooling, dry coil	Section 1.5.5.3.2.3
			2	76	0	8000	3000		
AE306	73.40	69.62	1	74	0	5000	2000	Low Cooling, wet coil	Section 1.5.5.3.2.4
			2	76	0	8000	3000		
Return Air Comparative Dry-Bulb Economizer Outdoor Air Control									
AE326	73.40	69.62	1	74	0	5000	2000	Low Cooling, wet coil	Section 1.5.5.3.3.1. AE306 with dry-bulb economizer.
			2	76	0	8000	3000		
Return Air Comparative Enthalpy Economizer Outdoor Air Control									
AE345	76.82	36.32	1	74	0	5000	2000	Low Cooling, dry coil	Section 1.5.5.3.3.2. AE305 with enthalpy economizer.
			2	76	0	8000	3000		

<sup>a</sup> Zone loads do not include outdoor air ventilation loads introduced by the mechanical equipment.

**Table B-6. Section 1.5.5.3 Constant Volume Terminal Reheat Case Descriptions – SI Units**

Case	Ambient		Zone	Zone	Zone Loads (W)			Coil Load Type	Section Number
	DB	DP	Num.	DB	Sensible		Latent		
	(°C)	(°C)		(°C)	Heating	Cooling			
No Economizer									
AE301	-29.0	-29.0	1	21.111	2931	0	586.1	High Heating	Section 1.5.5.3.1
			2	22.222	2345	0	879.2		
AE303	15.5	-3.0	1	23.333	0	1465	586.1	Low Cooling, dry coil	Section 1.5.5.3.2.1
			2	24.444	0	2345	879.2		
AE304	26.9	22.1	1	23.889	0	2931	586.1	High Cooling, wet coil	Section 1.5.5.3.2.2
			2	25.000	0	3517	879.2		
AE305	24.9	2.4	1	23.333	0	1465	586.1	Low Cooling, dry coil	Section 1.5.5.3.2.3
			2	24.444	0	2345	879.2		
AE306	23.0	20.9	1	23.333	0	1465	586.1	Low Cooling, wet coil	Section 1.5.5.3.2.4
			2	24.444	0	2345	879.2		
Return Air Comparative Dry-Bulb Economizer Outdoor Air Control									
AE326	23.0	20.9	1	23.333	0	1465	586.1	Low Cooling, wet coil	Section 1.5.5.3.3.1. AE306 with dry-bulb economizer.
			2	24.444	0	2345	879.2		
Return Air Comparative Enthalpy Economizer Outdoor Air Control									
AE345	24.9	2.4	1	23.333	0	1465	586.1	Low Cooling, dry coil	Section 1.5.5.3.3.2. AE305 with enthalpy economizer.
			2	24.444	0	2345	879.2		

<sup>a</sup> Zone loads do not include outdoor air ventilation loads introduced by the mechanical equipment.

**Table B-7. Section 1.5.5.4 Variable Air Volume Terminal Reheat Case Descriptions – I-P Units**

Case	Ambient		Zone Num.	Zone DB (°F)	Zone Loads (Btu/h) <sup>a</sup>			Coil Load Type	Section Number
	DB (°F)	DP (°F)			Sensible		Latent		
			Heating	Cooling					
No Economizer									
AE401	-20.20	-20.20	1	70	10000	0	2000	High Heating	Section 1.5.5.4.1
			2	72	8000	0	3000		
AE403	59.90	26.60	1	74	0	5000	2000	Low Cooling, dry coil	Section 1.5.5.4.2.1
			2	76	0	8000	3000		
AE404	80.42	71.78	1	75	0	10000	2000	High Cooling, wet coil	Section 1.5.5.4.2.2
			2	77	0	12000	3000		
AE405	76.82	36.32	1	74	0	5000	2000	Low Cooling, dry coil	Section 1.5.5.4.2.3
			2	76	0	8000	3000		
AE406	73.40	69.62	1	74	0	5000	2000	Low Cooling, wet coil	Section 1.5.5.4.2.4
			2	76	0	8000	3000		
Return Air Comparative Dry-Bulb Economizer Outdoor Air Control									
AE426	73.40	69.62	1	74	0	5000	2000	Low Cooling, wet coil	Section 1.5.5.4.3.1. AE406 with dry-bulb economizer.
			2	76	0	8000	3000		
Return Air Comparative Enthalpy Economizer Outdoor Air Control									
AE445	76.82	36.32	1	74	0	5000	2000	Low Cooling, dry coil	Section 1.5.5.4.3.2. AE405 with enthalpy economizer.
			2	76	0	8000	3000		

<sup>a</sup> Zone loads do not include outdoor air ventilation loads introduced by the mechanical equipment.

**Table B-8. Section 1.5.5.4 Variable Air Volume Terminal Reheat Case Descriptions – SI Units**

Case	Ambient		Zone Num.	Zone DB (°C)	Zone Loads (W)			Coil Load Type	Section Number
	DB (°C)	DP (°C)			Sensible		Latent		
			Heating	Cooling					
No Economizer									
AE401	-29.0	-29.0	1	21.111	2931	0	586.1	High Heating	Section 1.5.5.4.1
			2	22.222	2345	0	879.2		
AE403	15.5	-3.0	1	23.333	0	1465	586.1	Low Cooling, dry coil	Section 1.5.5.4.2.1
			2	24.444	0	2345	879.2		
AE404	26.9	22.1	1	23.889	0	2931	586.1	High Cooling, wet coil	Section 1.5.5.4.2.2
			2	25.000	0	3517	879.2		
AE405	24.9	2.4	1	23.333	0	1465	586.1	Low Cooling, dry coil	Section 1.5.5.4.2.3
			2	24.444	0	2345	879.2		
AE406	23.0	20.9	1	23.333	0	1465	586.1	Low Cooling, wet coil	Section 1.5.5.4.2.4
			2	24.444	0	2345	879.2		
Return Air Comparative Dry-Bulb Economizer Outdoor Air Control									
AE426	23.0	20.9	1	23.333	0	1465	586.1	Low Cooling, wet coil	Section 1.5.5.4.3.1. AE406 with dry-bulb economizer.
			2	24.444	0	2345	879.2		
Return Air Comparative Enthalpy Economizer Outdoor Air Control									
AE445	24.9	2.4	1	23.333	0	1465	586.1	Low Cooling, dry coil	Section 1.5.5.4.3.2. AE405 with enthalpy economizer.
			2	24.444	0	2345	879.2		

<sup>a</sup> Zone loads do not include outdoor air ventilation loads introduced by the mechanical equipment.



## Appendix C. References for Part I

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## 2.0 Part II: Production of Quasi-Analytical Solution (QAS) Results

### 2.1 Introduction

The term “quasi-analytical solution” is defined in American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Standard 140-2014, Section 3.1 (ANSI/ASHRAE 2014; also see Neymark and Judkoff et al. 2008) as:

“*[Q]uasi-analytical solution*: the mathematical solution of a model for a given set of parameters and simplifying assumptions, which is allowed to include minor interpretation differences that cause minor results variations. *Informative Note*: Such a solution may be computed by generally accepted numerical methods or other means, provided that such calculations occur outside the environment of a whole-building energy simulation program and can be scrutinized.”

Other terminology used in Part II is defined in Part I, Section 1.5.3.1.

The QAS results for all of the test cases specified in Part I, Section 1.5.5 are provided in Part IV along with the example simulation results. Also see the spreadsheet files “ResultsFCSZ.xlsm” and “ResultsCVVV.xlsm” included with the accompanying electronic files (see subfolder “PartIV-Files”). The results include summary heating and cooling coil loads and detailed outputs (e.g., mass flow rate, specific volume, enthalpy, etc.) at the output locations as specified in the test cases. See Section 3.5.1 for analysis of differences of other example simulation results versus the QAS results and versus each other.

Production of the QAS results required the development of a QAS model for each system type. The following sections describe the development of the QAS models, list key assumptions, and provide comparative figures in order to assess impacts of the development process. Section 2.2 describes the final QAS model calculations in detail, and how the QAS models evolved from the initial models provided with the original ASHRAE Research Project 865 (RP 865) research report (Yuill and Haberl 2002). Section 2.3 documents evolution of the QAS model results as this work progressed.

#### 2.1.1 Relative Importance of the QAS Results

The QAS results are important for a number of reasons:

- They are produced by a merged version of calculations initially prepared for the originating ASHRAE RP 865 final report (Yuill and Haberl 2002) by separate research teams at The Pennsylvania State University (PSU) and Texas A&M University (TAMU); as part of the initial preparation of the merged QAS, the PSU and TAMU calculation results were thoroughly compared and differences in the results were reconciled as described in Section 2.2.2.
- The merged versions of the calculations were prepared by the National Renewable Energy Laboratory (NREL) project team independently of the PSU and TAMU teams based on review of the RP 865 final PSU and TAMU calculations, and the final version of the merged calculation results was compared with the originating PSU and TAMU results as described in Section 2.3.
- The NREL team refinements of the merged calculation methods (which became the QAS) were further reviewed by the RP 865 authors (Yuill 2013, 2014; Haberl 2013)
- As the calculations exist outside of a whole-building energy simulation program, they are not constrained by a pre-existing whole-building modeling framework. Such external calculations

also cannot be affected by possible bugs in a larger whole building energy simulation program.

- The QAS results provided an early benchmark during the simulation trial process for both intended test specification descriptions and for simulation results by industry simulation trial participants; a number of software bugs, input errors, and test specification ambiguities were found and fixed in the industry working group models based on early simulation trial comparisons. During this process, only minor changes to the QAS were made.
- Documentation of this model may provide further insight into the test specification for some users, and provides an example vetted solution technique and modeling assumptions for solving the airside heating, ventilating, and air conditioning (HVAC) equipment modeling problem (i.e., is a repository for related modeling physics).

The QAS includes a number of simplifying assumptions to facilitate the solution, as documented in Sections 2.2 and 2.2.1. Tested software may have reasonable modeling assumptions that vary from those of the QAS.

## **2.2 Development of QAS Models**

ASHRAE RP 865 (Yuill and Haberl 2002) developed descriptions of seven air system types and had two independent teams develop models for each system type. PSU/University of Nebraska developed one model set (referred to here as PSU), and TAMU developed the other model set. Both model sets consist of equations implemented within a spreadsheet with an iterative solution process to solve the equations, given steady state ambient and zone conditions.

As part of RP 865, model development continued until the differences between the PSU and TAMU predicted total heating and cooling coil loads were generally less than 1% at six steady-state test conditions. The model developers did not have access to the other models or other model algorithms, only the model results. This process resulted in two independently developed calculation models for each system type. That they make similar predictions provides some confidence that the procedures used are accurate. However, both models utilize many assumptions in common.

The development of the QAS models began with a detailed comparison of the final RP 865 model results by the NREL project team. Patterns in the differences in the results between the models pointed to general areas where model calculations were likely to differ. The model equations were examined to determine specific equations and methods that differed. For each identified difference, a draft revision was developed, generally following the methods of one of the two model sets. A list of the differences along with proposed revisions was submitted to the original RP 865 authors for comment. Based on their comments, final revisions were made by the NREL team to create a single QAS model for each system type.

Overall, a small number of calculation differences explained nearly all variation in the original RP 865 model results. A full listing of calculation differences along with the resolution chosen for the final model is presented in Section 2.2.2.2. Further checking of the QAS is discussed in Section 2.3.3. Many of the items identified cause small differences that ordinarily may not be of concern, but in the context of vetting the QAS, understanding the differences and finding a common approach to these details is important.

### **2.2.1 Model Calculation Description**

The following documentation was developed for this project. In addition to the following detailed description, a pro-forma modeler report summary of the QAS is provided in Part III, Section 3.10. This

summary allows quick comparison with the pro-forma reports provided for the simulation programs used for developing additional Part IV example results (see Part III, Section 3.9 for those pro-forma reports).

The QAS models are contained in “QASv2.xlsm”, which is included with the accompanying electronic files (see subfolder “PartII-Files”). QAS models were created for the fan coil (FC), single zone (SZ), constant volume (CV), and variable air volume (VAV) system types. There is one worksheet tab for each system model. Instructions for using the spreadsheet are contained on the “ReadMe” tab in the spreadsheet.

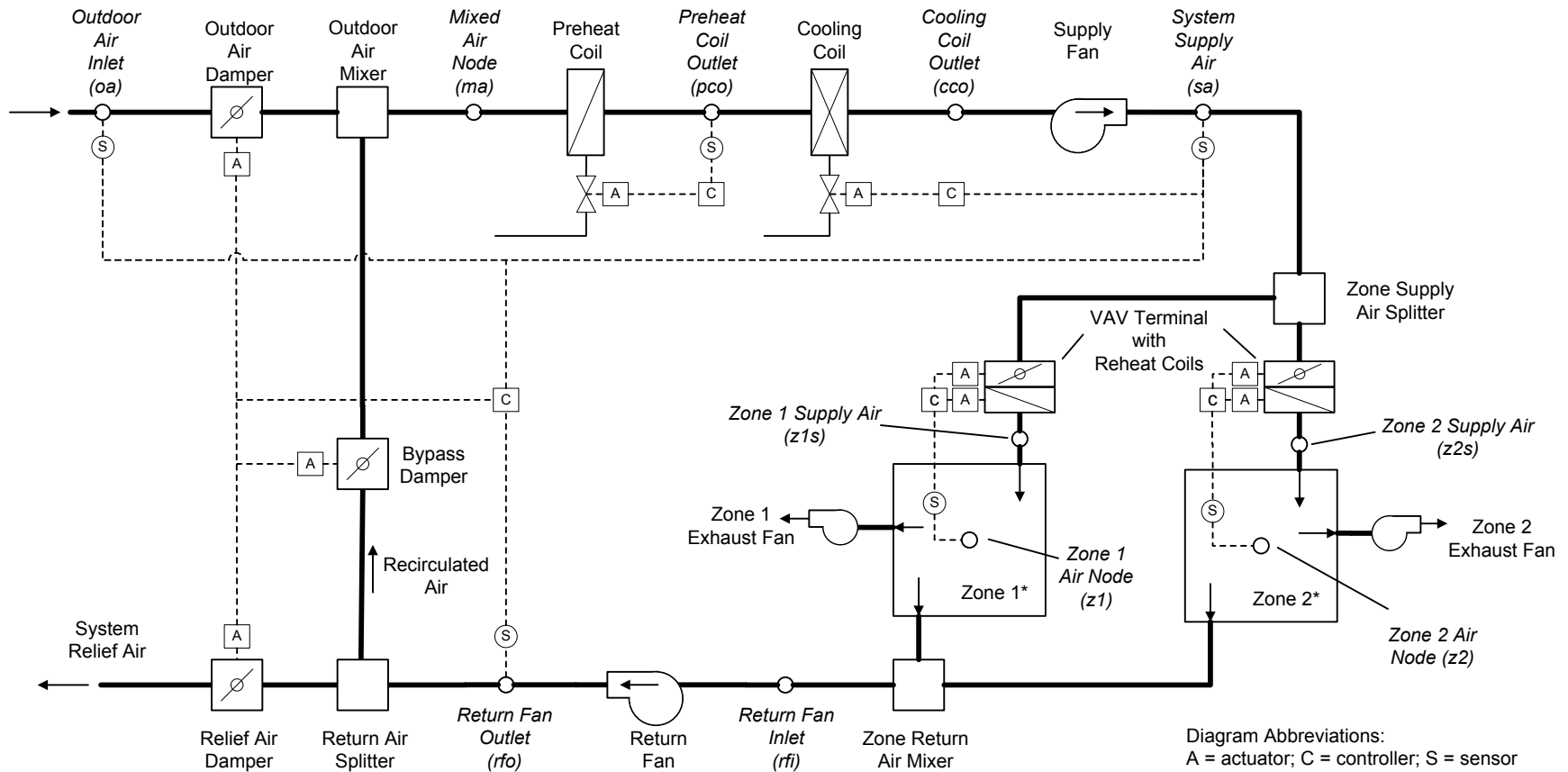
The QAS models implement standard engineering methods, including fundamental mass flow and energy balances. The basic premise of the test cases and the QAS models is that supply and zone-exhaust fans are volumetrically constant while the mass flows vary as the temperature and humidity ratio of the air entering the fans changes. The mass flow of the air entering the supply fan and exhaust fan(s) is calculated for each test case using the specified volumetric flow and the specific volume of air entering the fan. The specific volume of the air entering the fan is calculated iteratively with the model predicted entering air temperature and humidity ratio.

The system supply and exhaust fan mass flows define the key elements of a complete mass balance. The mass flow of outdoor air is defined as equal to the exhaust air mass flow when the economizer is not operating, and as equal to the system supply air mass flow when the economizer is operating. The return air mass flow is defined as the difference between the system supply air and the zone exhaust air mass flows. The system relief air mass flow is always the difference between the system outdoor air and the zone exhaust air mass flows.

Model calculation simplifications and assumptions not specified by the test cases include:

- Assumption of constant pressure (1 atm = 14.695975 psia) throughout the systems
- Use of ideal gas law to calculate air properties, except for the real gas correlation used for water vapor saturation pressure (e.g., see Sections 2.2.1.17 and 2.2.1.23)
- Use of constant specific heat of dry air (0.2403 Btu/(lb<sub>m,da</sub>·°F))
- Use of constant specific heat of water vapor (0.444 Btu/(lb<sub>m,w</sub>·°F))
- Use of constant specific heat of water (1.000 Btu/(lb<sub>m,w</sub>·°F))
- Cooling coil condensation occurs at the coil entering air dew-point temperature
- Cooling coil condensate leaves the system at the coil leaving air temperature.

An example schematic diagram is provided in Figure 2-1 to facilitate the discussion of the QAS calculations. The VAV system schematic is provided here because it has the most complexity. Readers may prefer to also refer to the FC, SZ, and CV schematic diagrams (see Part I, Figures 1-1, 1-3, and 1-4), as appropriate for the discussion.



\* Sensible and latent zone loads are specified for the base case in Section 1.5.5.3.1.4 and vary among the test cases. The zone thermostats sense only the zone air indoor dry-bulb temperature at their respective zone air nodes.

**Figure 2-1. Variable air volume air system schematic (VAV)**

**Note:** Valves indicated are for a typical hydronic system and are not explicitly required by the test specification. Coils can be of any type as long as they meet the operational requirements of the test specification.

The following calculations are provided in the QAS spreadsheet and summarized in the following subsections sequentially, beginning with the supply fan (where volumetric system supply airflow rate is defined and constant for the FC, SZ, and CV systems):

- Supply air mass flow rate (Section 2.2.1.1)
- Supply fan air temperature rise (Section 2.2.1.2)
- System supply air temperature, CV and VAV systems (Section 2.2.1.3)
- Zone supply air mass flow rates (Section 2.2.1.4)
- Zone supply air temperatures. (Section 2.2.1.5)
  - Note: integrates zone sensible load (heat gain or loss)
- Reheat coil energy, CV and VAV systems (Section 2.2.1.6)
- Zone air temperature (Section 2.2.1.7)
- Zone humidity ratio and zone latent heat (moisture) gains (Section 2.2.1.8)
- Zone exhaust air mass flow rates (Section 2.2.1.9)
- Return air mass flow rate (Section 2.2.1.10)
- Return fan inlet air temperature (Section 2.2.1.11)
- Return air humidity ratio (Section 2.2.1.12)
- Return fan air temperature rise (Section 2.2.1.13)
- Return air temperature (Section 2.2.1.14)
- Relief air mass flow rate (Section 2.2.1.15)
- Recirculated air mass flow rate (Section 2.2.1.16)
- Outdoor air humidity ratio (Section 2.2.1.17)
- Outdoor air mass flow rate and economizer operation (Section 2.2.1.18)
  - Economizer control logic (Section 2.2.1.18.1)
- Mixed air mass flow rate (Section 2.2.1.19)
- Mixed air temperature (Section 2.2.1.20)
- Mixed air humidity ratio (Section 2.2.1.21)
- Target after-coil air temperatures (Section 2.2.1.22)
- Cooling coil leaving air humidity ratio (Section 2.2.1.23)
  - Note: idealization of bypass factor (i.e., bypass factor = 0) is discussed here.
- Heating coil energy (FC and SZ systems) (Section 2.2.1.24)
- Preheat coil energy (CV and VAV systems) (Section 2.2.1.25)
- Cooling coil energy (All systems) (Section 2.2.1.26)
  - Sensible heat removal (Section 2.2.1.26.1)
  - Latent heat (moisture) removal (Section 2.2.1.26.2)
- Thermal properties of air throughout the system, and for ambient air, including:
  - Humidity ratio (Section 2.2.1.27.1)
  - Specific volume (Section 2.2.1.27.2)
  - Specific heat (Section 2.2.1.27.3)
  - Enthalpy (Section 2.2.1.27.4).

Variable names utilized in these descriptions closely follow those provided in the model spreadsheets. However, some deviations occur and are generally noted in the discussion. In particular, the FC and SZ system models have an “RO” prefix for some of the system variables.

### 2.2.1.1 Supply Air Mass Flow Rate

The supply air mass flow rate (TSMF) is the total of the zone supply air mass flow rate(s). The FC and SZ systems define one zone, and the zone supply air mass flow rate (ROSMF) is used directly as the supply air mass flow rate.

For the CV and VAV systems:

$$\text{TSMF} = \text{ROSMF} + \text{RTSMF} \quad (1)$$

Where

- TSMF = supply air mass flow rate (lb<sub>m,da</sub>/min)
- ROSMF = Zone 1 supply air mass flow rate (lb<sub>m,da</sub>/min), as defined in Section 2.2.1.4
- RTSMF = Zone 2 supply air mass flow rate (lb<sub>m,da</sub>/min), as defined for ROSMF in Section 2.2.1.4.

### 2.2.1.2 Supply Fan Air Temperature Rise

The calculation of the air temperature rise across the supply fan is combined with the calculation of the supply fan heat imparted to the supply air.

$$\text{SFR} = \text{SFH} \times \text{ACCSV} / \text{SASH} \quad (2)$$

Where

- SFR = supply fan air temperature rise caused by supply fan heat (°F)
- SFH = supply fan heat (Btu/ft<sup>3</sup>), as defined in Equation 3
- ACCSV = specific volume of the air entering the supply fan calculated at the supply fan entering air temperature and humidity ratio (ft<sup>3</sup>/lb<sub>m,da</sub>), see Section 2.2.1.27.2 for details
- SASH = specific heat of the air entering the supply fan calculated at the supply fan entering air humidity ratio (Btu/(lb<sub>m,da</sub>·°F)), see Section 2.2.1.27.3 for details.

The supply fan heat is calculated as:

$$\text{SFH} = \text{ASFPR} \times 0.006685 / (\text{DFE} / 100) \quad (3)$$

Where

- SFH = supply fan heat (Btu/ft<sup>3</sup>). This is an intermediate value from within a longer spreadsheet equation; this variable name is not used in the QAS models.
- ASFPR = the actual supply fan pressure rise at the test conditions (in. wg), as defined in Equation 4.
- DFE = design fan efficiency as percentage (0%–100%). This is equivalent to the “fan mechanical efficiency” defined in Part I, Section 1.5.3.1; a constant value of 70% is applied for all systems and cases.
- 0.006685 = conversion factor (Btu/(ft<sup>3</sup>·in. wg)). Derived from Borgnakke and Sonntag (2013), page 756, conversion factors for pressure and specific energy: 0.036126 lb<sub>f</sub>/(in<sup>2</sup>·in. wg) \* 144 in<sup>2</sup>/ft<sup>2</sup> \* 0.00128507 Btu/(lb<sub>f</sub>·ft).



For the FC, SZ, and CV systems, ASFPR is the supply fan pressure rise as specified in the test case and does not vary. For the VAV system, the supply fan pressure rise varies as follows:

$$\text{ASFPR} = (\text{TSMF} \times \text{ACCSV} / \text{NSCFM})^2 \times \text{SFPR} \quad (4)$$

Where

ASFPR = the actual supply fan pressure rise at the test conditions (in. wg), assuming fan pressure rises proportionally to the square of the volumetric airflow rate. This is an intermediate value from within a longer spreadsheet equation; this variable name is not used in the QAS models.

TSMF = supply air mass flow (lb<sub>m,da</sub>/min), as defined in Equation 1.

ACCSV = specific volume of the air entering the supply fan calculated at the supply fan entering air temperature and humidity ratio (ft<sup>3</sup>/lb<sub>m,da</sub>), see Section 2.2.1.27.2 for details.

NSCFM = design system supply airflow rate per the test specification (cfm); FC and SZ models use a variable named ROS.

SFPR = design supply fan pressure rise = 2 (in. wg).

### 2.2.1.3 System Supply Air Temperature (CV and VAV Systems)

The system supply air temperature is:

$$\text{SAT} = \text{TACC} + \text{SFR} \quad (5)$$

Where

SAT = system supply air temperature (°F)

TACC = target after-cooling coil air temperature (°F), as defined in Section 2.2.1.22

SFR = system supply air temperature rise (°F) caused by supply fan heat, as defined by Equation 2.

### 2.2.1.4 Zone Supply Air Mass Flow Rate

For the FC, SZ, and CV systems, the zone supply air mass flow rate is calculated from the specified zone volumetric flow rate and the specific volume of the air entering the supply fan:

$$\text{ROSMF} = \text{ROS} / \text{ACCSV} \quad (6)$$

Where

ROSMF = zone supply air mass flow rate (lb<sub>m,da</sub>/min)

ROS = the nominal zone supply air volume per the test case specification (cfm)

ACCSV = specific volume of the air entering the supply fan calculated at the supply fan entering air temperature and humidity ratio (ft<sup>3</sup>/lb<sub>m,da</sub>), see Section 2.2.1.27.2 for details.

For the VAV system, the zone supply air mass flow rate is:

$$\text{ROSMF} = \text{Maximum} (\text{ROMMF}, \text{ROIMF}) \quad (7)$$

Where

- ROSMF = zone supply air mass flow rate (lb<sub>m,da</sub>/min)
- ROMMF = zone minimum supply air mass flow rate (lb<sub>m,da</sub>/min), as defined in Equation 8
- ROIMF = zone ideal supply air mass flow rate (lb<sub>m,da</sub>/min), as defined in Equation 9.

The minimum zone mass flow rate is:

$$\text{ROMMF} = \text{ROEMF} \quad (8)$$

Where

- ROMMF = minimum zone supply air mass flow rate (lb<sub>m,da</sub>/min)
- ROEMF = zone exhaust air mass flow rate (lb<sub>m,da</sub>/min), as defined in Equation 15.

The ideal zone mass flow rate is:

$$\text{ROIMF} = \text{ROSL} / (60 \times \text{SASH} \times (\text{ROT} - \text{SAT})) \quad (9)$$

Where

- ROIMF = ideal zone supply air mass flow rate (lb<sub>m,da</sub>/min)
- ROSL = room sensible load (Btu/hr), input from test case specification
- 60 = conversion factor (minutes / hour)
- SASH = system supply air specific heat calculated at the supply fan entering air humidity ratio (Btu/(lb<sub>m,da</sub>·°F)), see Section 2.2.1.27.3 for details
- ROT = room set point temperature (°F), input from test case specification
- SAT = system supply air temperature (°F), as defined in Equation 5.

The CV and VAV systems define a second zone with zone supply air mass flow calculated the same as Equation 6 for the CV system and Equations 7, 8, and 9 for the VAV system, except the variables are prefaced with “RT” rather than “RO”.

### 2.2.1.5 Zone Supply Air Temperature

The zone supply air temperature is the supply air temperature required to bring the zone exactly to the zone set point temperature. The zone supply air temperature is calculated as:

$$\text{ROSAT} = \text{ROT} - (\text{ROSL} / (\text{ROSMF} \times 60 \times \text{SASH})) \quad (10)$$

Where

- ROSAT = zone supply air temperature (°F)
- ROT = room set point temperature (°F), input from test case specification
- ROSL = room sensible load (Btu/hr), input from test case specification
- ROSMF = room supply air mass flow (lb<sub>m,da</sub>/min), as defined in Section 2.2.1.4
- 60 = conversion factor (minutes / hour)
- SASH = system supply air specific heat calculated at the supply fan entering air humidity ratio (Btu/(lb<sub>m,da</sub>·°F)), see Section 2.2.1.27.3 for details.

The CV and VAV systems define a second zone with zone supply air temperature calculated the same as Equation 10 except the variables are prefaced with “RT” rather than “RO”.

### 2.2.1.6 Reheat Coil Energy (CV and VAV Systems)

The constant and variable volume systems have zone terminals with reheat coils. If the system supply air temperature (SAT) is below the required zone supply air temperature (ROSAT and RTSAT for Zones 1 and 2, respectively), the reheat coils heat the supply air to exactly the required temperature. The total reheat energy is:

$$QH_{\text{reheat}} = QH1_{\text{reheat}} + QH2_{\text{reheat}} \quad (11)$$

Where

$$\begin{aligned} QH_{\text{reheat}} &= \text{total reheat coil energy (Btu/hr)} \\ QH1_{\text{reheat}} &= \text{Zone 1 reheat coil energy (Btu/hr)} \\ QH2_{\text{reheat}} &= \text{Zone 2 reheat coil energy (Btu/hr)}. \end{aligned}$$

Zone reheat is calculated for each zone as

$$QH1_{\text{reheat}} = (\text{ROSMF} \times 60) \times \text{SASH} \times (\text{ROSAT} - \text{SAT}) \quad (12)$$

Where

$$\begin{aligned} QH1_{\text{reheat}} &= \text{Zone 1 reheat coil energy (Btu/hr)} \\ \text{ROSMF} &= \text{room supply air mass flow (lb}_{\text{m,da}}/\text{min}), \text{ as defined in Section 2.2.1.4} \\ 60 &= \text{conversion factor (minutes/hour)} \\ \text{SASH} &= \text{system supply air specific heat calculated at the supply fan entering air humidity ratio (Btu/(lb}_{\text{m,da}} \cdot ^\circ\text{F)}), \text{ see Section 2.2.1.27.3 for details} \\ \text{ROSAT} &= \text{required zone supply air temperature (}^\circ\text{F)}, \text{ as defined in Equation 10} \\ \text{SAT} &= \text{system supply air temperature (}^\circ\text{F)}, \text{ as defined in Equation 5.} \end{aligned}$$

The CV and VAV systems define a second zone with reheat energy ( $QH2_{\text{reheat}}$ ) calculated the same as Equation 12, except the variables are prefaced with “RT” rather than “RO”.

### 2.2.1.7 Zone Air Temperature

There is no calculation of zone air temperature. The zone air temperature is always exactly at the zone set point temperature by definition. The zone supply air temperature is calculated (see Equation 10) such that it offsets zone loads so the air leaves the zone at the zone air temperature set point.

### 2.2.1.8 Zone Humidity Ratio and Zone Latent Heat (Moisture) Gains

The zone humidity ratio calculation assumes the zone has no air volume capacitance, sensible heat capacitance, or moisture capacitance. This is the same as assuming the zone and materials have reached steady-state equilibrium conditions. Latent gains (specified as energy over unit time) are converted to a mass of water vapor using the enthalpy of saturated vapor as calculated by *ASHRAE Handbook of Fundamentals*, Chapter 1, Equation 31 (ASHRAE 2009), with more precise value (1061.15 Btu/lb<sub>m,w</sub>) applied here.

The zone humidity ratio is:

$$\text{ROHR} = \text{SAHR} + \text{GRO} / \text{ROSMF} \quad (13)$$

Where

- ROHR = zone humidity ratio (grains<sub>w</sub>/lb<sub>m,da</sub>)
- SAHR = supply air humidity ratio (grains<sub>w</sub>/lb<sub>m,da</sub>), as defined in Equation 48
- GRO = zone moisture gains (grains<sub>w</sub>/min), as defined by Equation 14
- ROSMF = zone supply air mass flow (lb<sub>m,da</sub>/min), as defined in Section 2.2.1.4.

The zone moisture gains are:

$$\text{GRO} = (\text{ROLL} / 60) / (1061.15 + 0.444 \times \text{ROT}) \times 7000 \quad (14)$$

Where

- GRO = zone moisture gains (grains<sub>w</sub>/min)
- ROLL = zone latent load (Btu/hour), input from test case specification
- 60 = conversion factor (minutes/hour)
- 1061.15 = specific enthalpy of saturated vapor at 0°F (Btu/lb<sub>m,w</sub>)
- 0.444 = specific heat of saturated vapor (Btu/lb<sub>m,w</sub>·°F)
- ROT = zone set point temperature (°F), input from test case specification
- 7000 = conversion (grains/lb<sub>m</sub>).

The CV and VAV systems define a second zone with values for ROHR and GRO calculated per Equations 13 and 14 except the variables are prefaced with “RT” rather than “RO”.

### 2.2.1.9 Zone Exhaust Air Mass Flow Rate

The zone exhaust fan mass flow rate is calculated as:

$$\text{ROEMF} = \text{ROE} / \text{ROSV} \quad (15)$$

Where

- ROEMF = zone exhaust air mass flow (lb<sub>m,da</sub>/min).
- ROE = zone exhaust airflow rate (cfm), input from test case specification.
- ROSV = zone air specific volume using zone air temperature and humidity ratio (ft<sup>3</sup>/lb<sub>m,da</sub>), see Section 2.2.1.27.2 for details.

The CV and VAV systems define a second zone with zone exhaust air flow calculated the same as Equation 15, except the variables are prefaced with “RT” rather than “RO”.

### 2.2.1.10 Return Air Mass Flow Rate

The return air mass flow rate (TRMF) is the sum of the zone return air mass flow rate(s). For all systems, the return air mass flow is determined by the system mass balance and is not volumetrically constant. The return air volumetric flow varies as required to move the required return air mass flow rate at the specific volume of the air entering the return fan. The FC and SZ system models define one zone and the Zone 1 return air mass flow rate is used directly as the return air mass flow rate.

$$\text{TRMF} = \text{RORMF} + \text{RTRMF} \quad (16)$$

Where

- TRMF = return air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ). This variable is not calculated for the FC and SZ systems, where  $\text{TRMF} = \text{RORMF}$  and RORMF is used directly.
- RORMF = Zone 1 return air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Equation 17.
- RTRMF = Zone 2 return air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Equation 17 with variables prefaced with “RT” rather than “RO” (CV and VAV systems only).

The zone return mass flow rate ( $\text{lb}_{\text{m,da}}/\text{min}$ ) for zone one is:

$$\text{RORMF} = \text{ROSMF} - \text{ROEMF} \quad (17)$$

Where

- RORMF = zone return air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ )
- ROSMF = zone supply air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Section 2.2.1.4
- ROEMF = zone exhaust air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Equation 15.

The CV and VAV systems define a second zone with zone return mass flow rate calculated the same as RORMF except variable are prefaced with “RT” rather than “RO”.

### 2.2.1.11 Return Fan Inlet Air Temperature

This quantity is not calculated for the FC system. For the SZ system, the return fan inlet temperature is:

$$\text{RFIT} = \text{ROT} + \text{RDHG} \quad (18)$$

Where

- RFIT = return fan inlet air temperature ( $^{\circ}\text{F}$ )
- ROT = Zone 1 set point temperature ( $^{\circ}\text{F}$ ), input from test case specification
- RDHG = return duct heat gain ( $^{\circ}\text{F}$ ); not utilized (= 0) in test case specification.

For the CV and VAV systems the return fan inlet temperature is:

$$\text{RFIT} = (\text{RORMF} \times \text{ROT} \times \text{ROSH} + \text{RTRMF} \times \text{RTT} \times \text{RTSH}) / (\text{TRMF} \times \text{RASH}) + \text{RDHG} \quad (19)$$

Where

- RFIT = return fan inlet air temperature ( $^{\circ}\text{F}$ )
- RORMF = Zone 1 return air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Equation 17
- ROT = Zone 1 set point temperature ( $^{\circ}\text{F}$ ), input from test case specification
- ROSH = specific heat of the air in Zone 1 calculated at the Zone 1 air temperature and humidity ratio ( $\text{Btu}/(\text{lb}_{\text{m,da}} \cdot ^{\circ}\text{F})$ ), see Section 2.2.1.27.3 for details
- RTRMF = Zone 2 return air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Equation 17 with variables prefaced with “RT” rather than “RO”
- RTT = Zone 2 set point temperature ( $^{\circ}\text{F}$ ), input from test case specification

- RTSH = specific heat of the air in Zone 2 calculated at the Zone 2 air temperature and humidity ratio (Btu/(lb<sub>m,da</sub>·°F)), see Section 2.2.1.27.3 for details
- TRMF = return air mass flow (lb<sub>m,da</sub>/min), as defined in Equation 16
- RASH = specific heat of the air entering the return fan calculated at the return fan inlet air temperature humidity ratio (Btu/(lb<sub>m,da</sub>·°F)), see Section 2.2.1.27.3 for details
- RDHG = return duct heat gain (°F); not utilized (= 0) in test case specification.

### 2.2.1.12 Return Air Humidity Ratio

For the FC and SZ systems the return air humidity ratio is:

$$RAHR = ROHR \quad (20)$$

Where

- RAHR = return air mass flow (grains<sub>w</sub>/lb<sub>m,da</sub>)
- ROHR = Zone 1 humidity ratio (grains<sub>w</sub>/lb<sub>m,da</sub>).

For the CV and VAV systems the return air humidity ratio is:

$$RAHR = (ROHR \times RORMF + RTHR \times RTRMF) / TRMF \quad (21)$$

Where

- RAHR = return air humidity ratio (grains<sub>w</sub>/lb<sub>m,da</sub>)
- ROHR = Zone 1 humidity ratio (grains<sub>w</sub>/lb<sub>m,da</sub>), as defined in Section 2.2.1.8
- RORMF = Zone 1 return air mass flow (lb<sub>m,da</sub>/min), as defined in Equation 17
- RTHR = Zone 2 humidity ratio (grains<sub>w</sub>/lb<sub>m,da</sub>), as defined by ROHR in Section 2.2.1.8
- RTRMF = Zone 2 return air mass flow (lb<sub>m,da</sub>/min), as defined in Equation 17 with variables prefaced with “RT” rather than “RO”
- TRMF = return air mass flow (lb<sub>m,da</sub>/min), as defined in Equation 16.

### 2.2.1.13 Return Fan Air Temperature Rise

The SZ, CV, and VAV systems have a return fan. For all three systems, the QAS models treat the return fan as a variable volume device with the volumetric flow varying as required to move the required return air mass flow rate at the specific volume of the air entering the return fan. The QAS models combine the calculation of the temperature rise of the return air with the calculation of the return fan heat imparted to the return air.

$$RFR = RFH \times RFISV / RASH \quad (22)$$

Where

- RFR = return fan air temperature rise caused by supply fan heat (°F)
- RFH = return fan heat (Btu/ft<sup>3</sup>), as defined in Equation 23
- RFISV = specific volume of the air entering the return fan calculated at the return fan entering air temperature and humidity ratio (ft<sup>3</sup>/lb<sub>m,da</sub>), see Section 2.2.1.27.2 for details
- RASH = specific heat of the air entering the return fan calculated at the return fan entering air humidity ratio (Btu/(lb<sub>m,da</sub>·°F)), see Section 2.2.1.27.3 for details.

The return fan heat is calculated as:

$$\text{RFH} = \text{ARFPR} \times 0.006685 / (\text{DFE} / 100) \quad (23)$$

Where

- RFH = return fan heat (Btu/ft<sup>3</sup>). This is an intermediate value from within a longer spreadsheet equation; this variable name not used in the QAS models.
- ARFPR = actual return fan pressure rise at test conditions (in. wg), as defined in Equation 24.
- DFE = design fan efficiency as percentage (0-100%). This is equivalent to the “fan mechanical efficiency” defined in Part I, Section 1.5.3.1; a constant value of 70% is applied for all systems and cases.
- 0.006685 = conversion factor (Btu/(ft<sup>3</sup>·in. wg)). See Section 2.2.1.2.

And

$$\text{ARFPR} = (\text{TRMF} \times \text{RFISV} / \text{NRCFM})^2 \times \text{RFPR} \quad (24)$$

Where

- ARFPR = the actual return fan pressure rise at the test conditions (in. wg), based on that fan pressure rises proportionally to the square of the volumetric airflow rate. This is an intermediate value from within a longer spreadsheet equation; this variable name is not used in the QAS models.
- TRMF = return air mass flow (lb<sub>m,da</sub>/min), as defined in Equation 16. The SZ system uses RORMF instead, as defined in Equation 17.
- RFISV = specific volume of the air entering the return fan calculated at the return fan entering air temperature and humidity ratio (ft<sup>3</sup>/lb<sub>m,da</sub>), see Section 2.2.1.27.2 for details.
- NRCFM = design system return airflow rate per the test specification (cfm).
- RFPR = design return fan pressure rise = 1 (in. wg).

#### 2.2.1.14 Return Air Temperature

For the FC system the return air temperature is:

$$\text{RAT} = \text{ROT} \quad (25)$$

Where

- RAT = return air temperature (°F)
- ROT = Zone 1 set point temperature (°F), input from test case specification.

For the SZ, CV, and VAV systems the return air temperature is:

$$\text{RAT} = \text{RFIT} + \text{RFR} \quad (26)$$

Where

- RAT = return air temperature (°F)
- RFIT = return fan inlet temperature (°F), as defined in Section 2.2.1.11
- RFR = return fan air temperature rise (°F/lb<sub>m,da</sub>), as defined in Equation 22.

### 2.2.1.15 Relief Air Mass Flow Rate

This quantity is not explicitly calculated. The mixed air condition calculations assume the relief air mass flow is:

$$\text{ReliefMF} = \text{TRMF} - \text{RCMF} \quad (27)$$

Where

ReliefMF = relief air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ). This variable is not used or calculated by the QAS models.

TRMF = return air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Equation 16. The FC and SZ systems use RORMF as defined in Equation 17 instead of TRMF.

RCMF = recirculated mass flow of air ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Equation 28.

### 2.2.1.16 Recirculated Air Mass Flow Rate

The recirculated air mass flow is:

$$\text{RCMF} = \text{TSMF} - \text{OAMF} \quad (28)$$

Where

RCMF = recirculated air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ). This variable is not calculated for the FC system, where  $\text{RCMF} = \text{RORMF}$ , and RORMF (see Equation 17) is used directly.

TSMF = supply air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Equation 1.

OAMF = outdoor air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Section 2.2.1.18.

### 2.2.1.17 Outdoor Air Humidity Ratio

The atmospheric humidity ratio is calculated from the specified outdoor dry-bulb and dew-point temperatures for each test case with Equations 5, 6, and 22 from *2009 ASHRAE Handbook of Fundamentals*, Chapter 1 (ASHRAE 2009). The outdoor humidity ratio in  $\text{grains}_w/\text{lb}_{\text{m,da}}$  is calculated as:

$$\text{OAHR} = 7000 \times W \quad (29)$$

Where

OAHR = humidity ratio ( $\text{grains}_w/\text{lb}_{\text{m,da}}$ ) for outdoor air

W = humidity ratio ( $\text{lb}_{\text{m,w}}/\text{lb}_{\text{m,da}}$ ) for outdoor air

7000 = conversion factor ( $\text{grains}/\text{lb}_m$ ).

The humidity ratio is calculated as:

$$W = 0.621945 \times p_w / (p - p_w) \quad (30)$$

Where

W = humidity ratio ( $\text{lb}_{\text{m,w}}/\text{lb}_{\text{m,da}}$ ) for outdoor air



0.621945 = ratio of molecular masses, water to dry air  
 $p_w$  = water vapor pressure (psia)  
 $p$  = atmospheric pressure = 14.695975 psia.

The water vapor pressure is calculated as:

$$p_w = p_{ws}(T_{dp}) \quad (31)$$

Where

$p_w$  = water vapor pressure (psia)  
 $p_{ws}(T_{dp})$  = saturation water vapor pressure (psia) at dewpoint temperature.

The water vapor saturation pressure at the dew-point temperature ( $p_{ws}(T_{dp})$ ) is calculated as:

$$p_{ws}(T_{dp}) = e^{LNPWS} \quad (32)$$

For temperatures from -148°F to 32°F:

$$LNPWS = C_1 / T_{dp} + C_2 + C_3 \times T_{dp} + C_4 \times T_{dp}^2 + C_5 \times T_{dp}^3 + C_6 \times T_{dp}^4 + C_7 \times \ln T_{dp} \quad (33)$$

Where

$LNPWS$  = natural log of the water vapor saturation pressure at temperature  $T_{dp}$   
 $T_{dp}$  = absolute dew-point temperature of outdoor air (OADP) in °R  
 $C_1$  = -1.0214165 E+04  
 $C_2$  = -4.8932428 E+00  
 $C_3$  = -5.3765794 E-03  
 $C_4$  = 1.9202377 E-07  
 $C_5$  = 3.5575832 E-10  
 $C_6$  = -9.0344688 E-14  
 $C_7$  = 4.1635019 E+00.

For temperatures over liquid water from 32°F to 392°F:

$$LNPWS = C_8 / T_{dp} + C_9 + C_{10} \times T_{dp} + C_{11} \times T_{dp}^2 + C_{12} \times T_{dp}^3 + C_{13} \times \ln T_{dp} \quad (34)$$

Where

$LNPWS$  = natural log of the water vapor saturation pressure at temperature  $T_{dp}$   
 $T_{dp}$  = absolute dew-point temperature of outdoor air (OADP) in °R  
 $C_8$  = -1.0440397 E+04  
 $C_9$  = -1.1294650 E+01  
 $C_{10}$  = -2.7022355 E-02  
 $C_{11}$  = 1.2890360 E-05  
 $C_{12}$  = -2.4780681 E-09  
 $C_{13}$  = 6.5459673 E+00.

The equations above calculate a humidity ratio from the specified test case dry-bulb and dew-point temperature that differs from the test case specified equivalent humidity ratio. The equations rely upon perfect gas laws while the equivalent moisture parameters provided in the test case definitions

specified in Part I are calculated based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011). For given dry-bulb and dew-point temperatures, this real gas correlation predicts a humidity ratio that differs from that predicted by the perfect gas laws used in the above equations. Table 2-1 compares the QAS model predicted outdoor air humidity ratios with those published as equivalent moisture parameters in the test specification. The differences are an indicator of the impact of using the perfect gas equations rather than a real gas correlation. To explore the impact of these differences on predicted heating and cooling coil loads, the specification ambient humidity ratio was input directly in to the QAS models. Table 2-2 presents the resulting coil loads compared to those predicted by the QAS models using dew-point temperature and the above equations. The impact on sensible coil loads is small (< 0.04%) while the impact on latent coil load is larger (< 3.1%).

**Table 2-1. Outdoor Air Humidity Ratio (kg<sub>w</sub>/kg<sub>da</sub>): QAS predicted vs. Specified**

Case	Specification	QAS Model	Diff.
AE201	0.000260	0.0002589	-0.41%
AE203	0.002948	0.002936	-0.41%
AE204	0.016849	0.016774	-0.45%
AE205	0.004510	0.004491	-0.42%
AE206	0.015625	0.015556	-0.44%

**Table 2-2. Predicted Coil Loads from QAS OAW vs. Specified OAW**

Case	Specification OAW (kWh)			QAS Predicted OAW (kWh)			Relative Difference (%)		
	Sensible	Latent	Total	Sensible	Latent	Total	Sensible	Latent	Total
AE201	-8.360	0.000	-8.360	-8.360	0.000	-8.360	0.00%		0.00%
AE203	0.856	0.000	0.856	0.856	0.000	0.856	0.00%		0.00%
AE204	3.574	2.075	5.649	3.574	2.054	5.628	-0.01%	-0.98%	-0.37%
AE205	1.916	0.000	1.916	1.916	0.000	1.916	0.00%		0.00%
AE206	1.706	0.989	2.696	1.706	0.970	2.677	-0.01%	-1.89%	-0.70%
AE226	1.562	1.857	3.419	1.562	1.800	3.362	-0.03%	-3.07%	-1.68%
AE245	2.208	0.000	2.208	2.208	0.000	2.208	0.00%		0.00%

### 2.2.1.18 Outdoor Air Mass Flow Rate and Economizer Operation

The outdoor air mass flow in the FC system is:

$$OAMF = ROEMF \quad (35)$$

Where

OAMF = outdoor air mass flow (lb<sub>m,da</sub>/min). This variable is not calculated. ROEMF is used directly.

ROEMF = zone exhaust air mass flow (lb<sub>m,da</sub>/min), as defined in Equation 15.

The outdoor air mass flow in the SZ, CV, and VAV systems is calculated similarly but is modified by the economizer control logic.

When the economizer does not operate, the outdoor air mass flow is:

$$\text{OAMF} = \text{OAMF}_{\min} \quad (36)$$

Where

OAMF = outdoor air mass flow (lb<sub>m,da</sub>/min)  
 OAMF<sub>min</sub> = minimum outdoor air mass flow (lb<sub>m,da</sub>/min).

And the minimum outdoor air mass flow is:

$$\text{OAMF}_{\min} = \text{ROEMF} + \text{RTEMF} \quad (37)$$

Where

ROEMF = Zone 1 exhaust air mass flow (lb<sub>m,da</sub>/min), as defined in Equation 15  
 RTEMF = Zone 2 exhaust air mass flow (lb<sub>m,da</sub>/min), as defined for ROEMF in Equation 15;  
 this term present only for CV and VAV systems.

When the economizer operates at 100% the outdoor air mass flow is:

$$\text{OAMF} = \text{TSMF} \quad (38)$$

Where

OAMF = outdoor air mass flow (lb<sub>m,da</sub>/min)  
 TSMF = system supply air mass flow (lb<sub>m,da</sub>/min), as defined in Equation 1.

When the economizer operates at less than 100% outdoor air then

$$\text{OAMF} = ((\text{TSMF} \times \text{MASH} \times \text{ECOT}) - (\text{TSMF} \times \text{RASH} \times \text{RAT})) / (\text{OADB} \times \text{OASH} - \text{RAT} \times \text{RASH}) \quad (39)$$

Where

MASH = mixed air specific heat calculated at the mixed air humidity ratio (Btu/(lb<sub>m,da</sub>·°F)),  
 see Section 2.2.1.27.3 for details  
 ECOT = economizer mixed air temperature setpoint (°F), as defined in Equation 40  
 RASH = specific heat of the air entering the return fan calculated at the return fan entering air  
 humidity ratio (Btu/(lb<sub>m,da</sub>·°F)), see Section 2.2.1.27.3 for details  
 RAT = return air temperature (°F), as defined in Equation 26  
 OADB = outdoor air temperature (°F), as defined in test specification input  
 OASH = outdoor air specific heat calculated at the outdoor air humidity ratio (Btu/(lb<sub>m,da</sub>·°F)),  
 see Section 2.2.1.27.3 for details.

**Note:** Operation at less than 100% outdoor air does not occur for the current set of economizer test cases.

### 2.2.1.18.1 Economizer Control Logic

The economizer operates only when economizer operation is selected,  $MAT_{min} > ECOT$ , and:

- Dry-bulb temperature economizer operation is selected and  $OADB < RAT$ , or
- Enthalpy economizer operation is selected and  $OAE < RAE$ .

Where

OADB	= outdoor air dry-bulb temperature (°F), input from test case specification.
RAT	= return air temperature (°F), as defined in Equation 26.
OAE	= outdoor air enthalpy (Btu/lb <sub>m,da</sub> ), see Section 2.2.1.27.4 for details.
RAE	= return air enthalpy (Btu/lb <sub>m,da</sub> ), see Section 2.2.1.27.4 for details.
$MAT_{min}$	= the mixed air temperature calculated by Equation 42 assuming the outdoor air mass flow is equal to $OAMF_{min}$ (see Equation 37). $MAT_{min}$ is an intermediate value from within a longer spreadsheet equation; this variable name is not used in the QAS models.
ECOT	= economizer mixed air temperature set point (°F), as defined in Equation 40.

If operating, the outdoor air flow is 100% outdoor air when:

$$OADB \geq ECOT, \text{ i.e., } RAT > OADB \geq ECOT$$

If operating, the outdoor air flow is between  $OAMF_{min}$  and 100% when:

$$OADB < ECOT, \text{ i.e., } RAT > OADB < ECOT$$

**Note:**  $OADB < ECOT$  does not occur for the current set of economizer test cases.

The economizer set point is:

$$ECOT = SASP - SFR \tag{40}$$

Where

ECOT	= economizer mixed air temperature setpoint (°F). ECOT differs from TACC (Section 2.2.1.22) only in the CV and VAV systems when $MAT_{min} < ECOT$ . For those conditions, supply air temperature falls below the economizer set point because the preheat coil only operates to maintain a preheat coil exiting air temperature of 45°F.
SASP	= system supply air temperature set point (°F), input from test case specification.
SFR	= system supply air temperature rise caused by supply fan heat (°F), as defined in Equation 2.

### 2.2.1.19 Mixed Air Mass Flow Rate

$$MAMF = TSMF \tag{41}$$

Where

MAMF	= mixed air mass flow (lb <sub>m,da</sub> /min). This variable is not calculated in the QAS models; TSMF is used directly.
------	--

TSMF = system supply air mass flow (lb<sub>m,da</sub>/min), as defined in Equation 1.

### 2.2.1.20 Mixed Air Temperature

The mixed air temperature in all systems is calculated as:

$$\text{MAT} = (\text{RCMF} \times \text{RAT} \times \text{RASH} + \text{OAMF} \times \text{OADB} \times \text{OASH}) / (\text{TSMF} \times \text{MASH}) \quad (42)$$

Where

MAT	= mixed air temperature (°F)
RCMF	= recirculated air mass flow (lb <sub>m,da</sub> /min), as defined in Equation 28
RAT	= return air temperature (°F), as defined in Section 2.2.1.14
RASH	= specific heat of the air entering the return fan calculated at the return fan entering air humidity ratio (Btu/(lb <sub>m,da</sub> ·°F)), see Section 2.2.1.27.3 for details
OAMF	= outdoor air air mass flow (lb <sub>m,da</sub> /min), as defined in Section 2.2.1.18
OADB	= outdoor air temperature (°F), as defined in test specification input
OASH	= outdoor air specific heat calculated at the outdoor air humidity ratio (Btu/(lb <sub>m,da</sub> ·°F)), see Section 2.2.1.27.3 for details
TSMF	= supply air mass flow (lb <sub>m,da</sub> /min), as defined in Equation 1
MASH	= mixed air specific heat calculated at the mixed air humidity ratio (Btu/(lb <sub>m,da</sub> ·°F)), see Section 2.2.1.27.3 for details.

### 2.2.1.21 Mixed Air Humidity Ratio

The mixed air humidity ratio (MAHR) in all systems is calculated as:

$$\text{MAHR} = (\text{RCMF} \times \text{RAHR} + \text{OAMF} \times \text{OAHR}) / (\text{TSMF}) \quad (43)$$

Where

MAHR	= mixed air humidity ratio (grains <sub>w</sub> /lb <sub>m,da</sub> )
RCMF	= recirculated air mass flow (lb <sub>m,da</sub> /min), as defined in Equation 28
RAHR	= humidity ratio of the return air (grains <sub>w</sub> /lb <sub>m,da</sub> ), as defined in Section 2.2.1.12
OAMF	= outdoor air air mass flow (lb <sub>m,da</sub> /min), as defined in Section 2.2.1.18
OAHR	= outdoor air humidity ratio (grains <sub>w</sub> /lb <sub>m,da</sub> ), see Section 2.2.1.17 for details
TSMF	= supply air mass flow (lb <sub>m,da</sub> /min), as defined in Equation 1.

### 2.2.1.22 Target After-Coil Air Temperatures

The target after-heating-coil (or preheat coil) and after-cooling-coil air temperatures are the temperatures of the air leaving the respective coil. The target air temperatures control coil operation and are the basis for calculating coil energy use. In heating mode, the target air temperature after the heating coil (TAHC) will be greater than the mixed air temperature (MAT), and the target air temperature after the cooling coil (TACC) will be the same as the TAHC. In cooling mode, the TAHC will equal the MAT and the TACC will be less than the MAT. The supply fan is located after the coils and the target after-coil air temperatures anticipate the added supply fan heat.

For the FC and SZ systems the target after-coil air temperatures have an “RO” prefix and are calculated as:

$$\text{TAHC} = \text{Maximum} ((\text{ROSAT} - \text{SFR}), \text{MAT}) \quad (44)$$

$$\text{TACC} = \text{Minimum} ((\text{ROSAT} - \text{SFR}), \text{TAHC}) \quad (45)$$

Where

- TAHC = target air temperature after the heating coil, before the cooling coil (°F).
- TACC = target air temperature after the cooling coil, before the supply fan (°F). For FC and SZ systems, the effect of Equations 44 and 45 are that  $\text{TACC} = \text{ROSAT} - \text{SFR}$ .
- MAT = mixed air temperature (°F), as defined in Equation 42.
- ROSAT = zone supply air temperature (°F), as defined in Equation 10.
- SFR = supply air temperature rise (°F) caused by supply fan heat, as defined in Equation 2.

For the CV and VAV system, the target after-coil air temperatures are calculated as:

$$\text{TAHC} = \text{Maximum} (\text{PHSP}, \text{MAT}) \quad (46)$$

$$\text{TACC} = \text{Minimum} ((\text{SASP} - \text{SFR}), \text{TAHC}) \quad (47)$$

Where

- TAHC = target air temperature after the heating coil, before the cooling coil (°F)
- TACC = target air temperature after the cooling coil, before the supply fan (°F)
- PHSP = preheat coil set point (°F), input from test case specification (= 45°F, constant value for the test cases)
- MAT = mixed air temperature (°F), as defined in Equation 42
- SASP = system supply air temperature set point (°F), input from test case specification
- SFR = system supply air temperature rise caused by supply fan heat (°F), as defined in Equation 2.

### 2.2.1.23 Cooling Coil Leaving Air Humidity Ratio

No cooling coil bypass assumed (bypass factor = 0). The cooling coil leaving humidity ratio is:

$$\text{SAHR} = \text{Minimum} (\text{MAHR}, \text{SSPHR}) \quad (48)$$

Where

- SAHR = cooling coil leaving air (supply air) humidity ratio (grains<sub>w</sub>/lb<sub>m,da</sub>)
- MAHR = mixed air humidity ratio (grains<sub>w</sub>/lb<sub>m,da</sub>), as defined in Equation 43
- SSPHR = saturation humidity ratio (100% saturated air) of the cooling coil leaving air (grains<sub>w</sub>/lb<sub>m,da</sub>), as defined in Equation 49.

The saturation humidity ratio (100% saturated air) of the cooling coil leaving air is then calculated as:

$$\text{SSPHR} = 0.621945 \times (p_{ws} / (14.695975 - p_{ws})) \times 7000 \quad (49)$$

Where

- SSPHR = saturation humidity ratio of the cooling coil leaving air (grains<sub>w</sub>/lb<sub>m,da</sub>)
- 0.621945 = ratio of molecular masses, water to dry air
- $p_{ws}$  = saturation water vapor pressure (psia), see Equation 50

14.695975 = atmospheric pressure (psi)  
 7000 = conversion factor (grains/lbm).

Equation 6 from *ASHRAE Handbook of Fundamentals* (2009), Chapter 1 is used to calculate the saturation water vapor pressure ( $p_{ws}$ ):

$$\ln(p_{ws}) = C_8/T + C_9 + C_{10} \times T + C_{11} \times T^2 + C_{12} \times T^3 + C_{13} \times \ln(T) \quad (50)$$

Where

$p_{ws}$  = water vapor saturation pressure (psia)  
 T = cooling coil leaving air absolute temperature (TACC) in °R  
 $C_8, C_9, C_{10}, C_{11}, C_{12},$  and  $C_{13}$  are the same as in Equation 34.

### 2.2.1.24 Heating Coil Energy (FC and SZ Systems)

The heating coil is located in the mixed air stream before the cooling coil and fan. It energizes any time the mixed air temperature is below the target after-heating coil air temperature ( $ROMAT < ROTAHC$ ). The heating coil energy is calculated as:

$$QH = (ROSMF \times 60) \times MASH \times (TAHC - MAT) \quad (51)$$

Where

QH = heating coil energy (Btu/hr)  
 ROSMF = supply air mass flow ( $lb_{m,da}/min$ ), as defined in Section 2.2.1.4  
 60 = conversion factor (minutes/hour)  
 MASH = mixed air specific heat calculated at the mixed air humidity ratio ( $Btu/(lb_{m,da} \cdot ^\circ F)$ ), see Section 2.2.1.27.3 for details  
 TAHC = target air temperature after the heating coil ( $^\circ F$ ), as defined in Equation 44  
 MAT = mixed air temperature ( $^\circ F$ ), as defined in Equation 42.

### 2.2.1.25 Preheat Coil Energy (VAV and CV Systems)

The preheat coil is located in the mixed air stream before the heating coil and supply fan. It energizes when the mixed air temperature is below the target after-heating-coil air temperature ( $MAT < TAHC$ ) and warms the mixed air exactly to the target after-heating-coil air temperature.

$$QH_{preheat} = (TSMF \times 60) \times MASH \times (TAHC - MAT) \quad (52)$$

Where

$QH_{preheat}$  = preheat coil energy (Btu/hr)  
 TSMF = supply air mass flow ( $lb_{m,da}/min$ ), as defined in Equation 1  
 60 = conversion factor (minutes/hour)  
 MASH = mixed air specific heat calculated at the mixed air humidity ratio ( $Btu/(lb_{m,da} \cdot ^\circ F)$ ), see Section 2.2.1.27.3 for details  
 TAHC = the target after-heating-coil air temperature ( $^\circ F$ ), as defined in Equation 46  
 MAT = mixed air temperature ( $^\circ F$ ), as defined in Equation 42.

### 2.2.1.26 Cooling Coil Energy (All Systems)

The cooling coil is located in the mixed air stream after the heating coil and before the supply fan. It energizes any time air temperature entering the coil is above the target after-cooling-coil air temperature. The variable names in the description and equations below are from the CV and VAV systems. The FC and SZ systems use the same variable names, but with an “RO” prefix.

The total energy to cool the supply air mass is the energy to cool the entering moist air mass from the coil entering air temperature (MAT or TAHC) to the target after-cooling-coil air temperature (TACC). If the cooling coil leaving air humidity ratio (SAHR) is below its entering air humidity ratio (MAHR), then condensation has occurred. The mass of vapor condensed is calculated based on the difference of the entering and leaving humidity ratios. When there is condensation, the latent cooling portion of the total cooling load is calculated as the energy to condense water vapor at the coil entering air dew-point temperature. All other cooling energy is sensible. A general discussion regarding classification of coil loads is provided in *2012 ASHRAE Handbook, HVAC Systems and Equipment*, pp. 23.14, 23.15, “Determining Refrigeration Load” (ASHRAE 2012).

#### 2.2.1.26.1 Sensible Heat Removal

The cooling coil sensible energy ( $QC_{\text{sensible}}$ ) is:

$$QC_{\text{sensible}} = QCS_{\text{air}} + QCS_{\text{vapor}} + QCS_{\text{water}} \quad (53)$$

Where

- $QCS_{\text{air}}$  = energy to cool the moist air leaving the cooling coil from the coil entering air temperature to the target after-cooling-coil air temperature (Btu/hr), as defined in Equation 54.
- $QCS_{\text{vapor}}$  = energy to cool any condensing vapor from the coil entering temperature to the coil entering air dew-point temperature (Btu/hr), as defined in Equation 55.
- $QCS_{\text{water}}$  = energy to cool any condensed moisture from the coil entering air dew-point to the target after-cooling-coil air temperature. (Btu/hr), as defined in Equation 56. The assumption that condensed moisture leaves the system at the cooling coil leaving air temperature is based on PSU’s model (Yuill and Haberl 2002, Yuill 2010). Other coil models may have a different leaving condensate temperature and/or assume moisture condenses at a different temperature.

Energy to cool the air leaving the coil (dry air and uncondensed vapor) is calculated as:

$$QCS_{\text{air}} = TSMF \times 60 \times SASH \times (TAHC - TACC) \quad (54)$$

Where

- $QCS_{\text{air}}$  = energy to cool air leaving the cooling coil (Btu/hr)
- TSMF = total system supply air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Equation 1
- 60 = conversion factor (minutes/hour)
- SASH = specific heat of the air leaving the cooling coil ( $\text{Btu}/(\text{lb}_{\text{m,da}} \cdot ^\circ\text{F})$ ), see Section 2.2.1.27.3 for details
- TAHC = the target after-heating-coil air temperature ( $^\circ\text{F}$ ), as defined in Section 2.2.1.22
- TACC = the target after-cooling-coil air temperature ( $^\circ\text{F}$ ), as defined in Section 2.2.1.22.



Energy to cool condensing vapor before it condenses is:

$$QCS_{\text{vapor}} = \text{TSMF} \times 60 \times ((\text{MAHR} - \text{SAHR}) / 7000 \times 0.444 \times (\text{TAHC} - \text{DPAHC})) \quad (55)$$

Where

$QCS_{\text{vapor}}$	= energy to cool condensing vapor before it condenses (Btu/hr)
TSMF	= total system supply air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Equation 1
60	= conversion factor (minutes/hour)
MAHR	= mixed air humidity ratio ( $\text{grains}_w/\text{lb}_{\text{m,da}}$ ), as defined in Equation 43
SAHR	= system supply air humidity ratio ( $\text{grains}_w/\text{lb}_{\text{m,da}}$ ), as defined in Equation 48
TAHC	= target after-heating-coil air temperature ( $^{\circ}\text{F}$ ), as defined in Section 2.2.1.22
DPAHC	= dew-point temperature of the air entering the cooling coil ( $^{\circ}\text{F}$ )
0.444	= specific heat of water vapor ( $\text{Btu}/(\text{lb}_{\text{m,w}} \cdot ^{\circ}\text{F})$ )
7000	= conversion factor ( $\text{grains}/\text{lb}_m$ ).

Energy to cool the condensed moisture is:

$$QCS_{\text{water}} = \text{TSMF} \times 60 \times ((\text{MAHR} - \text{SAHR}) / 7000 \times 1 \times (\text{DPAHC} - \text{TACC})) \quad (56)$$

Where

$QCS_{\text{water}}$	= energy to cool condensed vapor (Btu/hr)
TSMF	= total system supply air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Equation 1
60	= conversion factor (minutes/hour)
MAHR	= mixed air humidity ratio ( $\text{grains}_w/\text{lb}_{\text{m,da}}$ ), as defined in Equation 43
SAHR	= system supply air humidity ratio ( $\text{grains}_w/\text{lb}_{\text{m,da}}$ ), as defined in Equation 48
7000	= conversion factor ( $\text{grains}/\text{lb}_m$ )
1	= specific heat of liquid water ( $\text{Btu}/(\text{lb}_{\text{m,w}} \cdot ^{\circ}\text{F})$ )
DPAHC	= dew-point temperature of the air entering the cooling coil ( $^{\circ}\text{F}$ )
TACC	= target after-cooling-coil air temperature ( $^{\circ}\text{F}$ ), as defined in Section 2.2.1.22.

### 2.2.1.26.2 Latent Heat (Moisture) Removal

The cooling coil latent load is calculated as the heat of evaporation at the coil entering air dew-point temperature multiplied by the mass of condensed moisture.

$$QC_{\text{latent}} = \text{TSMF} \times 60 \times (\text{MAHR} - \text{SAHR}) / 7000 \times (1075.21 - 0.556 \times (\text{DPAHC} - 32)) \quad (57)$$

Where

$QC_{\text{latent}}$	= cooling coil latent load (Btu/hr).
TSMF	= supply air mass flow ( $\text{lb}_{\text{m,da}}/\text{min}$ ), as defined in Equation 1.
MAHR	= mixed air humidity ratio ( $\text{grains}_w/\text{lb}_{\text{m,da}}$ ), as defined in Equation 43.
SAHR	= supply air humidity ratio ( $\text{grains}_w/\text{lb}_{\text{m,da}}$ ), as defined in Equation 48.
DPAHC	= dew-point temperature of the air entering the cooling coil ( $^{\circ}\text{F}$ ).
60	= conversion factor (minutes/hour).
7000	= conversion factor ( $\text{grains}/\text{lb}_m$ ).
1075.21	= specific enthalpy of evaporation at $32^{\circ}\text{F}$ ( $\text{Btu}/\text{lb}_{\text{m,w}}$ ). This value appears in <i>2009 ASHRAE Handbook of Fundamentals</i> (ASHRAE 2009), p. 1.8.
0.556	= change in specific enthalpy of evaporation per degree ( $\text{Btu}/(\text{lb}_{\text{m,w}} \cdot ^{\circ}\text{F})$ ).
32	= temperature ( $^{\circ}\text{F}$ ).

### 2.2.1.27 Air Properties

The original RP 865 work (Yuill and Haberl 2002) and the QAS models calculate moist air properties (e.g., humidity ratio and specific volume) using perfect (or ideal) gas laws as presented in *2009 ASHRAE Handbook of Fundamentals*, Chapter 1 (ASHRAE 2009). (Note: *ASHRAE Terminology* [ASHRAE 1991] treats a perfect gas and an ideal gas as equivalent.)

#### 2.2.1.27.1 Humidity Ratio

Humidity ratio calculations are discussed in sections 2.2.1.8, 2.2.1.12, 2.2.1.17, 2.2.1.21, and 2.2.1.23.

#### 2.2.1.27.2 Specific Volume

Specific volume at various system locations is calculated using *2009 ASHRAE Handbook of Fundamentals*, Chapter 1, Equation 28 (ASHRAE 2009):

$$v = ((R_{da} \times (T + 459.67) / p) \times (1 + 1.607858 \times W / 7000)) / 144 \quad (58)$$

Where

- $v$  = moist air specific volume (ft<sup>3</sup>/lb<sub>m,da</sub>)
- $R_{da}$  = dry air gas constant = 53.35 ft·lb<sub>f</sub>/(lb<sub>da</sub>·°R)
- $T$  = temperature of the location for which specific volume is calculated (°F)
- 459.67 = conversion constant: 0°F = 459.67°R
- $p$  = atmospheric pressure = 14.695975 psia
- 1 = one unit of dry air (dimensionless)
- 1.607858 = ratio of molecular masses, dry air to water; the quantity (1.607858 × W / 7000) in Equation 58 represents the dimensionless ratio of (moles water) / (moles dry air)
- $W$  = humidity ratio (grains<sub>w</sub>/lb<sub>m,da</sub>) of moist air at the location for which specific volume is calculated
- 7000 = conversion factor (grains/lb<sub>m</sub>)
- 144 = conversion factor (inch<sup>2</sup>/ft<sup>2</sup>).

Each calculation of specific volume uses the air temperature and humidity ratio for the location of interest. Table 2-3 lists the location of the temperature and humidity ratio used for each calculation; locations within “( )” can be found in Figure 2-1.

**Table 2-3. Specific Volume Calculation Input Locations**

Var	Description	Location of T and W
ACCSV	Specific volume after cooling coil (same as supply fan entering air).	Air leaving cooling coil (cco). This is the same as the air entering supply fan
ROSV	Specific volume of zone one supply air	Zone one (z1)
RTSV	Specific volume of zone two supply air	Zone two (z2)
RFISV	Specific volume of return fan inlet	Air entering return fan (rfi)

### 2.2.1.27.3 Specific Heat

The specific heat of moist air at various system locations is calculated as:

$$SH = CpAir + 0.444 \times W / 7000 \quad (59)$$

Where

- SH = specific heat (Btu/(lb<sub>m,da</sub>·°F))
- CpAir = specific heat of dry air 0.2403 (Btu/(lb<sub>m,da</sub>·°F))
- 0.444 = specific heat of water vapor (Btu/(lb<sub>m,w</sub>·°F))
- W = humidity ratio (grains<sub>w</sub>/lb<sub>m,da</sub>) of the location for which specific volume is being calculated
- 7000 = conversion factor (grains/lb<sub>m</sub>).

Each calculation of specific heat uses the humidity ratio for the location of interest as described in Table 2-4; locations within “( )” can be found in Figure 2-1. The humidity ratio used for each calculation is as follows:

**Table 2-4. Specific Heat Calculation Input Locations**

Var	Description	Location of W
SASH	Supply air specific heat	Air entering supply fan (cco)
ROSH	Zone one supply air specific heat	Air entering supply fan (cco)
RTSH	Zone two supply air specific heat	Air entering supply fan (cco)
RASH	Return air specific heat	Air entering return fan (rfi)
OASH	Outdoor air specific heat	Outdoor air (oa)
MASH	Mixed air specific heat	Mixed air (ma)

### 2.2.1.27.4 Enthalpy

When the enthalpy economizer is selected, the calculation of the return air enthalpy (RAE) and outdoor air enthalpy (OAE) are required. The enthalpy at various system locations is calculated as:

$$h = CpAir \times DBT + (W / 7000) \times (1061.15 + 0.444 \times DBT) \quad (60)$$

Where

- h = enthalpy of moist air (Btu/lb<sub>m,da</sub>), based on an enthalpy of 0 Btu/lb<sub>m,da</sub> at 0°F for dry air, and an enthalpy of 1061.15 Btu/lb<sub>m,w</sub> at 0°F for water vapor
- CpAir = specific heat of dry air 0.2403 (Btu/(lb<sub>m,da</sub>·°F))
- DBT = temperature of the system location for which enthalpy is calculated (°F)
- W = humidity ratio (grains<sub>w</sub>/lb<sub>m,da</sub>) of the system location for which enthalpy is calculated
- 7000 = conversion factor (grains/lb<sub>m</sub>)
- 1061.15 = specific enthalpy of saturated vapor at 0°F (Btu/lb<sub>m,w</sub>)
- 0.444 = specific heat of water vapor (Btu/(lb<sub>m,w</sub>·°F)).

Each calculation uses the temperature and humidity ratio for the location indicated in Table 2-5 for the respective values; locations within “( )” can be found in Figure 2-1.

**Table 2-5. System Location of Variables for Enthalpy Calculations**

Var	Description	Source of DBT and W
OAE	Outdoor air enthalpy	Outdoor air temperature and humidity ratio (oa)
RAE	Return air enthalpy	Return air temperature and humidity ratio (rfo). The humidity ratio does not change across the return fan so the humidity ratio at rfo is not explicitly calculated and the equation uses the humidity ratio of air entering the fan (rfi) instead.

### **2.2.2 RP 865 Spreadsheet Models Review: Differences Identification and Resolution**

The development of QAS models began with a detailed comparison of the final RP 865 spreadsheet models. The RP 865 researchers individually provided the most current versions of their models for the seven system types developed. The review and detailed comparison looked at the models for six of the seven system types:

- Fan coil (FC)
- Single zone (SZ)
- Constant air volume with zone reheat (CV) system
- Variable air volume with reheat (VAV) system
- Dual duct constant air volume (DDCAV) system
- Dual duct variable air volume (DDVAV) system.

The first step was to compare the provided models (Haberl 2010; Haberl et al. 2002; Yuill 2010) with the published results of Yuill and Haberl (2002). These models are included with the accompanying files, as indicated in Readme-Airside-HVAC.docx there. Setting model inputs to those specified in the RP 865 input specification found no agreement between the model and published results except for the fan coil system. This broad disagreement was traced to the published specification indicating a return duct heat gain of 1°F, while the published results were calculated with no return duct heat gain.

After specifying a return duct heat gain of 0°F, the provided models replicate the results published in the project report in most but not all cases. In particular, the TAMU models do not replicate many results reported in the RP 865 final report. However, the provided TAMU models do replicate the results reported in a separate TAMU report for most cases. The PSU model results also vary slightly from the published results in some cases. Differences between the RP 865 Report values and the provided models include:

- The coil loads for the TAMU CV system model do not agree with the PSU published loads attributed to the TAMU model. This is a small but significant difference that impacts heating and latent coil loads in all test cases.
- The coil loads for the TAMU VAV system model do not agree with the PSU published loads attributed to the TAMU model. The differences primarily impact predicted cooling loads.
- The coil loads for the TAMU DDCAV system model do not agree with the PSU published loads attributed to the TAMU model. This is a small but significant difference that impacts heating and latent coil loads in all test cases.
- The PSU DDVAV values are different from the PSU report values for test Case 2, where it appears the report values for all modes are actually for the enthalpy economizer mode.
- The PSU DDVAV values are slightly different from the PSU report for Case 4.

The models provided by the RP 865 researchers are the most current and reflect the on-going refinement before the Standard 140 adaptation work began (Haberl 2010; Yuill 2010). Figures 2-2 through 2-6, and the accompanying electronic files as presented in Section 2.3, provide bar charts that can be used to compare the reported values with the provided models.

### 2.2.2.1 Comparison Methodology

Predicted heating and cooling coil loads and detailed air properties (e.g., temperature, humidity ratio, specific volume, mass flow) from the PSU and TAMU models were compared to isolate model differences. The specific model results compared are defined in Part I, Section 1.6.5. The comparisons were conducted at test case ambient conditions and using nomenclature of the original RP 865 work. Table 2-6 presents a summary of the ambient and zone conditions. During the comparison an additional test case was added with greater latent loads (Case 7). This case is roughly equivalent to a fully occupied conference room with extremely humid (100%) exterior conditions. The zones are idealized adiabatic zones where the specified sensible and latent gains represent the only zone loads.

The ambient conditions in this comparison are close to but not the same as the ambient conditions used in the proposed Standard 140 Air Side HVAC test cases (see Part I of this report). In the final proposed Standard 140 adaptation of the original research, the test case dry-bulb temperature conditions were changed from integer degrees Fahrenheit to exact tenths of a degree Celsius, and the moisture conditions were adjusted to minimize round-off error of dew-point temperature and relative humidity so that the provided Typical Meteorological Year 2 (TMY2) weather format could be as close to the specified conditions as possible.

**Table 2-6. Test Case Ambient and Zone Conditions Summary (I-P units)**

	Test Type	Ambient		Zone 1			Zone 2		
		DB	WB	Zone T	Sensible Gains	Latent Gains	Zone T	Sensible Gains	Latent Gains
		F	F	F	Btu/hr	Btu/hr	F	Btu/hr	Btu/hr
Case 1	Heating	-20	-20	70	-10000	2000	72	-8000	3000
Case 2	Heating	30	20	71	-2000	2000	73	1000	3000
Case 3	Cooling	60	45	74	5000	2000	76	8000	3000
Case 4	Cooling	80	75	75	10000	2000	77	12000	3000
Case 5	Cooling	77	55	74	5000	2000	76	8000	3000
Case 6	Cooling	74	70	74	5000	2000	76	8000	3000
Case 7	Cooling	75	75	75	10000	10000	75	10000	20000

Results from the RP 865 models were compared for each test case. When differences were found in the model results, patterns in the differences and direct model inspection were used to identify calculation differences. Once identified, one of the models was revised to address the identified issue in the same manner as the other model to facilitate further comparisons. Next, new results were generated, and the residual differences used to guide further checks.

The model results of the unmodified models (which we began with) were quite close for all system types (see accompanying files PartII-Files\QAS-Results-FCSZ.xlsm and QAS-Results-CVVV.xlsm for figures showing initial differences for all system types). Only a small number of differences were resolved to get to the ending models. Some of the model differences and modifications were obvious errors, others involved more complex issues. The original RP 865 work did not separate latent and sensible cooling and several identified calculation differences were related to the sensible/latent coil load split. The worst

disagreement was a 25% difference in total cooling load for Case 7 with dry-bulb temperature economizer, where one of the models had an equation error.

In the final comparisons, most model coil load residual differences are within 0.01%, indicating substantially all differences between the two models are explained. In a few cases, there are differences as large as 0.12%, indicating some residual difference is not explained.

Tables 2-7a and 2-7b present the starting and ending percent difference, respectively, between the TAMU and PSU models for the single zone (SZ) system without economizer. This system type had the lowest residual difference. The low residual difference indicates substantially all calculation differences have been identified. The starting and ending residual differences for the FC, VAV and DDVAV system models (not shown) are similar, as are the economizer runs for those systems.

Tables 2-8a and 2-8b present the starting and ending percent differences between the Constant Volume Reheat (CV) models. This system type had the largest residual difference. This residual difference indicates at least some difference in the model calculations was not identified. The starting and ending match for the Dual Duct Constant Volume system (not shown) is similar.

**Table 2-7a. Starting Coil Load Differences TAMU vs. PSU, SZ System – No Economizer**

Case	Zone 1				Zone 2			
	Heating	Cooling			Heating	Cooling		
	Total	Sensible	Latent	Total	Total	Sensible	Latent	Total
1	-0.05%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%
2	0.05%	0.00%	0.00%	0.00%	0.17%	0.00%	0.00%	0.00%
3	0.00%	-0.82%	0.00%	-0.82%	0.00%	-1.05%	0.00%	-1.05%
4	0.00%	-1.11%	-0.68%	-1.05%	0.00%	-1.30%	-1.03%	-1.19%
5	0.00%	-0.37%	0.00%	-0.37%	0.00%	-0.43%	0.00%	-0.43%
6	0.00%	-0.54%	-2.53%	-1.17%	0.00%	-0.74%	-3.09%	-1.51%
7	0.00%	-2.51%	-0.31%	-1.28%	0.00%	-3.70%	-0.35%	-1.49%

**Table 2-7b. Ending Coil Load Differences TAMU vs. PSU, SZ System – No Economizer**

Case	Zone 1				Zone 2			
	Heating	Cooling			Heating	Cooling		
	Total	Sensible	Latent	Total	Total	Sensible	Latent	Total
1	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%
2	-0.01%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%
3	0.00%	0.03%	0.00%	0.03%	0.00%	0.00%	0.00%	0.00%
4	0.00%	0.00%	0.00%	0.00%	0.00%	0.01%	0.00%	0.00%
5	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%
6	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%
7	0.00%	0.00%	0.01%	0.00%	0.00%	0.00%	0.00%	0.00%

**Table 2-8a. Starting Coil Load Differences TAMU vs. PSU, CV – No Economizer**

Case	Cooling			Heating			
	Sensible	Latent	Total	Preheat	Zone 1 Reheat	Zone 2 Reheat	Total
1	0.00%	0.00%	0.00%	0.99%	0.06%	0.07%	0.21%
2	-1.54%	0.00%	-1.54%	0.00%	0.07%	0.09%	0.08%
3	-0.25%	0.00%	-0.25%	0.00%	0.14%	0.17%	0.16%
4	-1.24%	0.82%	-0.41%	0.00%	0.35%	0.29%	0.30%
5	-0.16%	0.00%	-0.16%	0.00%	0.14%	0.17%	0.16%
6	-0.93%	0.83%	-0.31%	0.00%	0.14%	0.17%	0.16%
7	-2.48%	1.09%	-0.45%	0.00%	0.35%	0.23%	0.27%

**Table 2-8b. Ending Coil Load Differences TAMU vs. PSU, CV – No Economizer**

Case	Cooling			Heating			
	Sensible	Latent	Total	Preheat	Zone 1 Reheat	Zone 2 Reheat	Total
1	0.00%	0.00%	0.00%	0.01%	0.01%	0.01%	0.01%
2	-0.12%	0.00%	-0.12%	0.00%	0.00%	0.01%	0.00%
3	-0.02%	0.00%	-0.02%	0.00%	0.00%	0.01%	0.01%
4	-0.01%	-0.01%	-0.01%	0.00%	0.03%	0.02%	0.01%
5	-0.02%	0.00%	-0.02%	0.00%	0.00%	0.01%	0.01%
6	-0.02%	-0.01%	-0.01%	0.00%	0.00%	0.01%	0.01%
7	-0.01%	0.00%	-0.01%	0.00%	0.03%	0.02%	0.00%

### 2.2.2.2 Identified Model Differences

Several calculation differences between the models were found. Each difference was evaluated, and a determination made as to preferred calculation method. The tables that follow present a complete listing of the identified differing calculation procedures. The preferred method chosen for the QAS model approach is discussed in the column labeled “Comment”. Table 2-9 presents issues impacting all system types in the model set. Table 2-10 presents items which impact only the specified system types.

The following abbreviations are used in the items column of the tables below. Variable names in the PSU and TAMU columns are specific to the original PSU or TAMU models and are not defined here or in the abbreviations section of this document. They are included for guidance when looking at the original models.

- MADB = mixed air dry-bulb temperature
- OAMF = outdoor air mass flow rate
- OASH = outdoor air specific heat
- OAT = outdoor air temperature
- Qsens = zone sensible heat gain (positive) or loss (negative)
- RAMF = return air mass flow rate
- RASH = return air specific heat
- RAT = return air temperature
- SAMF = supply air mass flow rate
- SAT = supply air temperature

Tdp = dew-point temperature  
Tset = zone air set point temperature  
TSMF = Total system supply air mass flow rate  
XXXX = the variable that is in error and being discussed.

*1993 HoF* = *1993 ASHRAE Handbook of Fundamentals* (ASHRAE 1993)  
*2005 HoF* = *2005 ASHRAE Handbook of Fundamentals* (ASHRAE 2005)  
*2009 HoF* = *2009 ASHRAE Handbook of Fundamentals* (ASHRAE 2009)  
*2012 HS&E* = *2012 ASHRAE HVAC Systems and Equipment* (ASHRAE 2012)

lb<sub>m,da</sub> = pounds of dry air  
lb<sub>m,w</sub> = pounds of water  
ln p<sub>w</sub> = natural logarithm of the water vapor partial pressure

The file CVreheat.xls, referenced in Table 2-10, is included with the accompanying files in subfolder \PartII-Files\RP-865-Spreadsheets\TAMU.



**Table 2-9. Differences Found In All System Models**

Item	PSU Model	TAMU Model	Comment
Constant pressure specific heat, dry air (Btu/(lb <sub>m,da</sub> ·°F)) (used in enthalpy & other calcs)	0.24	0.2402 (Constant set and referenced by calculations, except a few which have 0.24 specified directly)	<p>The <i>1993 HoF</i> &amp; <i>2009 HoF</i>, specify specific heat of dry air as 0.2404 Btu/(lb<sub>m,da</sub>·°F) at 80°F. Hyland/Wexler (ASHRAE 2005) standardizes to 0.24036 Btu/(lb<sub>m,da</sub>·°F) in all heating calculations and 0.24033 Btu/(lb<sub>m,da</sub>·°F) in all cooling calculations.</p> <p>Two additional sources that provide temperature dependent formulations were consulted. For the current test cases, the temperature range of interest is from minus 20°F to 85°F. Borgnakke and Sonntag (2013) provides a cubic function that predicts 0.2402 Btu/(lb<sub>m,da</sub>·°F) to 0.2404 Btu/(lb<sub>m,da</sub>·°F) over the range of interest. Andreas (2005) provides a cubic function that predicts 0.24014 Btu/(lb<sub>m,da</sub>·°F) to 0.24039 Btu/(lb<sub>m,da</sub>·°F) over the range of interest.</p> <p>The QAS models use a constant 0.2403Btu/(lb<sub>m,da</sub>·°F) . Both temperature dependent functions predict this value for 70°F air, and it is mid-range of the values predicted for the temperatures of interest. The impact of this change is generally a 0.1% increase in heating case loads, though reheat in the CV system test cases increases by up to 0.4%.</p>
Rate of change of h <sub>fg</sub> (enthalpy of evaporation), (Btu/(lb <sub>m,w</sub> ·°F))	0.556	0.56506 Source is not given. Very close to per degree values calculated from <i>2009 HoF</i> , Chapter 1, Table 3.	<p>The <i>2009 HoF</i> ideal gas law equations use a specific heat of water = 1.0 Btu/(lb<sub>m,w</sub>·°F), a specific heat of evaporation = 0.556 Btu/(lb<sub>m,w</sub>·°F), and a specific heat of water vapor of 0.444 Btu/(lb<sub>m,w</sub>·°F). Calculating the per degree enthalpy difference from <i>2009 HoF</i> (ASHRAE 2009), Chapter 1, Table 3 between 60°F and 70°F results in specific heat of water = 1.0 Btu/(lb<sub>m,w</sub>·°F), specific heat of evaporation = 0.565 Btu/(lb<sub>m,w</sub>·°F), and specific heat of water vapor = 0.4434 Btu/(lb<sub>m,w</sub>·°F). The TAMU model used 0.565 Btu/(lb<sub>m,w</sub>·°F) for the specific heat of evaporation and 0.444 Btu/(lb<sub>m,w</sub>·°F) for that of water vapor.</p> <p>The impact of using a specific enthalpy of evaporation of 0.556 Btu/(lb·°F) versus 0.565 Btu/(lb·°F) is a decrease in latent coil load of 0.15% in the wet coil cases and as much as a 0.003% change in sensible coil loads.</p> <p>The QAS models will utilize 0.556 Btu/(lb<sub>m,w</sub>·°F).</p>
Specific enthalpy of water vapor, h <sub>g</sub> (Btu/(lb <sub>m,w</sub> ·°F))	0.444	0.444	<p>Not different but reviewed because this is associated with the specific heat of evaporation.</p> <p>The <i>2009 HoF</i> ideal gas law equations use specific heat of water = 1.0 Btu/(lb<sub>m,w</sub>·°F), a specific heat of evaporation = 0.556 Btu/(lb<sub>m,w</sub>·°F), and a specific heat of water vapor of 0.444 Btu/(lb<sub>m,w</sub>·°F). Calculating the per-degree</p>

Item	PSU Model	TAMU Model	Comment
			<p>enthalpy difference from <i>2009 HoF</i>, Chapter 1, Table 3 between 60°F and 70°F results in specific heat of water = 1.0 Btu/(lb<sub>m,w</sub>·°F), specific heat of evaporation = 0.565 Btu/(lb<sub>m,w</sub>·°F), and specific heat of water vapor of 0.4434 Btu/(lb<sub>m,w</sub>·°F). The TAMU model used 0.565 Btu/(lb<sub>m,w</sub>·°F) for the specific heat of evaporation and 0.444 Btu/(lb<sub>m,w</sub>·°F) for that of water vapor.</p> <p>The <i>Handbook of Physical Constants and Functions for Use in Atmospheric Boundary Layer Studies</i> (Andreas 2005) equations predict a near-constant value of 0.44378 Btu/(lb<sub>m,w</sub>·°F) over the range of interest.</p> <p>For the current test cases, the impact of using a specific heat of water vapor of 0.444 Btu/(lb<sub>m,w</sub>·°F) versus 0.44378 Btu/(lb<sub>m,w</sub>·°F) is a 0.003% increase in latent loads and a 0.001% decrease in sensible cooling and heating loads.</p> <p>The QAS models will utilize 0.444 Btu/(lb<sub>m,w</sub>·°F).</p>
Enthalpy of water vapor, h <sub>g</sub> , at 0°F (Btu/(lb <sub>m,w</sub> ·°F))	1061	1061	QAS models use 1061.15 per <i>2009 HoF</i> , Chapter 1, Table 3.
Gas constant Ra (ft·lb <sub>f</sub> /(lb <sub>m,da</sub> ·°R)) (used in all specific volume calcs)	53.352	53.35	<i>1993 HoF</i> through <i>2005 HoF</i> use 53.352, while the <i>2009 HoF</i> uses a new molar constant of 53.35 ft·lb <sub>f</sub> /(lb <sub>m,da</sub> ·°R). The QAS models use 53.35 ft·lb <sub>f</sub> /(lb <sub>m,da</sub> ·°R).
Molecular mass ratio (dimensionless ratio of molar masses)	1.6078 = dry air to water; 0.62198 water to dry air.	1.6078 = dry air to water; 0.62198 water to dry air.	The QAS models use the new <i>2009 HoF</i> molar mass ratios of 1.607858 for dry air to water and 0.621945 for water to dry air.
Coefficients used in calculation of the water vapor saturation pressure of outdoor air	Coefficients largely from <i>1989 HoF</i> . Differ from coefficients used in the calculation of the same property in the cooling coil leaving air temperature which match <i>2009 HoF</i> (and editions back to 1993).	Coefficients from <i>2009 HoF</i> and consistent with all Handbooks since the 1989 HoF. Consistent throughout calculations.	TAMU coefficients are as specified in <i>2009 HoF</i> . Coefficients used by PSU may be oversight since values elsewhere in the same model are from newer sources. Difference in predicted humidity ratio is 0.026% in the cold test case and less than 0.001% for cooling cases. The predicted humidity ratio for test case 2 in the original work differs by 0.6%, but this test case is not used in the Part I, Section 1.5.5 tests. The QAS models use the coefficients from the <i>2009 HoF</i> .

Item	PSU Model	TAMU Model	Comment
Fourth coefficient of correlation used to determine the water vapor saturation pressure of the air leaving the cooling coil ( $C_{11}$ in QAS Equation 50)	0.000012890363	0.000012890360	TAMU value is as specified in <i>2009 HoF</i> . PSU value is a typographical error. The QAS models use the <i>2009 HoF</i> value.
Default humidity ratio if ideal gas calculation returns low or negative value ( $lb_{m,w}/lb_{m,da}$ )	0 if negative	Varies by system type between 0.0001 and 0.00000001. Used if calculated value is below the default value	No impact on coil loads but has significant impact on detailed output for Case 2 in the original RP 865 test cases. Required partly because neither model implements equation to calculate humidity ratio when dry-bulb temperature is below freezing ( <i>2009 HoF</i> , Chapter 1, Equation 37). This does not impact the Part I, Section 1.5.5 test cases because the cases below freezing (AE101, AE201, AE301, AE401) are at saturated conditions (wet-bulb temperature equals dry-bulb temperature) and the two equations return the same value at that point. QAS model implements <i>2009 HoF</i> , Chapter 1, Equation 37 for calculating humidity ratio when the dry-bulb temperature is below freezing. This equation can still return negative values under some conditions. The QAS uses default humidity ratio of 0 in these circumstances. This new equation has not been as extensively checked as other aspects of the solution. If new test cases are developed with non-saturated, below freezing air temperature, the results should be carefully scrutinized.
Sensible/latent coil load assignment	The PSU model calculates the latent coil load as the product of the condensed moisture and the enthalpy at the entering air dry-bulb temperature. Sensible coil load includes energy to cool the condensed water from CCEAT to CCLAT with a $C_p = 1.0$ Btu/(lb·°F).	TAMU calculates the coil latent load as the product of the condensed moisture and the enthalpy at the entering air dew-point temperature. Sensible coil load includes energy to cool the to-be-condensed water vapor from CCEAT to CCEADP at 0.444 Btu/(lb·°F).	The QAS models calculate latent coil load using the enthalpy at the cooling coil entering air dew-point temperature.

Item	PSU Model	TAMU Model	Comment
Coil sensible load calc – final temperature of condensed moisture	Assumes that the condensate leaves at the cooling coil leaving air temperature.	Assumes the condensate leaves at the cooling coil entering air dew-point temperature. Calculates sensible cooling as total cooling using enthalpy difference (hin-hout) and then subtracts out latent load [(Win-Wout)*Hfg] and the enthalpy of the liquid condensate [(Win-Wout)*Hw]. See Equation 39 in 2012 HS&E, Chapter 23. Hfg is taken at the cooling coil entering air dew-point temperature. Hw is taken at the same dew-point temperature meaning the liquid is assumed to leave the system at the dew-point temperature. No energy is calculated to cool the condensate below the entering air dew-point temperature while 2012 HS&E suggests condensate is cooled to the leaving air wet-bulb temperature.	<p>Degree of sub-cooling is significant variable. In real systems, condensate will be at all temperatures from the entering air dew-point temperature to slightly below the leaving air temperature. These test cases assume bypass factor = 0 which constrains the condensate to be no lower than the leaving air temperature. The actual temperature of the condensate as it goes down the drain will depend on the coil and drain pan configuration. Approach here is arbitrary.</p> <p>The QAS models follow the PSU approach and assume condensate is cooled from the entering air dew point to the cooling coil leaving air temperature.</p> <p>The following modifications to the TAMU model are presented in case future efforts use the TAMU models. They are not needed for the current QAS development.</p> <p>To adjust the TAMU model, the Qsensible equation must be changed. The equation as implemented:</p> $Q_{\text{sensible}} (QCS) = m_{\text{air}}[(h_{\text{in}} - h_{\text{out}}) - (W_{\text{in}} - W_{\text{out}})(h_{\text{fg}, T_{\text{dp}, \text{in}}} + h_{\text{w}, T_{\text{dp}, \text{in}}})]$ <p>The existing equation reformatted for clarity. The last term is removing the condensate from the system:</p> $QCS = m_{\text{air}}[(h_{\text{in}} - h_{\text{out}}) - (W_{\text{in}} - W_{\text{out}}) \times h_{\text{fg}, T_{\text{dp}, \text{in}}} - (W_{\text{in}} - W_{\text{out}}) \times h_{\text{w}, T_{\text{dp}, \text{in}}}]$ <p>The modified equation is:</p> $QCS = m_{\text{air}}[(h_{\text{in}} - h_{\text{out}}) - (W_{\text{in}} - W_{\text{out}}) \times h_{\text{fg}, T_{\text{dp}, \text{in}}} - (W_{\text{in}} - W_{\text{out}}) \times h_{\text{w}, T_{\text{ccladb}, \text{in}}}] \text{ \{Note: "ccladb" is cooling coil leaving air dry-bulb temperature\}}$ <p>Calculation of <math>h_{\text{w}, T_{\text{ccladb}, \text{in}}}</math> must be added.</p>
Fan temperature rise	Uses constant of 0.006689 Btu/ft <sup>3</sup> ·in. wg.	Uses effective constant of 0.00667175 Btu/(ft <sup>3</sup> ·in. wg). Actually uses (in SZ model): 0.0001573 hp / (cfm·in. wg) in calculation of	QAS models use 0.006685 Btu/ft <sup>3</sup> ·in. wg. Derived from Borgnakke and Sonntag (2013), page 756: 0.036126 lb <sub>r</sub> /in <sup>2</sup> per in.wg. × 144in <sup>2</sup> /ft <sup>2</sup> × 0.00128507 Btu per lb <sub>r</sub> -ft.

Item	PSU Model	TAMU Model	Comment
		horsepower (row 216); 2544.85 Btu·hr / hp in Dtfan calculation (rows 242 & 244); and 60 minutes in hour in Mdesign calculation (row 240)	
Model convergence/ iteration	As sent recalculates manually. Changed to calculate automatically until stable or until 200 iterations completed.	Depends upon system type. Iterates 20 times for FC model, 16 for several others, and 11 for another.	For heating cases, TAMU model humidity ratio output not completely stable in several cases. Number of iterations of all TAMU models was increased to 30, which allowed humidity ratios to converge. QAS models are currently only implemented in PSU format using Excel™ with iterative calculation turned on, with maximum iterations set to 200 and maximum change to 0.0001
Constant volume return fans	PSU assumes return fan design cfm is the supply cfm minus exhaust cfm. The actual return fan cfm is calculated as the supply mass flow minus exhaust mass flow multiplied by the return fan inlet air specific volume. Any difference between design and actual cfm is considered an off-design condition and an exponential correction to the return fan pressure drop is applied.	Assumes that the pressure specified by the test specification is the operating pressure of the return fan at the test case conditions. Constant volume return fan power is not varied except as volumetric flow changes between test cases. No adjustment to pressure drop is calculated.	Specification states return fan pressure varies by square of the volumetric flow. The specification assumes that reduced/increased return volumetric flow leads to reduced/increased pressure drop. As the specification is written even the constant volume system return fan pressure will vary. In buildings controlled for pressurization this is a common arrangement, but in many fan systems the supply, exhaust, and return fans self-adjust to balance with all fans maintaining constant volume flow as well as possible.  QAS models assume pressure drop varies by the square of the volumetric flow for all systems and fans.

<b>Item</b>	<b>PSU Model</b>	<b>TAMU Model</b>	<b>Comment</b>
Variable volume fans	Follows RP 865 document. Assumes fan static pressure varies with the square of the actual-to-design flow ratio. The specified cfm is design; the actual cfm is based upon the mass flow and the specific volume of the air entering the fan.	Implements fan coefficients from Brandemuehl et al. (1993), which are close to the RP 865 specification but do not vary fan static pressure exactly as the square of the actual to design airflow ratio.	In both models there are several assumptions built in. The specification states fan pressure drop varies with the square of the volumetric flow; the QAS models implements this.

**Table 2-10. System Type Model Specific Differences**

<b>System Type</b>	<b>Item</b>	<b>PSU Model</b>	<b>TAMU Model</b>	<b>Comment</b>
FC	Denominator of mixed air dry-bulb temperature calculation $MADB = (RAMF \times RAT \times RASH + OAMF \times OAT \times OASH) / (TSMF \times XXXX)$	Uses room supply air specific heat (SASH) for the “XXXX” term.		Verified bug in PSU model. QAS model uses mixed air specific heat (MASH) instead.
FC, SZ	Zone supply air temperature calculation $SAT = Tset - Qsens / (SAMF \times XXXX)$	Uses the supply air specific heat for the “XXXX” term.	Uses the room air specific heat, which includes mass from zone latent gains, for the “XXXX” term.	Verified bug in TAMU model. QAS models use supply air specific heat.
FC, SZ	Coefficient in correlation of dew-point temperature with the partial pressure of water vapor.	Not used.	For dew point < 32°F: second term of calculation uses 26.412 °F/ln p <sub>w</sub> , Handbook of Fundamentals uses 26.142 °F/ln p <sub>w</sub>	Verified bug in TAMU model. Dewpoint correlation is implemented in QAS models use 26.142 °F/ln p <sub>w</sub> .
SZ	Specific volume of CCLAT		TAMU equation for specific volume of the cooling coil leaving air: $Vccladb = Ra \times Tccladb (1 + 1.6078 \times Wcclaw) / (70.7262 \times Pfan,enter)$ . Formula implementing equation in cell G797 references cell G792, a blank line, for the Wcclaw term. Wcclaw reference changed to cell G791.	Verified bug in TAMU model. Not applicable to QAS models.
CV, DDCV	Return air humidity ratio calculation		Uses zone supply mass flow to weight the individual zone return air streams. Line 291 in CVreheat.xls uses m1,ccladb from line 118 and m2,ccladb from line 168 to weight the zone humidity ratios. Return humidity ratio should be calculated using m1,w/EXHAUST and m2,w/EXHAUST from lines 264 and 266.	Verified bug in TAMU model. QAS models use zone return mass flow.

System Type	Item	PSU Model	TAMU Model	Comment
SZ	PSU mixed air humidity ratio	The PSU mixed air humidity ratio calculation checks for return air mass flow. If yes, it calculates a weighted value between the return air and the minimum outside air flow. If no, it assumes the economizer is operating and uses the outdoor air humidity ratio directly for the mixed air humidity ratio. The problem is the return air mass flow that is checked (RORMF) is the room return air mass flow, so it never finds the economizer operating even when it is. The spreadsheet calculates a separate variable, the recirculated air mass flow (RCMF), with the economizer logic. This variable is the value that should be referenced in the calculation of mixed air humidity ratio.		Verified bug in PSU model. QAS model references the correct value.
CV	Re-circulated air mass flow rate		In CVreheat.xls, lines 577 and 579 adjust the outdoor and re-circulated mass flow rates to achieve the desired mixed air temperature in the free cooling mode. The sum of rows 118 (m1,ccladb) and 168 (m2,ccladb) equal mma (line 262). Per model documentation in row 559, Mbp = Mma-Moa, therefore Mbp+Moa should equal Mma. But lines 577 and 579, the adjusted Mbp and Moa, do not equal Mma. The formulas in cells H579 through V579 do not account for the fact that the total mass flow (mma) changes with each iteration. Formula in cell G579 should be changed to Mpa, Adjust = Mma – Moa,adjust, i.e., (G118+G168)-G577, and the equation copied to columns H through V. The values moa,adjust + mbp,adjust always equal the currently predicted mass flow.	Verified bug in TAMU model. Not applicable to QAS model.



### 2.2.2.3 QAS Model Creation

Once the RP 865 differences were identified and the preferred method of calculation for each issue was determined, a QAS model was created for each system type. Given the final agreement between the two model sets (PSU and TAMU) and the distribution of the preferred calculations between the model sets, the model set chosen as the basis for the QAS models was somewhat arbitrary. The PSU model set was chosen, and modifications were made to implement the preferred calculation method for each of the identified model differences. Further modifications were made to the QAS models, as presented in Section 2.3.

#### 2.2.2.3.1 Changed Outdoor Humidity Ratio Calculation

The original RP 865 models apply outdoor air dry-bulb and wet-bulb temperatures to describe the ambient conditions. The final test case specification describes ambient conditions in terms of outdoor air dry-bulb and dew-point temperatures to provide better consistency with the provided TMY2 weather files. The QAS model was modified to calculate ambient moisture properties based upon the specified dew-point temperature (see Section 2.2.1.17).

Table 2-11 shows a comparison of the predicted humidity ratio from using wet-bulb versus dew-point temperature based equations, respectively.

**Table 2-11. Comparison of QAS Ambient W from DP Input vs. W from WB Input, Final Test Spec**

Case	Wet Bulb		Dew Point		% dif
	WB(°C)	W (kg/kg)	DP (°C)	W (kg/kg)	
AE201	-29.0	0.000258927	-29.0	0.000258927	-0.00%
AE203	7.206	0.002923683	-3.0	0.002935884	0.42%
AE204	23.441	0.016769722	22.1	0.016773557	0.02%
AE205	13.027	0.004473716	2.4	0.004490867	0.38%
AE206	21.523	0.01555347	20.9	0.01555525	0.01%

Table 2-12 compares the predicted coil loads from the wet-bulb temperature versus dew-point temperature based humidity ratio equations. There is no impact on coil sensible loads, and the impact on latent coil latent loads is < 0.2%.

**Table 2-12. Predicted Coil Loads from QAS: DP Input vs. WB Input, Final Test Spec**

Case	QAS-WB equations (kWh)			QAS-DP equations (kWh)			Relative Difference (%)		
	Sensible	Latent	Total	Sensible	Latent	Total	Sensible	Latent	Total
AE201	-8.360	0.000	-8.360	-8.360	0.000	-8.360	0.00%		0.00%
AE203	0.856	0.000	0.856	0.856	0.000	0.856	0.00%		0.00%
AE204	3.574	2.053	5.627	3.574	2.054	5.628	0.00%	0.05%	0.02%
AE205	1.916	0.000	1.916	1.916	0.000	1.916	0.00%		0.00%
AE206	1.706	0.970	2.676	1.706	0.970	2.677	0.00%	0.06%	0.02%
AE226	1.562	1.798	3.360	1.562	1.800	3.362	0.00%	0.09%	0.05%
AE245	2.208	0.000	2.208	2.208	0.000	2.208	0.00%		0.00%

## 2.3 Quasi-Analytical Solution Results

### 2.3.1 New Ambient Test Case Conditions

The test case ambient conditions specified in Part I are changed from the original RP 865 ambient conditions. These changes are summarized in Tables 2-13 and 2-14 for initial and final revisions respectively.

The RP 865 test cases provided dry-bulb and wet-bulb temperature in degrees Fahrenheit. The test specification (see Part I) provides ambient weather data as TMY2 formatted files. The TMY2 file format provides:

- Dry-bulb temperature to the nearest tenth of a degree Celsius
- Relative humidity to the nearest whole percent
- Dew-point temperature to the nearest tenth of a degree Celsius.

To minimize errors associated with using the provided weather files, initial new ambient cases (before the first round of simulation trials) were established to be as close as possible to the original test conditions while minimizing conversion issues. The initial new test cases are specified in SI units with:

- Dry-bulb temperature specified exactly with tenth of a degree Celsius precision.
- Moisture variables chosen to minimize (balance) weather file round-off error of relative humidity (nearest percent) and dew-point temperature (nearest tenth of a degree Celsius).

After the initial simulation trials the test case ambient conditions were adjusted so that:

- Dry-bulb temperature specified exactly with tenth of a degree Celsius precision.
- Dew-point temperature specified exactly with tenth of a degree Celsius precision.

These final test case ambient conditions are very close to the initial revision values with the dew-point and wet-bulb temperature changing by 0.002°C or less.

**Table 2-13. Initial New Ambient Conditions**

Case	New Case Conditions (SI) <sup>a</sup>					New Case Conditions (I-P)		Original Case Conditions (I-P)	
	DB (°C)	W (g/gda)	WB (°C)	DP (°C)	RH (%)	DB (°F)	WB (°F)	DB (°F)	WB (°F)
AE201	-29.0	0.002600	-29.000	-29.002	99.983	-20.20	-20.200	-20	-20
AE203	15.5	0.002948	7.207	3.002	27.023	59.90	44.972	60	45
AE204	26.9	0.016850	23.44	22.098	75.011	80.42	74.191	80	75
AE205	24.9	0.004510	13.029	2.400	23.048	76.82	55.452	77	55
AE206	23.0	0.015630	21.525	20.902	87.980	73.40	70.744	74	70

a. Humidity ratio (W) is exact. The wet-bulb temperature (WB), dew-point temperature (DP), and relative humidity (RH) are equivalent variables calculated to the precision shown based upon the Hyland-Wexler real gas model (ASHRAE 2005) as implemented by the CYTSoft psychrometric calculator (CYTSoft 2009).

**Table 2-14. Final Ambient Conditions**

Case	New Case Conditions (SI) <sup>a</sup>				New Case Conditions (I-P) <sup>a</sup>			Original Case Conditions (I-P)	
	DB (°C)	DP (°C)	RH (%)	WB (°C)	DB (°F)	DP (°F)	WB (°F)	DB (°F)	WB (°F)
AE201	-29.0	-29.0	100.000	-29.000	-20.20	-20.20	-20.20	-20	-20
AE203	15.5	-3.0	27.028	7.206	59.90	26.60	44.971	60	45
AE204	26.9	22.1	75.023	23.441	80.42	71.78	74.194	80	75
AE205	24.9	2.4	23.050	13.027	76.82	36.32	55.449	77	55
AE206	23.0	20.9	87.968	21.523	73.40	69.62	70.741	74	70

a. Dew-point temperature (DP) is exact. The relative humidity (RH) and wet-bulb temperature (WB) are equivalent variables calculated to the precision shown based upon the real gas model developed by ASHRAE RP-1485 (Herrmann et al. 2011) as implemented in the ASHRAE “LibHuAirProp” spreadsheet functions (Kretzschmar et al. 2011).

### 2.3.2 Comparison of QAS Model and Original RP 865 Results

This section presents results published in the RP 865 project report, results generated by the provided (unaltered) RP 865 models at the RP 865 ambient conditions, results generated by the provided (unaltered) RP 865 models at the initial new ambient conditions (see Table 2-13), and results from the QAS models for the final ambient conditions specified in Part I (see Table 2-14).

Figures 2-2 through 2-6 (see after Table 2-15) contain selected results sets described in Table 2-15 for all four systems (FC, SZ, CV, and VAV).

Figures in this section use case labels from the early simulation trials. The simulation-trial test cases used a prefix of “AET”, and the FC and SZ test cases had a suffix of “a” to identify the zone. All of the corresponding test cases in Part I have a prefix of “AE”, and the FC and SZ test cases only utilize a single zone description, so the “a” suffix is dropped.

Figure 2-2 presents a comparative plot of the heating coil loads for the FC and SZ system results. Differences between the original RP 865 model and the QAS model results are small. Differences between the RP 865 model results at the original and initial new ambient conditions are also small.

Figure 2-3 presents a comparative plot of the total cooling coil loads for the FC and SZ system results. Differences between the RP 865 and QAS model results are small with the largest differences occurring in the wet coil cases AET104a, AET204a, and AET226a. Results differences between the original and new ambient conditions are small but noticeable in the wet coil cases.

Figure 2-4 presents a comparative plot of the heating coil loads for the CV and VAV systems. As with the FC and SZ systems, the differences between the original RP 865 and the QAS model results are small and the differences between the RP 865 models at the original and initial new ambient conditions are small.

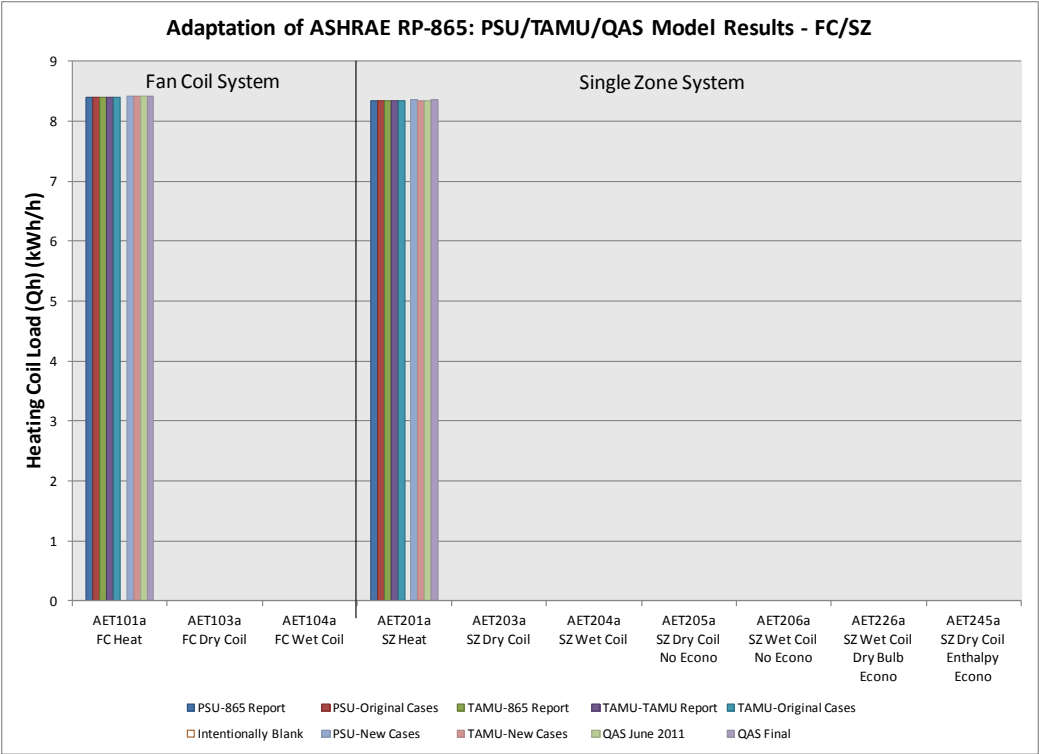
Figure 2-5 presents a comparative plot of the total cooling coil loads for the CV and VAV systems. Differences between the original RP 865 models and the QAS model results are small. Results differences between the original and new ambient conditions are small, but with noticeable differences in the wet coil cases.

Figure 2-6 presents a comparative plot of the zone reheat coil loads for the CV and VAV systems. There is very little difference between the original RP 865 models and the QAS model, and also very little difference between the RP 865 models at the original and initial new ambient conditions.

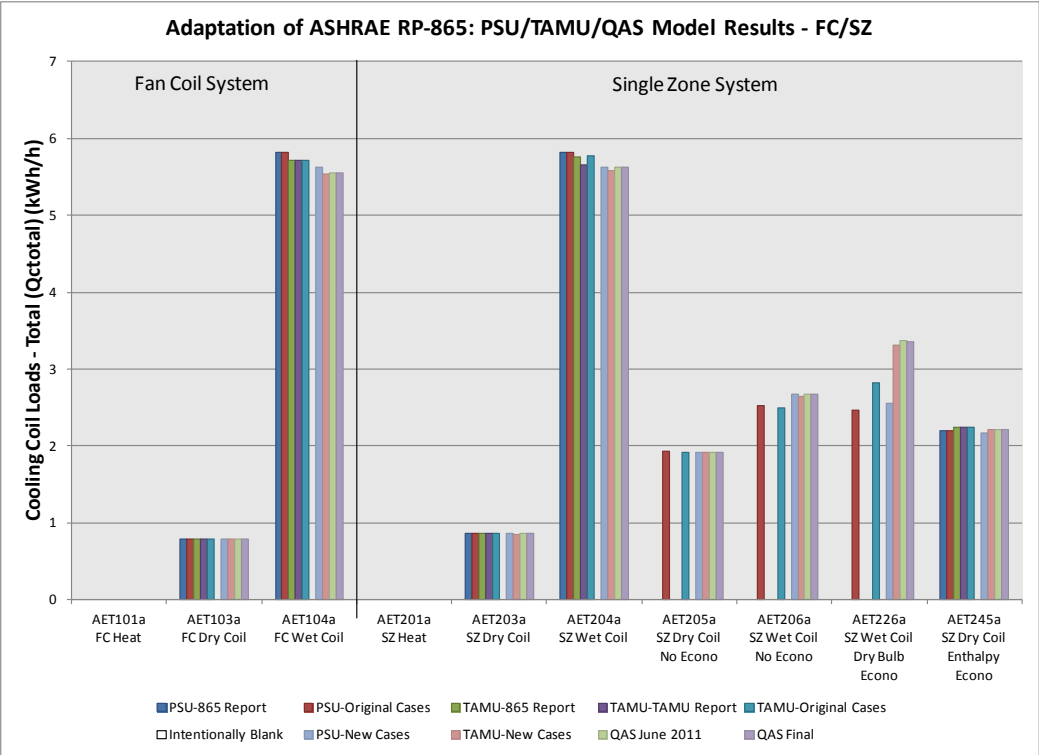
More comparative results figures for FC and SZ systems are contained in the spreadsheet “QAS-Results-FCSZ.xlsm”, which is included as part of the informative accompanying electronic files (see subfolder “PartII-Files”). Comparative results figures for the CV and VAV systems are contained in the spreadsheet “QAS-ResultsCVVV.xlsm”, which is included as part of the informative accompanying electronic files (see subfolder “PartII-Files”). The description of results in Table 2-15 also applies to the figures contained in both of these accompanying electronic files.

**Table 2-15. Results Sets by Ambient Conditions**

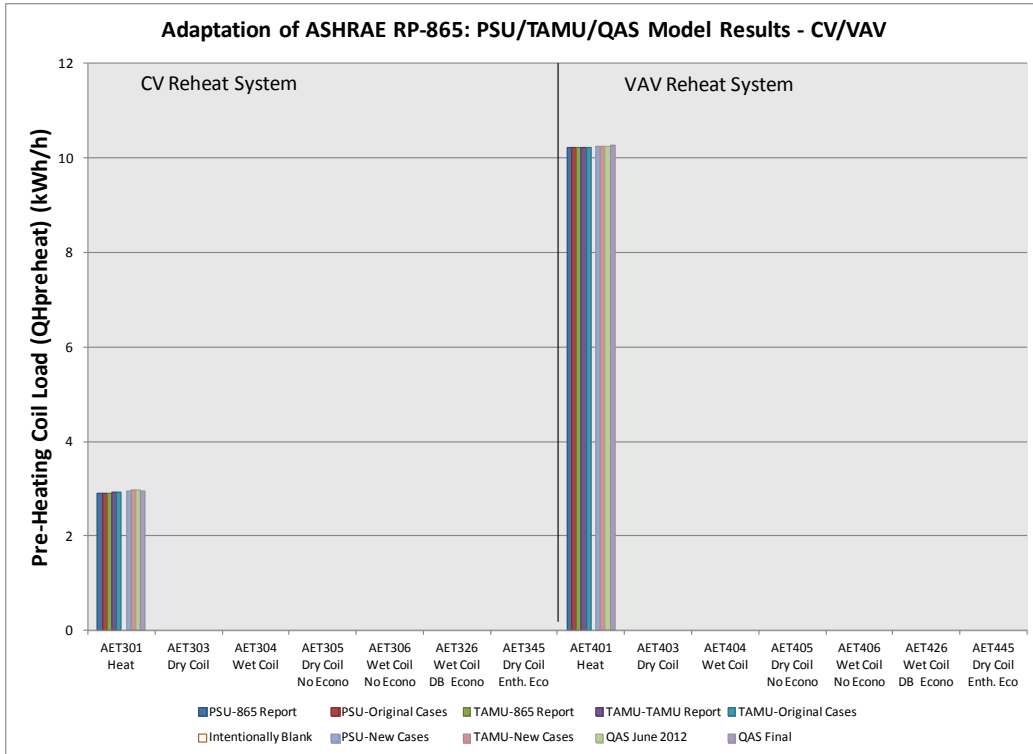
<b>Label Used in Figures</b>	<b>Description</b>
<i>Results at original RP 865 ambient conditions</i>	
PSU-865 Report	PSU results from RP 865 project report (Yuill and Haberl 2002)
PSU-Original Cases	PSU results from provided spreadsheet (Yuill 2010), implemented by NREL
TAMU-865 Report	TAMU results from RP 865-2002 project report (Yuill and Haberl 2002)
TAMU-TAMU Report	TAMU results from TAMU report (Haberl et. al 2002)
TAMU-Original Cases	TAMU results from provided spreadsheets (Haberl 2010), implemented by NREL
<i>Results separation placeholder</i>	
Intentionally Blank	Provides separation space for delineating results at original RP 865 ambient conditions (five data bars to the left of this space) from results at new SI ambient conditions (four data bars to right of this space). Results at new SI conditions are not expected to agree with results at original ambient conditions.
<i>Results at new SI ambient conditions specified in Part I</i>	
PSU-New Cases	PSU results from provided spreadsheet at initial new SI conditions
TAMU-New Cases	TAMU results from provided spreadsheets at initial new SI conditions
QAS June 2011	Results from the June 2011 QAS models for FC and SZ at initial new SI conditions. These were the first results presented to working group for the QAS FC and SZ system models.
QAS June 2012	Results from the June 2012 QAS models for CV and VAV at initial new SI conditions. These were the first results presented to working group for the QAS CV and VAV system models.
QAS Final	Results from final QAS models (completed November 2014) at the final SI conditions.



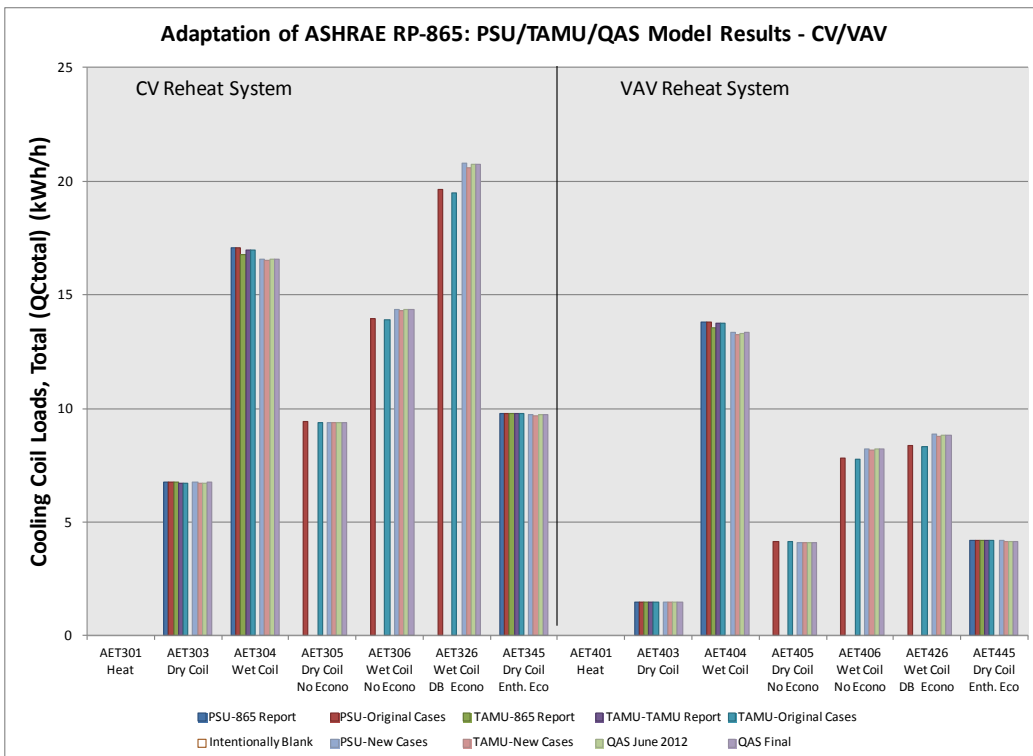
**Figure 2-2 FC and SZ Test Case Heating Coil Loads: RP 865 and QAS Models**  
 Note: Data labels shown in the legend are described in Table 2-15.



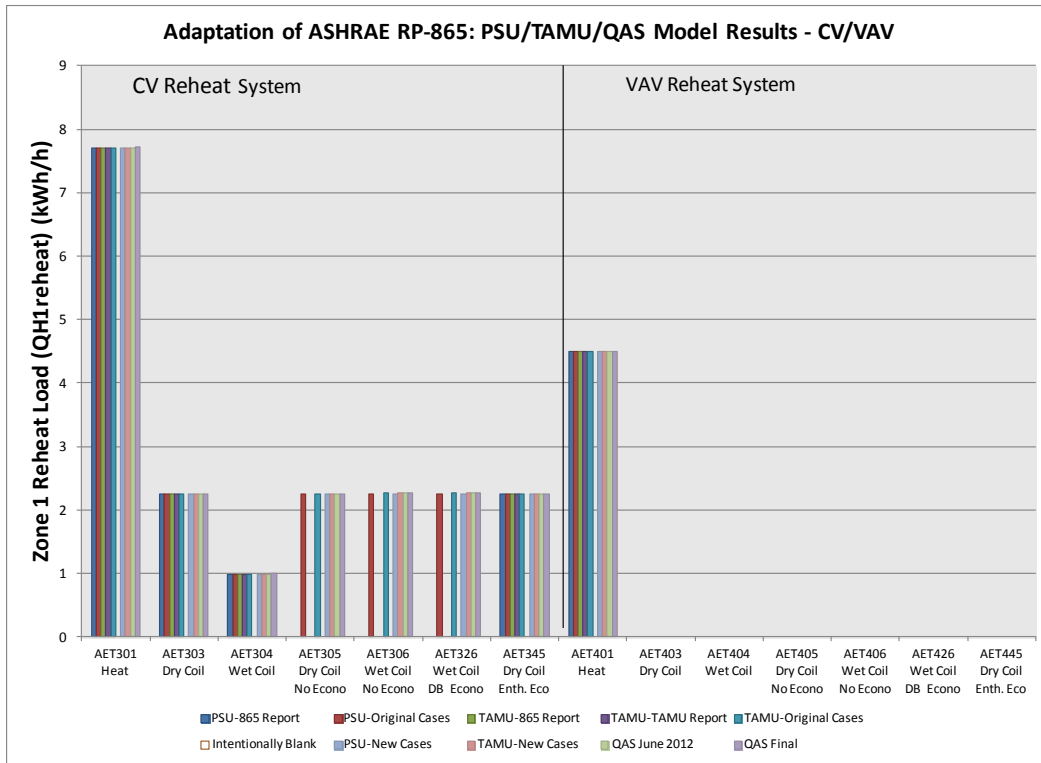
**Figure 2-3 FC and SZ Test Case Total Cooling Coil Loads: RP 865 and QAS Models**  
 Note: Data labels shown in the legend are described in Table 2-15.



**Figure 2-4 CV and VAV Test Case Pre-heat Coil Loads: RP 865 and QAS Models**  
 Note: Data labels shown in the legend are described in Table 2-15.



**Figure 2-5 CV and VAV Test Case Total Cooling Coil Loads: RP 865 and QAS Models**  
 Note: Data labels shown in the legend are described in Table 2-15.



**Figure 2-6 CV and VAV Test Case Zone 1 Re-heat Coil Loads: RP 865 and QAS Models**  
 Note: Data labels shown in the legend are described in Table 2-15.

### 2.3.3 Other QAS Results Checks

During the development of the QAS, several versions of the merged model were created. To document changes during the development process, early results presented to the SSPC140 working group are shown here along with the final QAS results. The early results are those presented to the working group in June 2011 for the FC and SZ systems and in June 2012 for the CV and VAV systems. From the earlier system models to the final QAS system models, results differences are attributable to:

- Changing physical constants (e.g., heat capacity of dry air) from earlier values (0.24 Btu/(lb·°F) or 0.2402 Btu/(lb·°F)) to the final constants used in the QAS Final (0.2403 Btu/(lb·°F))
- Finding and fixing errors in the implementation of the QAS models, such as use of more than a single value for a physical constant in different locations
- Errors in running the early models where inputs were not correctly specified for a given case
- Errors made in transcribing results from the early QAS models to the standard output spreadsheet
- Specification of exact ambient dew-point temperature in final test specification, and revised QAS equations to utilize ambient dew-point temperature rather than ambient wet-bulb temperature as input.

### 2.3.3.1 Revisions and Corrections

Specific model changes between the early and the final QAS results include:

- Implemented variable volume return fan in CV system. This was originally in the base model, but was removed for early result runs and then reinstated for final QAS.
- Deleted calculations for second zone from the FC model and changed several named ranges to remove "RO" prefix so more variable names are the same as parallel values in other system types.
- Changed calculation of coil outlet air relative humidity to never return a value greater than 100%; previously values of about 100.005% were observed.
- Changed coil load calculations to not return extremely small non-zero numbers (e.g., 0.00000001 kWh). These result from coil loads always being calculated using before-and-after temperature differences and humidity ratios even when the coil does not operate. Small differences occur due to the iterative solution.
- Change in "Inpws" coefficients for outdoor air humidity ratio calculation from ASHRAE (1989) values to ASHRAE (2009) values.
- Fixed typos (minor change) in coefficients used in calculation of the saturation pressure of water vapor ( $\ln(p_{ws})$ ) used in the supply air humidity ratio calculation.
- Separate equation added for humidity ratio of outdoor air when  $T < 32^{\circ}\text{F}$ ; previously, the equation for  $T > 32^{\circ}\text{F}$  was used.
- In FC/SZ system models, I-P to SI conversion factor of 3413 Btu/kWh updated to 3412.14 Btu/kWh; this was also applied to the "PSU-New Cases" and "PSU-Original Cases" results.
- Changed gas constant from  $53.352 \text{ ft}\cdot\text{lb}_f / \text{lb}_{m,da} \cdot ^{\circ}\text{R}$  to  $53.35 \text{ ft}\cdot\text{lb}_f / \text{lb}_{m,da} \cdot ^{\circ}\text{R}$  for specific heat calculations in SZ, CV, and VAV systems.
- Change molecular mass ratio (water/dry air) used in calculating saturation humidity ratios and  $P_w$  from 0.62198 to 0.621945.
- Changed molecular mass ratio (dry air/water) used in calculating specific volume from 1.6078 to 1.607858.
- Changed constant (enthalpy of water vapor,  $h_g$  at  $0^{\circ}\text{F}$ ) used in calculation of enthalpy from 1061 Btu/lb<sub>m,w</sub> to 1061.15 Btu/lb<sub>m,w</sub>.
- Changed constant (enthalpy of evaporation,  $h_{fg}$  at  $32^{\circ}\text{F}$ ) used in calculating cooling coil latent load from 1075.15 to 1075.21 Btu/lb<sub>m,w</sub>.
- Changed constant (conversion factor) in fan work to heat equation from  $0.0066717 \text{ Btu}/(\text{ft}^3 \cdot \text{in. wg})$  to  $0.006685 \text{ Btu}/(\text{ft}^3 \cdot \text{in. wg})$  in FC and SZ (replaces TAMU value that was accidentally left in QAS version 10 file with final QAS value).
- Changed constant (conversion factor) in fan work to heat equation from  $0.006689 \text{ Btu}/(\text{ft}^3 \cdot \text{in. wg})$  to  $0.006685 \text{ Btu}/(\text{ft}^3 \cdot \text{in. wg})$  in CV and VAV.
- Changed atmospheric pressure used in calculations from 14.696 psia to 14.695975 psia ( $1.01325 \text{ bar} \times 14.5038 \text{ bar/psi}$ )
- Added new calculation of ambient humidity ratio to utilize ambient inputs of dry-bulb and dew-point temperatures rather than dry-bulb and wet-bulb temperatures as discussed in Section 2.2.2.3.1.

### 2.3.3.2 Results comparisons from Aug 2012 to final version

The results differences between the early QAS results presented to the working group and the final QAS results are as follows:

- FC/SZ CCO leaving RH in Cases AE103, AE203, and AE205 are 0.14% lower.
- CV/VAV CCO leaving RH in Cases AE303, AE305, AE403, and AE405 are ~0.07% lower.



- FC heating coil load is 0.1% higher.
- SZ heating coil load is 0.15% higher.
- VAV pre-heating coil load is 0.05% higher.
- CV pre-heating is 0.8% lower. Attributable to addition of variable volume return fan with version 8 and increase in CpAir from 0.24 Btu/(lb·°F) to 0.2403 Btu/(lb·°F).
- FC sensible cooling is 0.1% to 0.5% lower. SZ sensible cooling is 0.03% to 0.13% higher for non-economizer cases and 0.5% to 0.7% lower for cases AE226 and AE245. These differences are attributable to inconsistent CpAir values in old results with 0.2402 Btu/(lb·°F) being used for everything except for system specific heat calculations, which were calculated with 0.24 Btu/(lb·°F). The outdoor air specific heat was calculated using 0.2402 Btu/(lb·°F).
- CV sensible cooling is ~0.3% higher. Attributable to addition of variable volume return fan and changing specific heat from 0.24 Btu/(lb·°F) to 0.2403 Btu/(lb·°F).
- VAV sensible cooling is the same except Case AE403, which is 0.08% lower.
- FC/SZ latent cooling is 0.1% lower.
- CV/VAV latent cooling is 0.01% lower.
- CV/VAV reheat is 0.02%–0.18% higher. Attributable to variation in CPair, which impacts CV system cooling and heating.
- Outdoor humidity ratio:
  - In FC/SZ, W is 0.15% lower in Cases AE103, AE105, AE203, and AE205 and differs by +0.02% to -0.02% in other cases.
  - In CV/VAV, W is 0.06% lower in Cases AE303, AE305, AE403, and AE405, and differs by +0.02% to -0.01% in other cases.
  - W is the same between systems in the final QAS results. Initial FC/SZ results are from June 2011, and initial CV/VAV results are from June 2012. Correction of some issues apparent in the early FC/SZ before development of the early CV/VAV results accounts for less change in outdoor humidity ratio for the CV/VAV results.
- FC/SZ supply air temp -0.04% low in Cases AE101 and AE201, and 0.03%-0.07% high in other cases.
- FC/SZ supply air W is -0.01% low in Cases AE101 and AE201, and differs by -0.15% to +0.06% in other cases.
- CV/VAV supply air W 0% to 0.08% low.
- FC/SZ/CV supply air mass flow is same.
- VAV mass flow 0.04% low for all cases.

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## 2.5 Abbreviations and Acronyms for Part II

Abbreviations listed do not include QAS variable names discussed in Section 2.2.1, QAS variables present in the models but not discussed, or PSU/TAMU model variables discussed in Section 2.2.2.2.

Btu	British thermal unit
cfm	cubic foot per minute
CV	airside HVAC system: constant air volume with zone reheat
FC	airside HVAC system: fan coil with heating and cooling
ft	foot
grains	measure of mass, 7000 grains = 1 pound
grains <sub>w</sub>	grains of water
h <sub>fg</sub>	enthalpy of evaporation
h <sub>g</sub>	enthalpy of water vapor
hp	horsepower
hr	hour
<i>HoF</i>	<i>Handbook of Fundamentals</i>
in	inch
lb <sub>f</sub>	pounds force
lb <sub>m,da</sub>	pounds mass of dry air
lb <sub>m,w</sub>	pounds mass of water
ln p <sub>w</sub>	natural log of the partial pressure of water vapor
min	minute
psia	air pressure: pounds per square inch
PSU	models developed by Gren Yuill while at Pennsylvania State University and University of Nebraska
QAS	quasi-analytical solution, see definition in Section 2.1
RP 865	ASHRAE Research Project 865
SZ	airside HVAC system: single zone with heating and cooling
TAMU	models developed by Jeff Haberl of Texas A&M University
VAV	airside HVAC system: variable air volume with zone reheat
wg	water gage

## 3.0 Part III: Production of Simulation Results

### 3.1 Introduction

In this section, we describe what the working group members did to produce example results with several detailed programs that were considered to represent the state of the art for building energy simulation in the United States, China, Japan, and the United Kingdom. The objectives of developing the simulation results were:

- To demonstrate the general applicability and usefulness of the test suite for a variety of state of the art building energy simulation programs with a variety of input structures and modeling approaches
- To improve the test procedure through field trials
- To eliminate input errors and internal modeling bugs in the participating computer programs such that the set of example results (see Part IV) is representative of the current range of legitimate modeling differences in today's state-of-the-art building energy simulation software
- To identify the range of disagreement that may be expected for simulation programs relative to the quasi-analytical solution (QAS) results that constitute a reliable set of theoretical results for these specific test cases (see Part IV).

The field trial effort took about 4 years and involved several revisions to the test specifications and subsequent re-execution of the computer simulations. The process was iterative in that executing the simulations led to the refinement of the test suite, and the results of the tests led to the improvement and debugging of the models. This process underscores the importance of the software development industry participation in this project via the ASHRAE Standard 140 project committee (SSPC 140), and via our previous collaborations with International Energy Agency software testing and validation projects (Judkoff and Neymark 2009). Extensive field trials and the resulting enhancements to the tests were much more comprehensive and cost effective with the voluntary participation of the national and international software-development industry experts.

Table 4-1 (see Part IV, Section 4.1) describes the programs used to generate the simulation results. Appendix III (Section 3.9) presents reports written by the modelers for each simulation program.

The tables and graphs in Part IV present the final results from all the simulation programs and QASs used in this study. The QAS results constitute a reliable set of theoretical results. Therefore, the primary purpose of including simulation results for the AE101 – AE445 test cases in Part IV is to allow simulationists to compare their relative agreement (or disagreement) with the QAS results *versus* the relative agreement of the other Part IV simulation results with the QAS results (i.e., a comparison with the state of the art in simulation). Perfect agreement among simulations and the QAS is not necessarily expected: this is because many programs contain simplifying assumptions to ease calculation burden (e.g., constant air density), and the QAS contains idealized simplifying assumptions (in order to be solved analytically) that cannot always be exactly reproduced by some simulation programs that are conceived and hardcoded with more realistic assumptions. The Part IV results give an indication of what sort of reasonable agreement is possible between simulation results and the QAS results.

Abbreviations and acronyms used in Sections 3.1 through 3.6 are given in Section 3.7. References cited in Section 3.1 through 3.6 are given in Section 3.8.

## 3.2 Selection of Simulation Programs and Modeling Rules for Simulations

The industry working group participating in this project made the initial selections of the simulation programs used in this study. The selection criteria required that:

- A program be a true simulation based on hourly weather data and calculation time increments of 1 hour or less
- A program be representative of the state of the art in whole-building energy simulation as defined by the working group participants making the selection.

The modeling rules were somewhat different (more stringent) for the simulation programs used for Part IV example results than for a given program to be normally tested with this procedure (see Part I, Section 1.3, Modeling Rules). For the Part IV simulation results, we allowed a variety of modeling approaches. However, we required that these cases be modeled in the most detailed way possible for each simulation program within the limits of the test specification (e.g., bypass factor = 0 is a specified idealization that some programs had difficulty modeling exactly).

To minimize the potential for user error, we encouraged more than one modeler to develop input files for each program. We also encouraged participants to check their results versus the QAS and versus other participants' simulation program results. We also distributed documentation of noteworthy isolated results differences and the probable diagnosis to each program participant during the simulation trials.

Where improvements to simulation programs or simulation inputs were made as a result of running the tests, such improvements must have a mathematical and physical basis and must be applied consistently across tests. In addition, all improvements were requested to be documented in modeler reports. Arbitrary modification of a simulation program's input or internal code just for the purpose of more closely matching a given set of results is not allowed. The diagnostic process of improving simulation models discussed in Section 3.4 also isolated input errors that were corrected, as noted there and in the modeler reports (Section 3.9).

## 3.3 Improvements to the Test Specification as a Result of the Field Trials

Based on comments by the ASHRAE SSPC 140 industry working group participants during the field trials and observations from the adaptation of the QASs, we made a number of improvements and revisions to the test specification. Although researching the comments and communicating specification revisions to the field trial participants was very time consuming, the importance of the accuracy and clarity of the test specification for this type of work cannot be overstated. This is the only method to our knowledge that can produce a test specification that is unambiguous to a variety of state-of-the-art simulation programs with a variety of input structures and modeling approaches.

The contribution of the SSPC 140 industry working group participants was particularly valuable because the expert software developer teams supplied continuous feedback throughout the 4-year field trial effort. Their feedback resulted in several revisions to the test specifications and subsequent re-execution of the computer simulations. This iterative process led to refinement of the test suite, and the results of the tests led to the improvement and debugging of the programs.

### 3.3.1 Development of the Initial Draft Test Specification (Simulation Trial #1)

It should be noted here that although the original ASHRAE RP 865 final report (Yuill and Haberl 2002) is technically complete, it does not document any simulation trials beyond development of the two

independent spreadsheet solutions (nor were the spreadsheet solutions externally documented). Therefore, the original RP 865 test specification had not been vetted for establishing the ability of a variety of simulation industry users to apply the test procedure independently for a variety of simulation programs. Such vetting has been established for previous NREL BESTEST work that has been adapted for ASHRAE Standard 140 (e.g., see Judkoff and Neymark 1995a, 1995b; Neymark and Judkoff 2002, 2004; Neymark and Judkoff et al. 2008a).

The initial draft test specification for only the FC and SZ single-zone systems (which are the simplest of the RP 865 systems) was distributed March 22, 2011. The test specification is based on ASHRAE RP 865, with extensive clarifications to address a greater variety of software applied for industry simulation trials and ASHRAE SSPC 140 working group comments that took place during initial test specification development. Primary early revisions were:

- Development of equivalent inputs
- Creation of artificial TMY2 weather data with constant values
- Development of an alternative zone specification for defining loads on the system.

The specification language was also revised to be more compatible with ANSI/ASHRAE mandatory language requirements.

During the initial test specification development phase, work also began to develop a Verified External Calculation Method (VECM), which is a merged version of the two separate external calculations developed for the original ASHRAE RP 865 work. The final version of this solution is defined as the QAS for this final report. Development of the QAS from the original RP 865 solutions is described in greater detail in Part II, Sections 2.2.2 and 2.3.

### ***3.3.2 Revisions to the Draft Test Specification***

Revisions to the test specification are summarized in the bullet list below. There were a total of six revision iterations, including the initial draft. Revisions addressed SSPC 140 working group comments as the work progressed. Many of the comments and resulting clarifications by the test specification authors were related to ambiguities in the original RP 865 test specification. For example, during the initial round of simulation trials it became apparent that some of the assumptions included in the RP 865 spreadsheet solutions were not clearly stated in the original RP 865 test specification and these details needed to be brought forward, clarified, or expanded. Some test specification issues were only recognizable after larger ambiguities, which caused observable results disagreements, were identified and fixed.

- **Revisions to Initial Draft (Simulation Trial #2)**
  - Distributed May 15, 2012
    - Added CV and VAV multi-zone systems
    - Numerous clarification revisions incorporating SSPC 140 comments from the initial simulation trial round, including changes to definitions, schematic diagrams, output requirements, etc.
  - Continue developing “VECM” QASs for FC, SZ, CV, and VAV systems (See Part II, Sections 2.2.2 and 2.3).
- **Adjustments to Initial Revisions (Simulation Trial #2a)**
  - Distributed Dec 11, 2012
    - Address SSPC 140 comments on specification of CV and VAV multi-zone systems, including: definition and specification clarifications related to fan heat

- and airflow modeling (supply and return fans), finalization of schematic diagrams, economizer control logic clarifications, coil operation clarifications, and output requirements.
  - Revisions to “VECM” QASs (see Part II, Sections 2.2.2 and 2.3).
- **Third Round of Revisions (Simulation Trial #3)**
  - Distributed May 30, 2013
    - Further clarifications to fan, airflow, and latent load modeling specifications and related informative notes and definitions for all four systems.
  - Begin development of detailed “VECM” QAS documentation.
- **Adjustments to Third Round of Revisions (Simulation Trial #3a)**
  - Distributed Apr 7, 2014
    - Further clarifications regarding: no duct leakage or air loss, economizer modeling, allow adjustment of coil modeling parameters among cases to achieve  $BF = 0$  in each case, adjustment of thermostat settings to match specified zone temperature.
  - VECM/QAS spreadsheet and documentation
    - Review iterations with lead author of RP 865, G. Yuill (2014)
    - Initial draft released to SSPC 140, Apr 7, 2014
    - Definition of NREL’s adaptation of the RP 865 spreadsheet solutions as a “quasi-analytical solution” as defined in ASHRAE Standard 140-2011 Addendum A, now Standard 140-2014 (ANSI/ASHRAE 2014; also see Neymark and Judkoff et al. 2008a) per working group agreement, Jun 30, 2014. (ASHRAE SSPC 140 2014a).
  - Simulation trial participants agreed to identify their programs in the example results (see Part IV), Jan -- Mar 2014 (ASHRAE SSPC 140 2014b, Moore 2014, Yoshida 2014); this indicated consensus that the quality of the test specification and example results content is satisfactory.
- **Final Revisions (Simulation Trial #3b)**
  - Distributed Oct 9, 2014
    - Final refinements regarding: specified ambient conditions based on outdoor dry-bulb and dew-point temperatures in the weather data file, terminology related to zone and coil loads, condensed moisture leaving the cooling coil, and editorial clarifications.
  - QAS spreadsheet and documentation updated for use of ambient dew-point temperature.

### 3.4 Improvements to Programs that Participated in the Simulation Trials

Simulations were performed for each test case with the participating computer programs. At each stage of the exercise, output data from the simulations were compared to the QAS and to each other, and the comparisons and probable diagnosis of disagreements were distributed to the simulation trial participants. Many of the participants included excerpts from selected comparisons in their modeler reports (see Section 3.9).

The test diagnostics revealed bugs, faulty algorithms, potential program improvements, input errors, or some combination of those in all of the tested models. Table 3-18 (see Section 3.6.1.2) summarizes software issues found using the airside HVAC equipment test cases. As discussed in Section 3.2, improvements to simulation programs or simulation inputs made by participants were required to have a

mathematical and a physical basis, and to be applied consistently throughout the test cases; all improvements were requested to be documented in modeler reports. For most of the improvements to the tested programs, illustration of how the test procedure was used to isolate and correct errors is sufficiently discussed in the modeler reports of Section 3.9 (Appendix III), and cross-referencing to each report is indicated in the subsections below using the issue description topic titles of Table 3-18. Where additional supporting information is needed, or where a supplementary modeler report was not provided, supplementary information is provided in the subsections below. Abbreviations used in the tables below are provided in Section 3.7.

### 3.4.1 DEEAP

Issues summarized in Table 3-18 (see Section 3.6.1.2) are described in the participant’s supplementary modeler report (see Section 3.9, Appendix III-A, Subsection 4a). An index of the error descriptions in that subsection is listed in Table 3-1.

**Table 3-1. Initial Simulation Trial Issues Index, DEEAP**

<b>Error Description</b>	<b>Cross Reference (Appendix III-A, 4a)</b>
Mass flow imbalance	“Issue 1”
<i>Input:</i> Fan heat not applied	“Issue 2”, “Issue 3”
Cooling coil outlet W constant at 50%	“Issue 4”
Enthalpy economizer control	“Issue 5”
<i>Input:</i> supply outlet temperature not set	“Issue 6”, “Issue 8”
Reheat coil disaggregation	“Issue 7”

After the items in Table 3-1 were addressed, additional issues (see Table 3-2) were uncovered from further review of intermediate diagnostic results, or occurred because of unintended consequences to the previous fixes.

**Table 3-2. Additional Simulation Trial Issues Index, DEEAP**

<b>Error Description</b>	<b>Cross Reference (Appendix III-A, 4a)</b>
Cooling coil entering W missing	First paragraph bullets, Item 1
Tighter iteration tolerance on W	First paragraph bullets, Item 2
<i>Input:</i> Reference density at supply fan inlet	First paragraph bullets, Item 3
Moist air cp based on different functions	First paragraph bullets, Item 4
Fan heat based on standard air properties	First paragraph bullets, Item 5
Zone latent load from hfg rather than hg	First paragraph bullets, Item 6

Results were not retained for the “cooling coil entering W missing” error. The effect on total cooling coil load from correcting the remaining errors was evaluated by comparing DEEAP results submitted Dec 17, 2012 versus Aug 28, 2012, as shown in Table 3-3 for the SZ system (which exhibited the greatest remaining improvement).



**Table 3-3. SZ System Total Cooling Coil Load Difference, DEEAP, Dec 17, 2012 vs. Aug 28, 2012**

Cases	Total Cooling Coil Load		% Difference
	28Aug2012	17Dec2012	
	(kWh/h)	(kWh/h)	
AET203a	0.82962805	0.85302956	2.8%
AET204a	5.65562017	5.62344006	-0.6%
AET205a	1.88615356	1.90992725	1.3%
AET206a	2.65201761	2.66370355	0.4%
AET226a	3.29546198	3.33495125	1.2%
AET245a	2.17861445	2.20203403	1.1%

### 3.4.2 DeST

The issues summarized in Table 3-18 (see Section 3.6.1.2) are described in the participant’s supplementary modeler report (see Section 3.9, Appendix III-B). An index of the error and selected results discussions in that report is listed in Table 3-4.

**Table 3-4. Simulation Trial Issues Index, DeST**

Issue Description	Cross Reference (Appendix III-B)
<i>Input:</i> Return fan position	Section 4
Constant system air density	Section 5, Subsection “Mass Flow Differences”

### 3.4.3 DOE-2.2

Issues summarized in Table 3-18 (see Section 3.6.1.2) are described in the participant’s supplementary modeler report (see Section 3.9, Appendix III-C). An index of the error and selected results discussions in that report is listed in Table 3-5.

**Table 3-5. Initial Simulation Trial Issues Index, DOE-2.2**

Issue Description	Cross Reference (Appendix III-C)
Zone latent gains calculation, FC system (2 errors)	Section 4
<i>Input:</i> Increase number of run days	Section 5.1
<i>Input:</i> Fan power curve to cube of flow	Section 5.1
<i>Input:</i> Minimum OA setting, CV econo runs	Section 5.2
<i>Input:</i> Minimum OA setting, SZ econo runs	Section 5.3
<i>Input:</i> Return fan efficiency	Section 5.3
<i>Input:</i> OWB	Section 5.4
System air density at mixed-air location properties	Section 6, 4 <sup>th</sup> paragraph

Input changes were made based on intermediate diagnostic output or for consistency with test specification revisions. The effects of these changes, indicated in Table 3-18, are based on the progression of DOE-2.2 output during the simulation trials shown in Table 3-6. For the issue related to system air density determined at mixed-air location properties, the percent differences versus the QAS indicated in Table 3-18 are based on Part IV, Tables 4.4-3 and 4.4-5 (see Section 4.4). This issue is being addressed in DOE-2.3 (see Appendix III-C, Section 8).

**Table 3-6. DOE-2.2 Total Cooling Coil Load Results, 2012 to 2015**

Total Cooling Coil Load									
Cases	Jun 2012 (kWh/h)	% Difference	May 2013 (kWh/h)	% Difference	Jan 2014 (kWh/h)	% Difference	Mar 2014 (kWh/h)	% Difference	Mar 2015 (kWh/h)
AE103	0.7755	0.00%	0.7755	0.19%	0.7769	0.00%	0.7769	0.00%	0.7769
AE104	5.1692	0.00%	5.1692	7.02%	5.5323	0.00%	5.5323	0.01%	5.5329
AE203	0.8479	0.00%	0.8479	0.00%	0.8479	-0.90%	0.8402	0.00%	0.8402
AE204	5.7190	0.00%	5.7190	0.00%	5.7190	-0.13%	5.7114	0.00%	5.7114
AE205	1.9161	0.00%	1.9161	0.00%	1.9161	-0.40%	1.9085	0.00%	1.9085
AE206	2.7429	0.00%	2.7429	0.00%	2.7429	-0.42%	2.7314	0.00%	2.7314
AE226	3.4213	0.00%	3.4213	0.00%	3.4213	-0.48%	3.4049	0.00%	3.4049
AE245	2.1910	0.00%	2.1910	0.00%	2.1910	-0.15%	2.1878	0.00%	2.1878
AE303	6.4347	0.00%	6.4347	0.00%	6.4347	-0.23%	6.4197	0.00%	6.4197
AE304	16.1180	0.00%	16.1180	0.00%	16.1180	-0.09%	16.1037	0.00%	16.1037
AE305	8.9434	0.00%	8.9434	0.00%	8.9434	-0.16%	8.9287	0.00%	8.9287
AE306	14.0454	0.00%	14.0454	0.00%	14.0454	-0.10%	14.0314	0.00%	14.0314
AE326	18.6722	0.00%	18.6722	6.81%	19.9438	0.00%	19.9438	0.00%	19.9438
AE345	9.2499	0.00%	9.2499	0.38%	9.2851	0.00%	9.2851	0.00%	9.2851
AE403	1.4419	-3.60%	1.3900	0.89%	1.4023	-0.02%	1.4021	0.00%	1.4021
AE404	13.4992	-0.94%	13.3720	0.05%	13.3790	-0.03%	13.3755	0.00%	13.3755
AE405	4.1604	-1.30%	4.1062	0.19%	4.1141	-0.01%	4.1138	0.00%	4.1138
AE406	8.2787	-1.38%	8.1647	0.13%	8.1752	0.00%	8.1749	0.00%	8.1749
AE426	8.9346	-1.36%	8.8129	0.17%	8.8282	0.03%	8.8308	0.00%	8.8308
AE445	4.2097	-1.20%	4.1593	0.25%	4.1698	0.01%	4.1701	0.00%	4.1701

One further issue, not covered in the supplementary modeler report, is listed in Table 3-7.

**Table 3-7. Additional Simulation Trial Issue Index, DOE-2.2**

Error Description	Reference
Detailed output report precision (T,W)	Hirsch (2014)

While full precision of temperature and humidity ratio values is carried through for the purpose of system modeling, detailed output report precision of these values is limited, which results in some calculated results differences noted in Part IV. This issue does not affect coil load output.

### 3.4.4 EnergyPlus

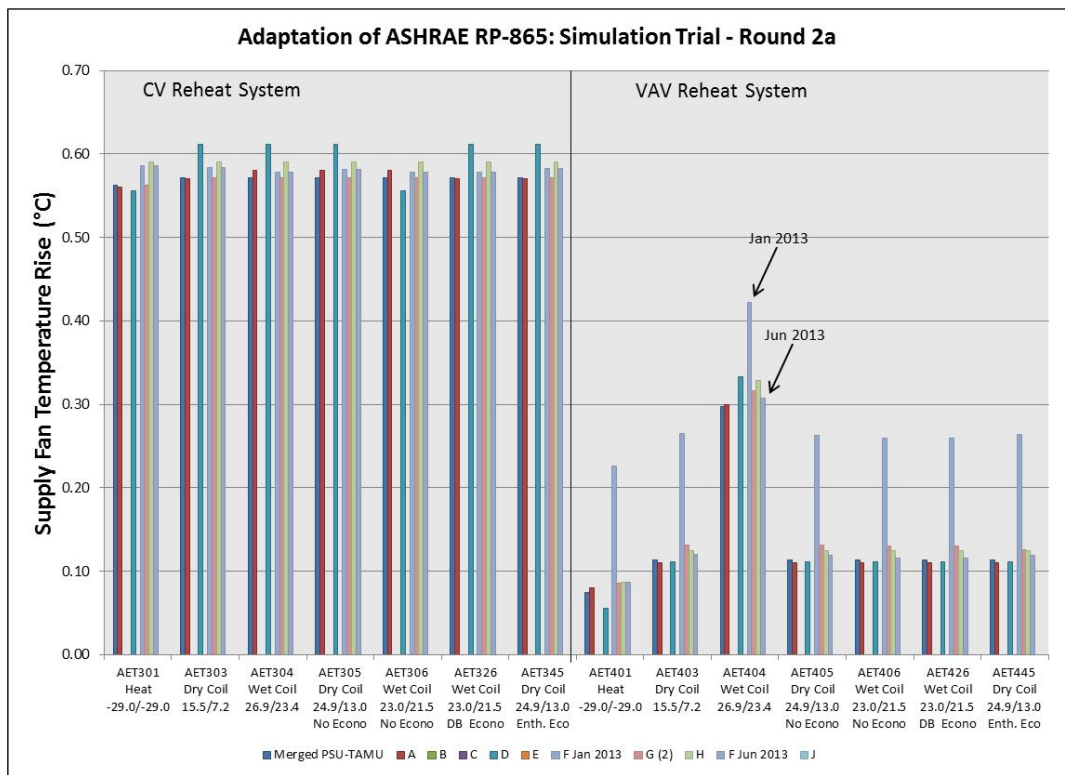
Issues summarized in Table 3-18 (see Section 3.6.1.2) are described in the participant’s supplementary modeler report (see Section 3.9, Appendix III-D). An index of the error and selected results discussions in that report is listed in Table 3-8.

**Table 3-8. Initial Simulation Trial Issues Index, EnergyPlus**

Issue Description	Cross Reference (Appendix III-D)
Heating coil auto-size: CV, VAV systems	Section 4
<i>Input:</i> Reheat coil maximum airflow setting	Section 5, Item e
<i>Input:</i> Fan power curves to cubic	Section 5, Items a,b
<i>Input:</i> Supply fan minimum flow fraction	Section 5, Item c
<i>Input:</i> Return fan airflow rate	Section 5, Item f
Constant altitude-adjusted standard air density	Section 3, last paragraph; also see Part IV, Section 4.4, Tables 4.4-3, 4.4-5.

Figures 3-1 through 3-4, not included in the supplementary modeler report, helped identify various issues. EnergyPlus results as of January 2013 (before input fixes) are identified as “F Jan 2013”; results after input fixes are labeled “F Jun 2013”, respectively. These are sorted by modeler report cross-reference item, as indexed in Table 3-8.

**3.4.4.1 Item 5a**



**Figure 3-1. CV and VAV Supply Fan Temperature Rise, Jan–Jun 2013**

### 3.4.4.2 Item 5b

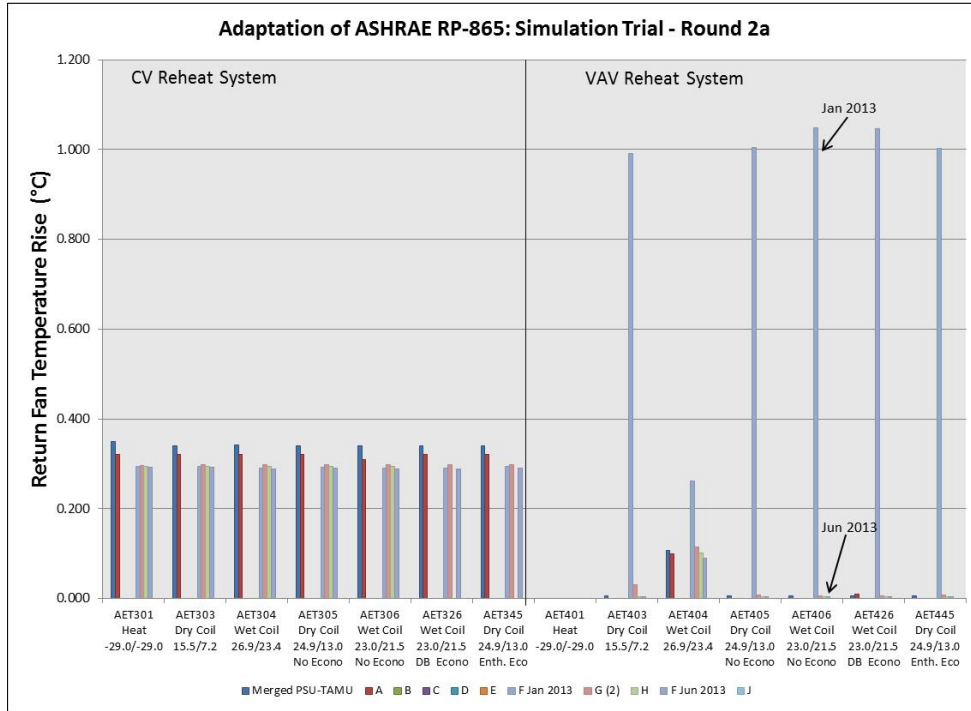


Figure 3-2. CV and VAV Return Fan Temperature Rise, Jan–Jun 2013

### 3.4.4.3 Item 5c

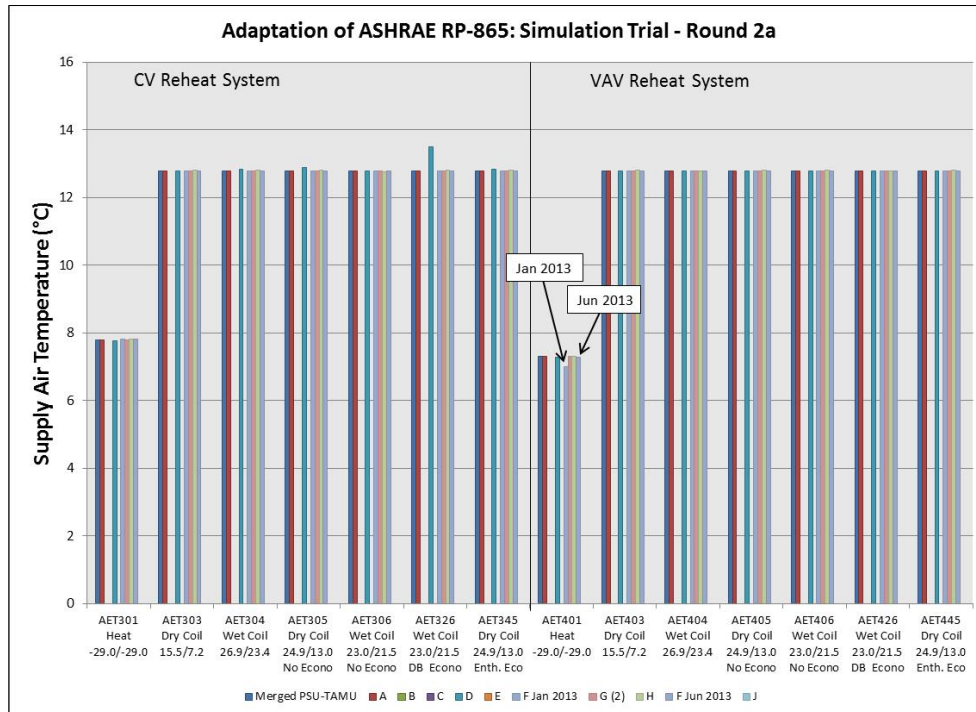
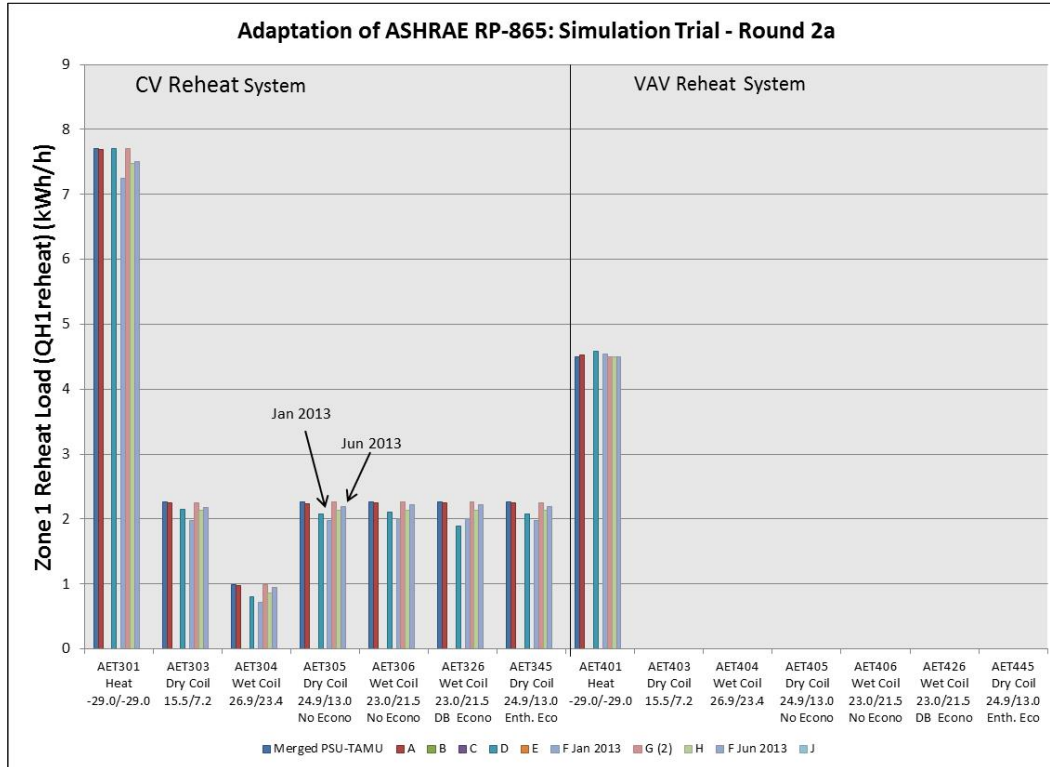


Figure 3-3. CV and VAV Supply Air Temperature, Jan–Jun 2013

### 3.4.4.4 Item 5e



**Figure 3-4. Zone 1 Reheat Coil Load, Jan–Jun 2013**

### 3.4.4.5 Input Revisions and Effect on Results

Input changes were made based on intermediate diagnostic output of Figures 3-1 through 3-4, and for consistency with test specification revisions. Result changes caused by correcting the input errors, indicated in Table 3-18, are based on the progression of EnergyPlus output during the simulation trials shown in Table 3-9. Final changes to results are for a minor adjustment to atmospheric pressure to match the specified value of 101325 kPa and had negligible effect on results (Henninger 2015).

**Table 3-9. EnergyPlus Total Cooling Coil Load Results, 2012 to 2015**

Total Cooling Coil Load					
Cases	Jun 6, 2012 (kWh/h)	% Difference	Jun 4, 2013 (kWh/h)	% Difference	Jan 14, 2015 (kWh/h)
AE103	0.745	2.71%	0.766	0.00%	0.766
AE104	5.644	0.56%	5.675	-0.04%	5.673
AE203	0.797	4.42%	0.833	0.00%	0.833
AE204	5.696	0.82%	5.743	-0.07%	5.739
AE205	1.877	1.88%	1.913	0.00%	1.913
AE206	2.693	1.82%	2.743	-0.16%	2.738
AE226	3.360	1.85%	3.422	-0.11%	3.418
AE245	2.187	0.93%	2.207	0.00%	2.207
AE303	6.048	7.08%	6.476	0.00%	6.476
AE304	16.223	3.56%	16.800	-0.13%	16.778
AE305	8.772	4.88%	9.199	0.00%	9.199
AE306	13.917	4.27%	14.511	-0.13%	14.493
AE326	19.691	5.52%	20.777	0.01%	20.780
AE345	9.073	4.81%	9.510	0.00%	9.510
AE403	1.584	-11.73%	1.399	0.00%	1.399
AE404	13.744	-0.24%	13.711	-0.17%	13.688
AE405	4.284	-4.34%	4.098	0.00%	4.098
AE406	8.625	-2.31%	8.426	-0.03%	8.423
AE426	9.253	-3.50%	8.929	0.02%	8.931
AE445	4.301	-3.62%	4.145	0.00%	4.145

### 3.4.4.6 Other

One further issue, not covered in the supplementary modeler report, is listed in Table 3-10.

**Table 3-10. Further Simulation Trial Issue Index, EnergyPlus**

Error Description	Reference
Latent gains throughout program (3 fixes)	Witte (2013)

During analysis by one of the test specification authors, an error was discovered in EnergyPlus in two routines that calculate the impact of return duct latent gains primarily from refrigerated display case under-case returns. The error was in the equation calculating the mass of added water where the air density, which would have been applied correctly for a volumetric flow rate, was mis-applied to the mass flow rate, resulting in a predicted humidity ratio that was not dimensionless. This bug has no impact on the current Airside HVAC test cases. However, its discovery led to a general review of the application of hf, hfg, and hg by the software developers, which found some calculations were using hfg instead of hg (or vice versa) and other calculations using density of moist air instead of dry air.

### 3.4.5 IES-VE

The software-developer modeling team submitted a pro-forma report (see Section 3.9, Appendix III-E), but not a supplementary modeler report. Discussion of issues summarized in Table 3-18 (see Section 3.6.1.2) are based on communications with the modelers during the simulation trials and described in the following sections. These issues are indexed in Table 3-11.

**Table 3-11. Simulation Trial Issues Index, IES-VE**

Error Description	Reference (following sections)
Zone latent load calculation	Section 3.4.5.1
Zone latent load hfg	Section 3.4.5.1, last paragraph
Outdoor air at standard air properties	Section 3.4.5.2

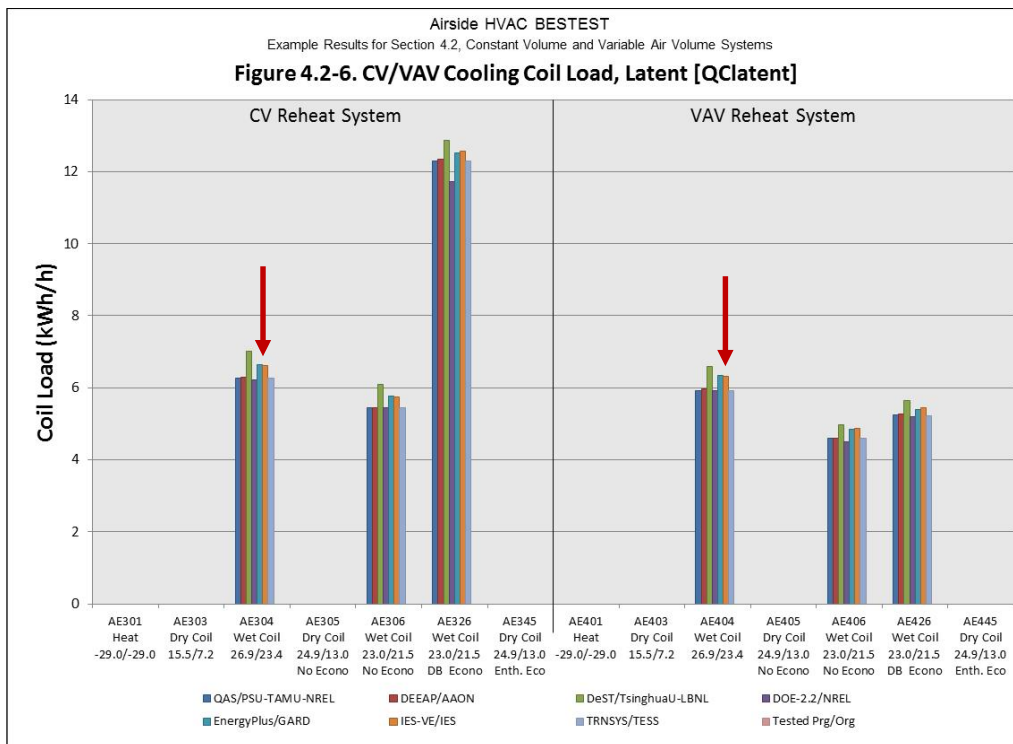
### 3.4.5.1 Zone latent load calculation

The following is excerpted from material distributed by the test specification authors to the software developer team in January 2015. (Neymark and Kennedy (2015), email communication) *[This is typical of material communicated to other working group participants during the simulation trials.]*

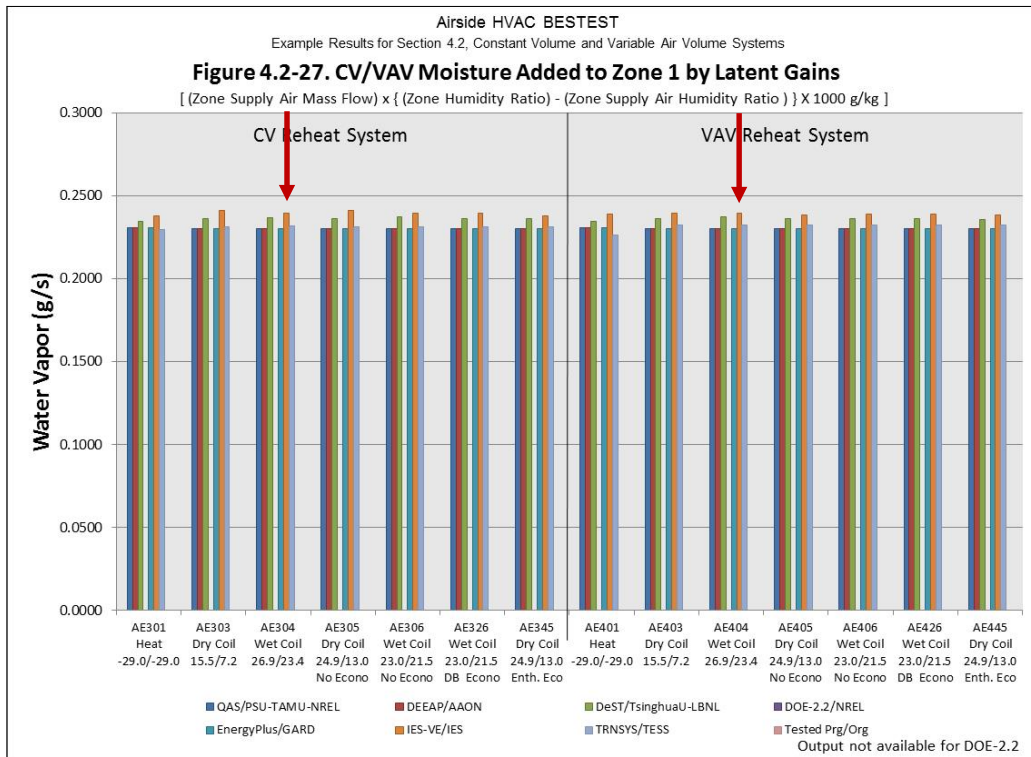
“Versus the QAS, the predicted latent cooling coil loads, shown in Figure 3-5, are high by up to 6%, although coil leaving RH (see the “RHcco” tab in the ResultsFCSZ.xlsm and ResultsCVVVV.xlsm files) is at 100% for the wet coil cases, indicating the model achieved zero bypass factor. Figure 3-6 shows the mass of water vapor added to Zone 1 of the CV and VAV systems. For developing the figure, this has been calculated from other reported results as:

$$\text{“(zone supply air mass flow rate)} \times ((\text{zone air humidity ratio}) - (\text{supply air humidity ratio}))$$

“Program “IES-VE” is indicated as adding more moisture than the QAS and most other programs. This could explain some of the difference in predicted latent cooling loads, or it could result from some other factor such as limited output precision of the humidity ratios (as occurs for Program “DOE-2.2/NREL”). One possible explanation is that some programs use the enthalpy of vaporization (hfg) while others use the enthalpy of saturated vapor (hg) to convert the latent gains to moisture.”



**Figure 3-5. CV/VAV Latent Cooling Load**



**Figure 3-6. Moisture Added to Zone 1 by Latent Gains, CV/VAV**

The effect on total cooling coil load from the first attempt at addressing this issue was evaluated by comparing IES-VE results submitted Feb 20, 2015, versus Oct 1, 2013, as shown in Table 3-12.

**Table 3-12. Cooling Coil Load Changes, IES-VE, Feb 20, 2015 vs. Oct 1, 2013**

Cases	Total Cooling Coil Load			Latent Cooling Coil Load		
	Oct 1, 2013 (kWh/h)	% Difference	Feb 20, 2015 (kWh/h)	Oct 1, 2013 (kWh/h)	% Difference	Feb 20, 2015 (kWh/h)
AE103	0.762	0.00%	0.762	0.000		0.000
AE104	5.605	0.04%	5.607	2.125	0.09%	2.127
AE203	0.830	0.00%	0.830	0.000		0.000
AE204	5.673	0.04%	5.675	2.125	0.14%	2.128
AE205	1.915	0.00%	1.915	0.000		0.000
AE206	2.696	0.00%	2.696	1.000	0.00%	1.000
AE226	3.380	-0.09%	3.377	1.829	-0.22%	1.825
AE245	2.210	0.00%	2.210	0.000		0.000
AE303	6.483	0.00%	6.483	0.000		0.000
AE304	16.655	0.04%	16.662	6.623	0.09%	6.629
AE305	9.195	0.00%	9.195	0.000		0.000
AE306	14.405	-0.01%	14.403	5.755	-0.09%	5.750
AE326	20.669	-0.07%	20.654	12.573	-0.12%	12.558
AE345	9.529	0.00%	9.529	0.000		0.000
AE403	1.399	0.00%	1.399	0.000		0.000
AE404	13.656	-0.08%	13.645	6.311	0.03%	6.313
AE405	4.111	0.00%	4.111	0.000		0.000
AE406	8.437	-0.02%	8.435	4.869	-0.02%	4.868
AE426	8.970	0.02%	8.972	5.455	0.02%	5.456
AE445	4.160	0.00%	4.160	0.000		0.000



Upon further review, as the initial change did not address most of the issue, the software developer noted that for calculating zonal moisture addition they are using an enthalpy of vaporization value of 2450 kJ/kg, and they plan to consider revising this assumption for their next software version (Aird 2015a).

### 3.4.5.2 Outdoor air at standard air properties

Another issue communicated to the software developer team (Neymark and Kennedy (2015), email communication) relates to mass flow differences as follows:

“Outdoor air mass flow is 0–1% higher than the quasi-analytical solution (QAS) result in heating cases and ~2% higher in all non-economizer cooling cases (e.g., see Part IV, Section 4.4, Table 4.4-11). Supply air mass flow is shown in Figures 3-7 and 3-8. Versus the QAS, supply air mass flow is 3% high in the FC and SZ system heating cases, ~1% high in the VAV system heating case, and 4% low in the CV system heating case and 2% low in the CV cooling cases. These differences result from Program IES-VE calculating supply, exhaust, and outdoor air mass flow rates at a single set of standard air conditions for all test cases, which results in mass flow rates that do not change among test cases. This issue impacts all coil loads, and inhibits diagnosis of any other issues within this range of difference. A constant-air-density approach (air density calculated at a single set of air conditions) is also applied by some other participant programs.

“Results differences versus the quasi-analytical solutions that likely stem from this include:

- “The 24% high CV system preheat coil load in Case AE301 (see Figure 3-9); other programs applying various constant-air-density approaches exhibit similar disagreement. This may be a result of the slightly high mass flow rate of cold outdoor air combined with a low system supply air mass flow rate, which results in the mixed air temperature being lower and preheat coil loads larger.
- “The 2%–4% low CV system sensible cooling load (see Part IV, Section 4.4, Figure 4.4-5).
- “The 3%–8% low CV system zone reheat coil load in all CV cases (see Part IV, Section 4.4, Figure 4.4-7).”

Based on Figure 3-9, the software developers have indicated they are considering a revision for the next version of their program to calculate air density at local system conditions, similar to the method in the QAS (Aird 2015b).

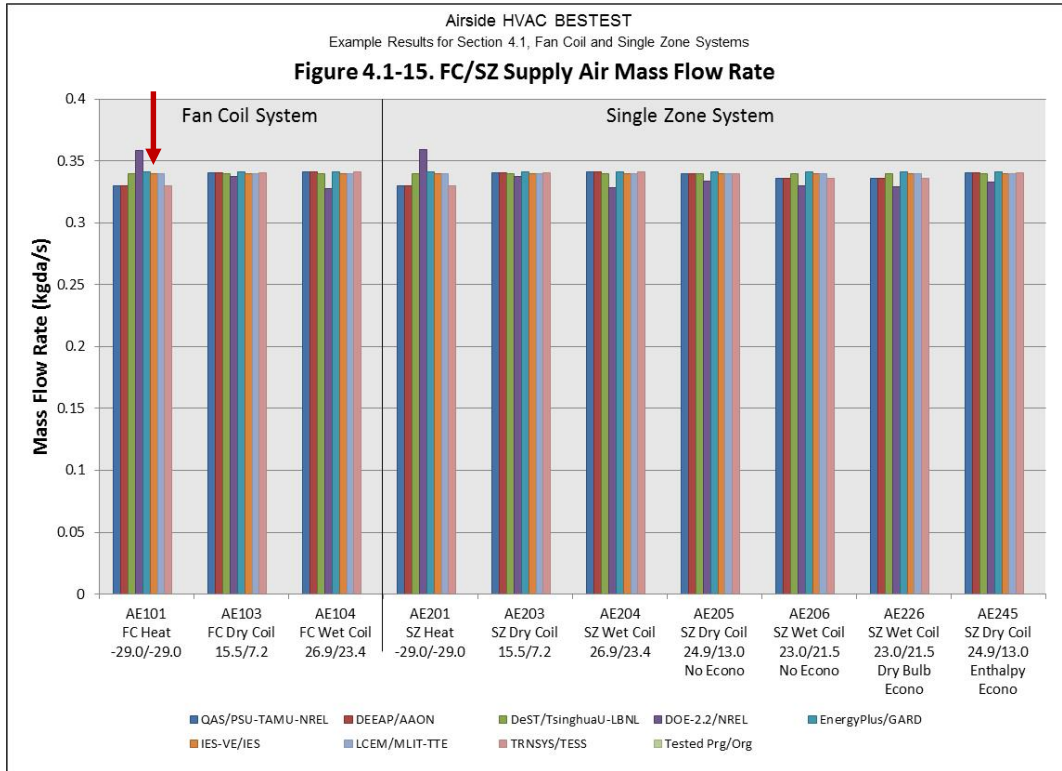


Figure 3-7. FC/SZ Supply Air Mass Flow Rate

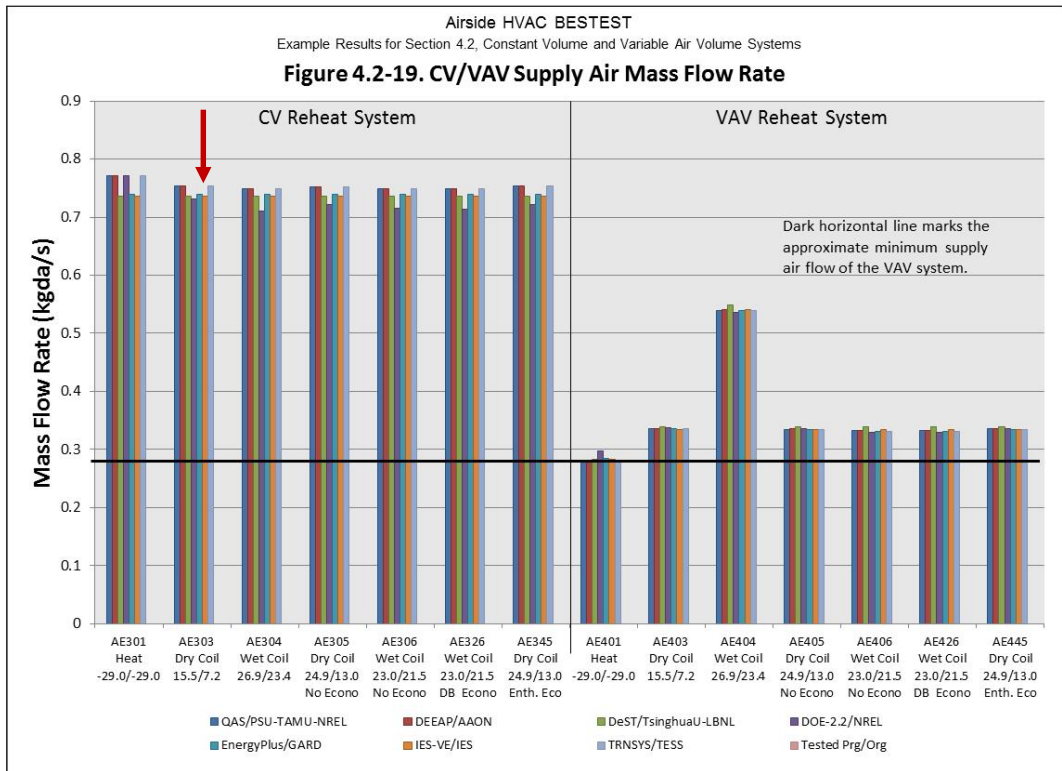
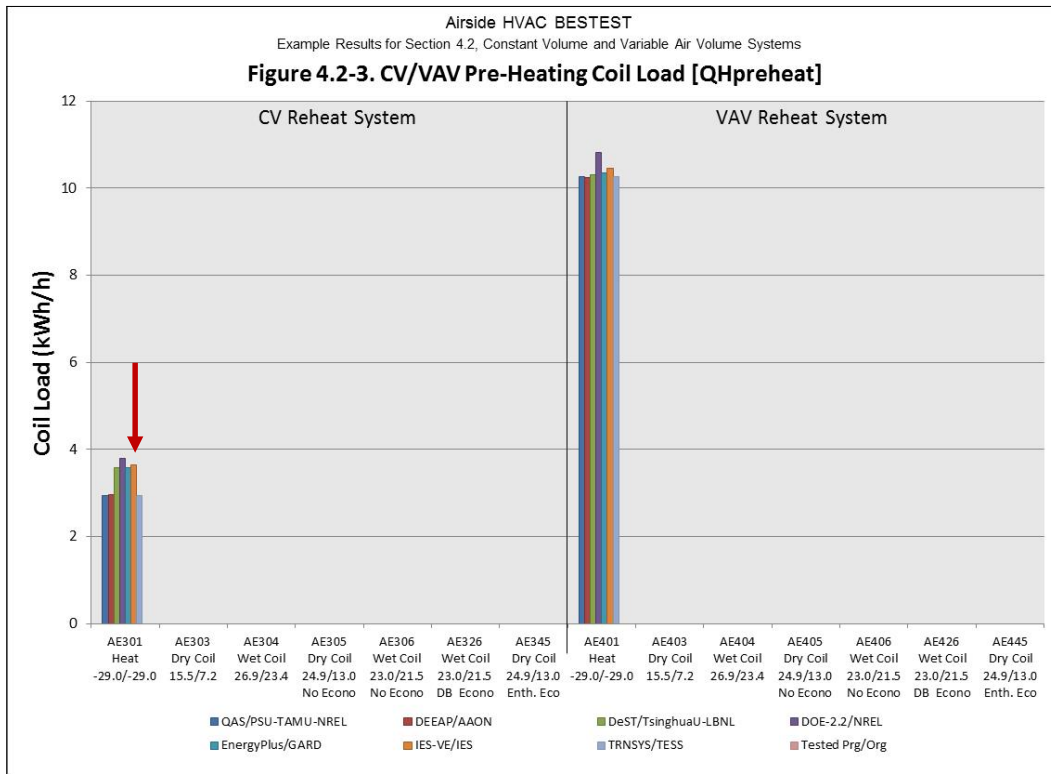


Figure 3-8. CV/VAV Supply Air Mass Flow Rate



**Figure 3-9. CV/VAV Preheat Coil Load**

### 3.4.6 LCEM

The issues summarized in Table 3-18 (see Section 3.6.1.2) are described in the participant’s supplementary modeler report (see Section 3.9, Appendix III-F). An index of the error and selected results discussions in that report is listed in Table 3-13.

**Table 3-13. Simulation Trial Issues Index, LCEM**

Issue Description	Cross Reference (Appendix III-F)
One coil model failed (other models ok)	Section 4, first paragraph
<i>Input:</i> Supply/return fan temperature rise	Section 4, remaining paragraphs
Constant system air density	Section 5, Subsection “Mass Flow Differences”

Input changes were made based on intermediate diagnostic output. Results changes caused by correcting the fan temperature rise inputs, indicated in Table 3-18, are based on the progression of LCEM output during the simulation trials shown in Table 3-14.

**Table 3-14. LCEM Total Cooling Coil Load Changes, 2014 to 2015**

<b>Total Cooling Coil Load</b>			
<b>Cases</b>	<b>20Feb2014 (kWh/h)</b>	<b>16Apr2015 (kWh/h)</b>	<b>% Difference</b>
<b>AE103</b>	0.7320	0.7626	<b>4.2%</b>
<b>AE104</b>	5.5963	5.6456	<b>0.9%</b>
<b>AE203</b>	0.8009	0.8304	<b>3.7%</b>
<b>AE204</b>	5.6647	5.7133	<b>0.9%</b>
<b>AE205</b>	1.8818	1.9110	<b>1.6%</b>
<b>AE206</b>	2.6473	2.6988	<b>1.9%</b>
<b>AE226</b>	3.2952	3.3911	<b>2.9%</b>
<b>AE245</b>	2.1714	2.2021	<b>1.4%</b>

### 3.4.7 TRNSYS

Issues summarized in Table 3-18 (see Section 3.6.1.2) are described in the participant’s supplementary modeler report (see Section 3.9, Appendix III-G, Subsection 4). An index of the error descriptions in that subsection is listed in Table 3-15.

**Table 3-15. Simulation Trial Issues Index, TRNSYS**

<b>Error Description</b>	<b>Cross Reference (Appendix III-G, Section 4)</b>
<i>Inputs:</i> Trial “2a” spec revisions (see Section 3.3): CV/VAV	“Early Round Input Revisions ....”
<i>Input:</i> Constant system pressure	“Delta AE201 – AE101 ....”
Condensate characterization	“Latent Cooling Loads”
Zone latent load from hfg	“Latent Cooling Loads”

The effect of input changes to the CV and VAV systems related to “Trial 2a spec revisions” (see Table 3-15) are evident from comparing total coil load results for Jan 12, 2013 versus Jun 4, 2012 in Table 3-16. After Jan 2013, additional input changes and program improvements (also see Table 3-15) were made based on intermediate diagnostic output, or for consistency with test specification clarifications. Results changes caused by correcting those errors, indicated in Table 3-18, are based on comparison of TRNSYS’ Nov 18, 2014 results versus Jan 12, 2013 results shown in Table 3-16.

**Table 3-16. TRNSYS Cooling Coil Load Results, 2012 to 2014**

Total Cooling Coil Load						Latent Cooling Coil Load			
Cases	Jun 4, 2012 (kWh/h)	% Difference	Jan 12, 2013 (kWh/h)	% Difference	Nov 18, (kWh/h)	Cases	Jan 12, 2013 (kWh/h)	% Difference	Nov 18, 2014 (kWh/h)
AE103	0.764	0.00%	0.764	2.10%	0.7800233	AE103	0		0
AE104	5.573	0.00%	5.573	-0.46%	5.5475416	AE104	2.095	-1.94%	2.05435989
AE203	0.832	0.00%	0.832	2.46%	0.852491	AE203	0		0
AE204	5.64	0.00%	5.64	-0.27%	5.6248592	AE204	2.095	-1.94%	2.05435409
AE205	1.89	0.00%	1.89	1.21%	1.912806	AE205	0		0
AE206	2.678	0.00%	2.678	-0.26%	2.670914	AE206	0.999	-3.14%	0.96765873
AE226	3.384	0.00%	3.384	-0.85%	3.3553451	AE226	1.841	-2.49%	1.79508705
AE245	2.185	0.00%	2.185	1.10%	2.2090263	AE245	0		0
AE303	6.217	7.27%	6.669	0.85%	6.7255393	AE303	0		0
AE304	15.906	5.20%	16.733	-0.94%	16.576497	AE304	6.394	-2.07%	6.26153312
AE305	8.859	5.06%	9.307	0.71%	9.3734179	AE305	0		0
AE306	13.693	5.89%	14.5	-0.98%	14.358428	AE306	5.568	-2.24%	5.44353742
AE326	19.706	7.15%	21.114	-1.60%	20.775744	AE326	12.552	-1.97%	12.3052644
AE345	9.188	5.00%	9.647	0.62%	9.7072744	AE345	0		0
AE403	1.523	-6.17%	1.429	1.30%	1.4475745	AE403	0		0
AE404	13.326	1.19%	13.484	-1.31%	13.307624	AE404	6.042	-2.00%	5.92139114
AE405	4.152	-2.26%	4.058	0.81%	4.0909896	AE405	0		0
AE406	8.313	0.19%	8.329	-1.32%	8.2187371	AE406	4.672	-1.77%	4.58915365
AE426	8.984	-0.49%	8.94	-1.52%	8.8045385	AE426	5.315	-1.80%	5.21958923
AE445	4.194	-1.98%	4.111	0.82%	4.1448235	AE445	0		0

### 3.5 Interpretation of Results

#### 3.5.1 Comparing Simulation Results to Quasi-Analytical Solution Results

Test cases were adapted from ASHRAE RP 865 (Yuill and Haberl 2002) that focus on system airflow and heat and mass balance. These are steady-state analytical verification tests, where simulation results are compared to a quasi-analytical solution (QAS) and to each other. The test systems include the Four-Pipe Fan Coil (FC), Single-Zone Air Conditioner (SZ), Constant Volume Terminal Reheat (CV), and Variable Air Volume Reheat (VAV) systems. The FC system is the simplest of these. It is a single-zone system with heating and cooling coils, zone air exhaust, and limited outdoor air (no economizer control), and it does not include a return air fan. In these test cases, the SZ system adds an economizer and a return air fan; the CV system further applies multiple (two) zones, system supply air temperature control, and terminal reheat coils; and the VAV system further applies a variable airflow supply fan and terminal zone supply air dampers. Some test specification details taken from ASHRAE RP 865, for compatibility with their original spreadsheet solutions, are:

- All systems apply a draw-through fan.
- Outdoor airflow replaces specified exhaust airflow, in test cases without economizer control.

The test cases are conducted at five different sets of steady-state outdoor and zone conditions in heating, dry-coil cooling, and wet-coil cooling modes, and with temperature and enthalpy economizer outdoor air control strategies applied to selected conditions. Primary compared output for these test cases includes coil sensible, latent and total loads; zone sensible and latent loads; and cooling-coil leaving-air relative humidity. Additional diagnostic outputs at various points in the systems include dry-bulb temperature (and the ability to isolate fan heat effects), humidity ratio, specific volume, enthalpy, and mass flow rate. For these in-depth cases, plant energy use related to coil loads and fan electricity consumption is not considered.

The tables and graphs in Part IV present the final results from all the simulation programs and the QAS used in this study. QAS detailed documentation, developed for this project, is presented in Part II. The QAS results constitute a reliable set of theoretical results: a secondary mathematical truth standard, as defined in ASHRAE Standard 140-2014 (ANSI/ASHRAE 2014), for cases AE101–AE445. Availability of the QAS reference helped participants identify modeling issues and test specification ambiguities. The QAS provides the basis for a narrower range of results disagreement than is generally available from software-to-software comparative tests without any mathematical truth standard (Judkoff and Neymark 2006; ASHRAE 2013, Ch. 19). The primary purpose of including simulation results along with QAS results in Part IV is to allow simulationists to compare the relative agreement (or disagreement) of simulation results with the QAS results. Perfect agreement among simulations and the QAS is not necessarily expected because many programs contain simplifying assumptions to ease the calculation burden (e.g., constant air density), and the QAS contains idealized simplifying assumptions (in order to be solved analytically) that cannot always be exactly reproduced by some simulation programs that are conceived and hardcoded with more realistic assumptions. The Part IV results give an indication of the level of agreement that is possible between simulation results and the QAS results.

The secondary mathematical truth standard of the QAS results does not represent absolute truth. It is important to reiterate the difference between a mathematical truth standard or secondary mathematical truth standard and an absolute truth standard. When applying mathematical truth standards, we only test the solution process for a model, not the appropriateness of the model itself; that is, we accept the given underlying physical assumptions while recognizing that these assumptions represent a simplification of physical reality. For example, a one-dimensional conduction model may be properly solved mathematically, but inappropriate where two-dimensional conduction dominates. By contrast, an approximate truth standard from an experiment tests both the solution process and the appropriateness of the model within the experimental uncertainty. The ultimate or absolute validation truth standard would be a comparison of simulation results with data from a *perfectly performed* empirical validation experiment, with all simulation inputs perfectly defined. In reality, an experiment is performed and the experimental object specified within some acceptable range of uncertainty. Such experiments are possible, but are fairly expensive. We recommend developing a set of empirical validation experiments in the future.

One must rely on engineering judgment to assess the significance of results that disagree. For simulation results that disagree significantly with the QAS results, investigating the source(s) of the difference(s) is worthwhile, but the existence of a difference does not necessarily mean that a program is faulty. However, our collective experience in this and previous BESTEST tasks has indicated that when programs show disagreement with quasi-analytical solution results, other simulation results, or both, we often find a bug, questionable algorithm, documentation problem, input error, or a combination of such issues. Therefore, investigating the sources of differences is strongly recommended.

Because of iterative correction of input errors, software bugs, and clarification of the test specification, the agreement among simulation results improved with each iteration of the simulation trials. Improvements to simulation programs or simulation inputs made by participants were required to have a mathematical and a physical basis, and to be applied consistently throughout the test cases. Arbitrary modification of a simulation program's input or internal code to more closely match a given set of results was not allowed. All improvements were requested to be documented in modeler reports. Improvements to simulation models are summarized in Table 3-18 (see Section 3.6.1.2), with additional details in Section 3.4 and in the participant modeler reports of Section 3.9.

Improvements to the simulation models are evident when the initial results set is compared to the final results set. Initial simulation results for total coil loads obtained are shown for the FC and SZ systems in Figure 3-10 and for the CV and VAV systems in Figure 3-12 (abbreviations along these figures' x-axes

are shorthand for the case descriptions given in Part I). Figures 3-10 and 3-12 show each participant's results after the first "blind" round of simulations, before the QAS results were distributed to the working group, for most of the programs. Two late-entering participating programs (IES-VE and LCEM) were allowed an initial non-blind round to accelerate their simulation trials, and asked to document intermediate revisions to their input files or programs in their modeler reports; these participants also benefitted from first seeing the test specification after revisions engendered by the initial simulation trials. The late entering participants were also a benefit to the test specification vetting process, as this introduced new test specification reviewers after its initial revisions. Initial results for the FC and SZ systems were received as follows:

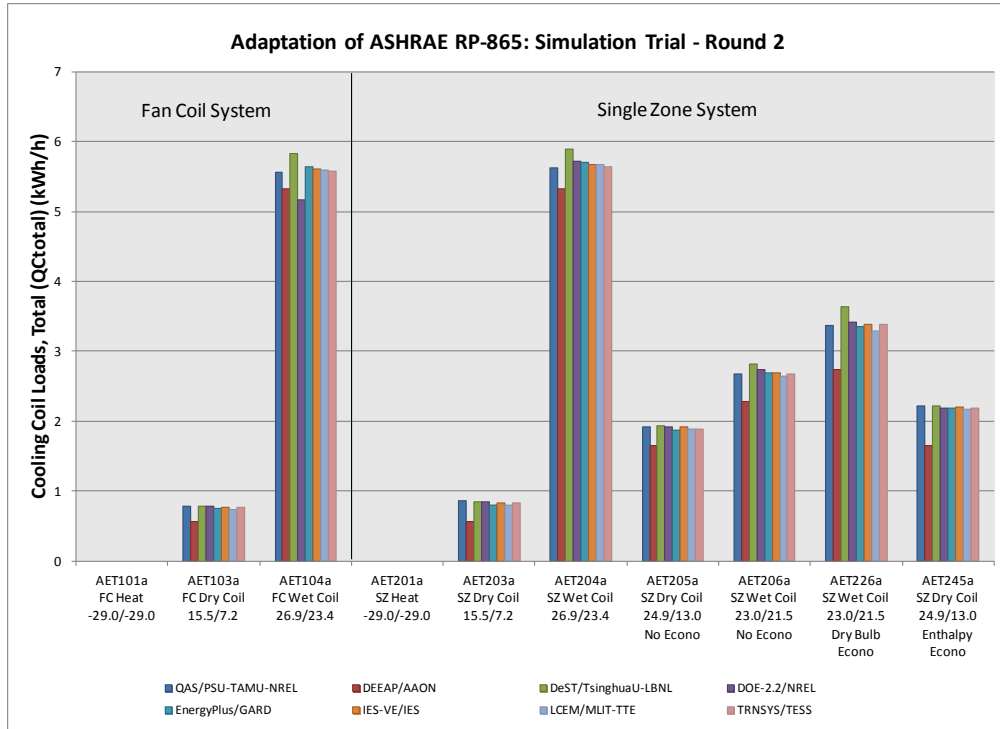
- DEEAP: Jun 11, 2012
- DeST: Jun 12, 2012
- DOE-2.2: Jun 20, 2012
- EnergyPlus: Jan 3, 2012 FC and SZ; Jun 7, 2012 CV and VAV
- IES-VE: Nov 7, 2013
- LCEM: Feb 20, 2014
- TRNSYS-17: Jun 8, 2012.

The results shown in Figure 3-10 indicate that for the FC and SZ systems there was initially 1% to 19% *average* disagreement for a given program versus the QAS results. The results shown in Figure 3-12 indicate that for the CV and VAV systems there was initially 2% to 37% average disagreement for a given program versus the QAS results. The additional complexity of specifying the CV and VAV systems (multi-zone with reheat), and inputting them in the models, generated more initial disagreements than for the simpler FC and SZ systems. This emphasizes the importance of testing the different systems.

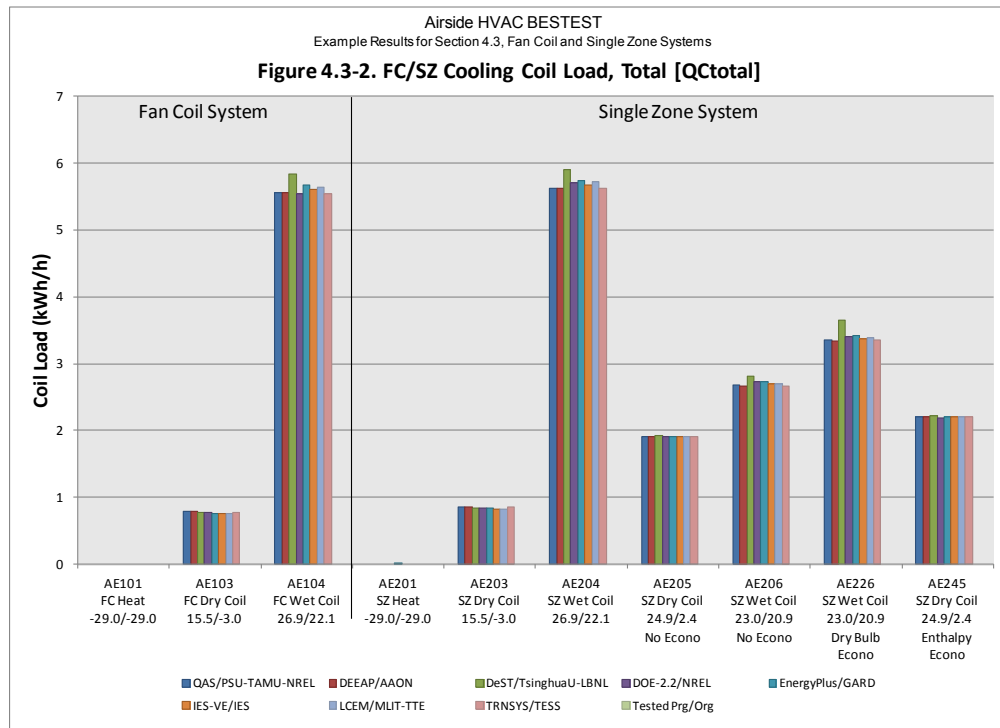
The final set of total coil load results for all the simulations and the QAS are shown in Figure 3-11 for the FC and SZ systems, and in Figure 3-13 for the CV and VAV systems (abbreviations along these figures' x-axes are the same as in Figure 3-10, except the temperature values are ODB/ODP instead of ODB/OWB). After correcting software errors and other model improvements using the diagnostic output, the mean of all simulated results of total cooling coil load for the tested programs are, on average, within 1.3% of the QAS results, with average variations among the test cases for a given program of up to 3%.

Final ranges of disagreement are further summarized in Table 3-17 for predictions of various outputs. This range of disagreement for each case is based on the difference between each simulation result versus the QAS result, divided by the QAS result. The outputs are disaggregated for each system type.

Greater cooling coil load disagreements occur in the wet-coil cases and appear to be related to difficulty modeling the 0-bypass factor idealization, especially for DeST, and in some cases for EnergyPlus. In the cooling cases, there are also 3%–4% differences (see Part IV) in the amount of moisture added to the zones by latent gains in DeST and IES-VE (this output is not available or cannot be calculated with sufficient precision in DOE-2.2).

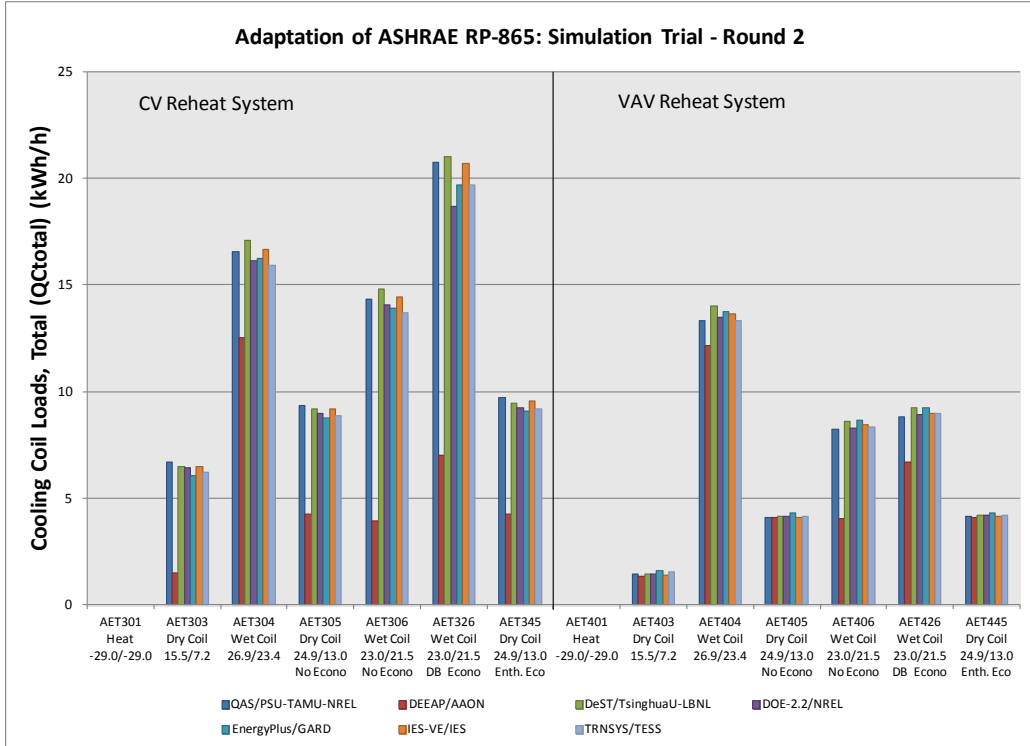


**Figure 3-10. Airside HVAC BESTEST—total coil load, FC and SZ, before BESTESTing**  
 (Abbreviations along x-axis describe test cases, numeric values are ODB/OWB; see Part I for details.)

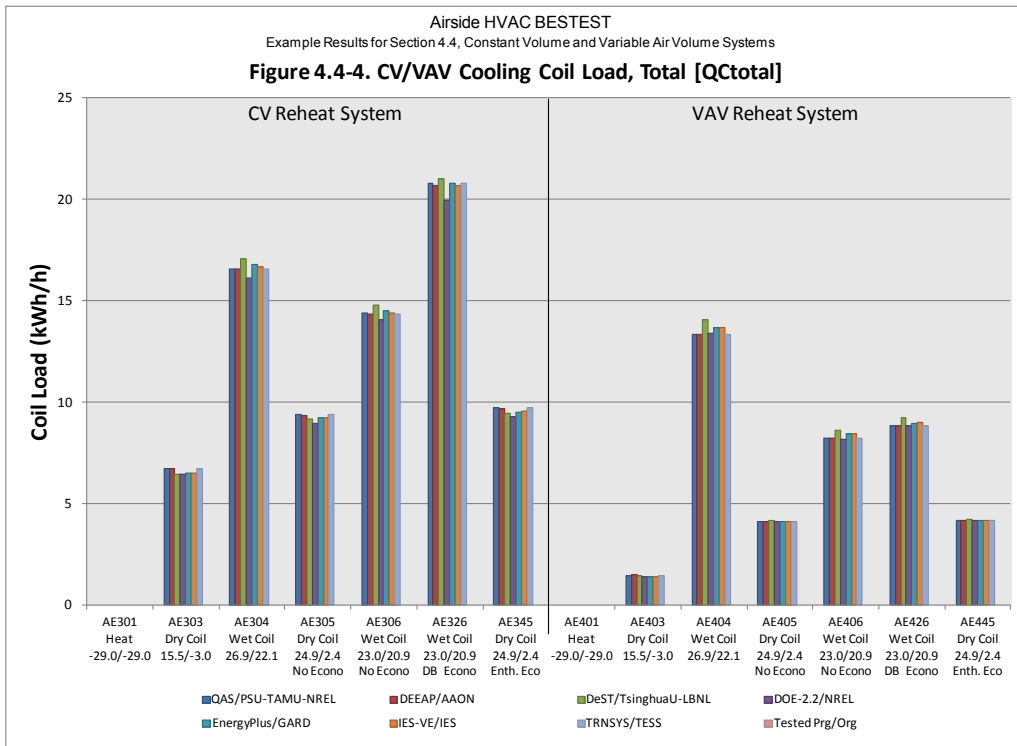


**Figure 3-11. Airside HVAC BESTEST—total coil load, FC and SZ, after BESTESTing**  
 (Abbreviations along x-axis describe test cases, numeric values are ODB/ODP; see Part I for details.)





**Figure 3-12. Airside HVAC BESTEST—total coil load, CV and VAV, before BESTESTing** (Abbreviations along x-axis describe test cases, numeric values are ODB/OWB; see Part I for details.)



**Figure 3-13. Airside HVAC BESTEST—total coil load, CV and VAV, after BESTESTing** (Abbreviations along x-axis describe test cases, numeric values are ODB/ODP; see Part I for details.)

**Table 3-17. Ranges of Disagreement<sup>a</sup> among Simulation Results**

<b>System</b> <b>Load Type</b>	<b>Fan Coil</b>	<b>Single Zone</b>	<b>Constant Volume with Reheat</b>	<b>Variable Air Volume with Reheat</b>
Total Cooling Coil	-3% to 5% <sup>b</sup>	-3% to 8% <sup>b</sup>	-5% to 3%	-4% to 5%
Sensible Cooling Coil	-4% to 0%	-4% to 2%	-5% to 0%	-4% to 1%
Latent Cooling Coil	-2% to 12% <sup>c</sup>	-1% to 14% <sup>c</sup>	-5% to 12% <sup>d</sup>	-2% to 11% <sup>d</sup>
Heating/PreHeat Coil (heating cases AEn01)	0% to 4% <sup>e</sup>	0% to 4% <sup>e</sup>	0% to 29% <sup>f</sup>	0% to 5% <sup>e</sup>
Reheat Coils (AEn01)	N/A	N/A	-4% to 0%	0% to 3%
Total (Preheat + Reheat) Heating Coil Load (heating cases AEn01)	N/A	N/A	0% to 4%	0% to 4%
Reheat Coils (cooling cases)	N/A	N/A	-20% to 0% <sup>g</sup>	N/A
Net Sensible Cooling (cooling coil sensible minus reheat coil)	N/A	N/A	-5% to 2%	N/A

<sup>a</sup> Range of percentage disagreement among the programs for each output type, among the cases for each system, is based on the difference between each simulation result versus the QAS result, divided by the QAS result.

<sup>b</sup> DeST has greatest disagreement. This relates to difficulty modeling the BF = 0 idealization in the wet coil cases; other programs are -3% to 2%.

<sup>c</sup> DeST, LCEM, and EnergyPlus have greatest disagreements. In DeST and EnergyPlus this may relate to difficulty modeling BF = 0; LCEM has low sensible and high latent loads, and balances on total, so may have issue with sensible/latent load attribution; other programs are -2% to 4%.

<sup>d</sup> DeST has greatest disagreement. This relates to difficulty modeling the BF = 0 idealization in the wet coil cases; other programs are -2% to 7%.

<sup>e</sup> DOE-2.2 has greatest disagreement, which may arise from using the mixed air (preheat coil outlet) density throughout the system; other programs are -1% to 2%.

<sup>f</sup> DeST, DOE-2.2, EnergyPlus, IES-VE are all > 20% in Case AE301, less difference for other cooling cases. The difference is related to application of constant air density throughout the system for these programs; this preheat coil load disagreement is compensated by lower reheat coil loads for these programs. DEEAP and TRNSYS are 0% to 1% disagreement.

<sup>g</sup> DeST and, DOE-2.2 are -11% to -20%, which may relate to some combination of difficulty modeling the BF = 0 idealization and their constant system air density assumptions; other programs are -8% to 0% disagreement. In general, lower reheat coil loads coincide with the lower sensible coil loads where they occur so that the system net sensible cooling load has a similar range of % disagreement as that for sensible cooling coil load.

Another cause of disagreement is that many of the tested programs apply simplifications to the air density calculations, such as applying constant air density (DeST, EnergyPlus, IES-VE, and LCEM) or applying an air density calculated from air conditions at a single location in the system to all other locations (DOE-2). The disagreements that may arise from various constant air density assumptions are magnified most in the CV system heating case (AE301), where the outdoor air dry-bulb temperature represents a low extreme (-29°C [-20.2°F]), and both preheat and reheat coils are applied. Reheat coil disagreements in the CV-system (AE300-series) cases are consistent with balancing the preheat-coil load disagreements in Case AE301 and the sensible cooling coil load disagreements in the other cases. Programs making constant air density assumptions that perform well in the current test cases may not perform as well with

different fan placement (blow-through versus draw-through) or with outdoor airflow specified directly as a volumetric flow. This is because of potentially greater mass flow sensitivity to ambient (boundary) conditions with these adjustments, which a standard-air-conditions assumption would not address.

DEEAP and TRNSYS are within 1% of the QAS for all coil load results, indicating that the models developed with these programs were able to better match the assumptions of the test cases.

Based on results after several simulation trial iterations and resulting model improvements, the programs appear reliable for modeling these air distribution systems under the conditions tested, although use of constant system air density by some simulation programs causes larger modeling disagreements at extreme-low outdoor air temperatures for the specified CV system. Without additional test cases having realistic hourly varying weather data, we do not have a means to scale, relative to typical annual simulations, the effect of disagreements identified in these in-depth diagnostic steady-state analytical verification tests. For future work we recommend developing an airside HVAC comparative test suite with annual hourly varying weather data, analogous to the HVAC BESTEST Volume 2 (Neymark and Judkoff 2004) working-fluid side test suite.

The QAS results may be used as a reference or benchmark against which other software can be tested. In the future, additional air distribution system comparative tests, with or without analytical verification, may explore modeling with less idealized testing parameters than were needed for the current set of steady-state analytical verification tests—e.g., more realistic coil bypass factors, typical (non-constant/artificial) hourly varying weather data, etc.—as discussed in the next section.

### **3.5.2 Test Cases for Future Work**

We suggest that additional work related to model testing and validation, outlined in the sections that follow, be considered.

#### **3.5.2.1 Additional Airside HVAC Equipment Test Cases**

This project developed a set of idealized in-depth diagnostic test cases for HVAC air distribution system models. During this project, participants discussed a number of important test case configurations that could not be included with the current test cases because of resource and project completion timing constraints. These test cases, some of which could be developed as Volume 2 of the Airside HVAC system tests, include:

- Realistic bypass factor  $> 0$  (coil outlet relative humidity  $< 100\%$ ); e.g.,  $BF = 0.05$  with accompanying coil details (e.g., apparatus dew point, geometry).
- Apply typical hourly weather data in one or more climates with hot humid summers and/or cold winters. In addition to providing an interesting range of weather dynamics, this would allow scaling of disagreements among computer programs with respect to annual and peak energy use and cost.
  - Check and revise dry-bulb and enthalpy economizer controls, as needed so that they provide net energy savings.
    - Consider specifying operation at lower dry-bulb temperatures.
    - Check for hours in free cooling mode.
    - Check control modeling with selected hourly output requirements.
  - Specify daily hourly output to analyze system interaction with solar gains fluctuations.
- Specify minimum outdoor air volume flow directly (instead of as replacing exhaust airflow), to check mass balance with different types of boundary conditions.

- Evaluate fan energy consumption and specify more detailed fan performance data.
- Duct heat gain/loss.
- Transition incrementally from the current cases to more realistic cases to employ the analytical verification foundation of the current test cases.
  - Consider applying the new cases to a subset of the current systems—e.g., the CV and VAV (multi-zone with reheat) systems—that sufficiently cover the physics and control aspects of the other modeled systems.

Other cases to consider that were discussed by various project participants at meetings and in their modeler reports or other communications include:

- Preheat coil in outdoor air stream (instead of in mixed air stream)
- Blow-through versus draw-through fan
- Fan motor in air stream
- Variations to VAV system:
  - One zone requiring heating, while other zone simultaneously requires cooling
  - Fan-powered terminals
  - Adjust VAV terminal minimum flow to be different from zone exhaust air flow
  - Other control schemes, e.g., reverse-acting terminals, supply temperature reset
- Other systems from RP 865 (Yuill and Haberl 2002): four-pipe induction unit, variable air volume dual duct, constant volume dual duct
- Duct leakage
- Altitude (atmospheric pressure) variation
- Uniform psychrometric calculations
- System and control response to transient conditions applying sub-hourly (e.g., 10-minute or 30-minute) interval analysis
  - More realistic thermostat controls (e.g., applying hysteresis, proportional control)
  - More realist coil controls (e.g., throttling range, supply air temperature reset based on warmest zone temperature)
- Water-side (coil) parametric variations
  - Also plant model tests: boiler, chiller, piping, cooling tower, etc
- Dedicated outside air system integrated with current systems
- Airflow calculations applying system pressure balancing, including more detailed fan performance curves and duct friction pressure drop [note: air pressure balancing seems to be beyond the current capability of most of the tested whole-building energy simulation programs].

The above listing is indicative of the large number of parametric and system configuration variations that are possible with HVAC air distribution systems. Prioritization of new test cases should have the goals of addressing fundamental and commonly applied modeling physics not addressed in the current cases, and of balancing realism versus the ability to diagnose results differences. The development of a set of more realistic cases with realistic weather data allows scaling of disagreements with respect to annual and peak energy use and cost. This follows the development precedent of HVAC BESTEST Volume 2 (Neymark and Judkoff 2004) comparative tests for modeling the working-fluid side of the cooling coil, which are based on the analytical verification tests of HVAC BESTEST Volume 1 (Neymark and Judkoff 2002).

### 3.5.2.2 Additional Building Thermal Fabric and Mechanical Equipment Test Cases

Additional building thermal fabric and mechanical equipment model test cases for consideration to be developed in the future are summarized in:

- *HVAC BESTEST Volume 2* (Neymark and Judkoff 2004, Section 2.5.2)

- *IEA BESTEST In-Depth Diagnostic Cases for Ground Coupled Heat Transfer Related to Slab-On-Grade Construction* (Neymark and Judkoff et al. 2008a, Section 2.5.4.1).
- *IEA BESTEST Multi-Zone Non-Airflow In-Depth Diagnostic Cases: MZ320 – MZ360* (Neymark and Judkoff et al. 2008b, Section 2.5.3.1).

## 3.6 Conclusions and Recommendations

### 3.6.1 Conclusions

#### 3.6.1.1 Major Accomplishments

The major accomplishments of this project were:

- Extended the BESTEST procedures to include in-depth diagnostic analytical verification test cases for the air distribution side of typical HVAC equipment, based on ASHRAE RP 865 (Yuill and Haberl 2002).
- Developed clear, unambiguous test specifications such that a variety of state of the art building simulation programs with a variety of input structures and modeling approaches can perform the tests.
- Developed quasi-analytical solution (QAS) results for all test cases, by comparing, reconciling, and merging the two original external spreadsheet solutions from RP 865, as described in Part II (see Sections 2.2.2 and 2.3).
- Improved accuracy to, or identified potential improvements for, all but one of the models that participated in the simulation trials of the test cases:
  - Seventeen program errors were diagnosed and fixed.
  - Initial total cooling coil load average disagreement ranges for each program of 1% to 19% for the FC and SZ systems and 2% to 37% for the CV and VAV systems were reduced to 0% to 3% for the FC, SZ, and VAV systems and to 0% to 4% for the CV system. This was accomplished by applying the diagnostics of the test cases to expose problems with the models and ambiguities in the original RP 865 test specification.
- Developed a set of national and international simulation results representative of the range of legitimate modeling differences for the current state of the art in whole-building energy simulation computer programs.
- Developed a new test suite suitable to initiate the ANSI/ASHRAE process for inclusion in Standard 140.

#### 3.6.1.2 Summary of Findings

This project built on ASHRAE RP 865 (Yuill and Haberl 2002) with the intent to include selected test cases in a future version of ASHRAE Standard 140 (current version is ANSI/ASHRAE 2014). The original research provided a technically sound conceptual starting point for the test specifications along with two separately developed external spreadsheets intended as quasi-analytical solutions. However, from this starting point there was substantial effort to develop and field test an unambiguous test specification usable by most state of the art whole building energy software and suitable for Standard 140. Also the two external spreadsheet solutions contained differences which were reconciled and then merged into the final QAS.

The test specifications were field-tested with a number of building-energy simulation programs from around the world. This method has proven effective at vetting test specifications and isolating the sources of predictive differences among simulation programs. The diagnostic procedures revealed bugs, faulty

algorithms, limitations, and input errors in all of the programs tested in this study. This includes the following models that ran the test cases – DEEAP, DeST, DOE-2.2, EnergyPlus, IES-VE, LCEM, and TRNSYS. Some important technology advances occurred by running the test cases:

- The QAS allowed diagnosis of errors and identification of potential areas for improvement that may have been missed in a comparative study of whole-building energy simulation programs versus each other, without a quasi-analytical solution benchmark.
- Of 26 found program errors or potential improvements, 17 were diagnosed and fixed, 2 are planned for investigation by the software authors, and 7 are unresolved at the time of this writing. Table 3-18 summarizes notable examples of error trapping—supporting details are given in Section 3.4 and in the modeler reports (see Section 3.9).
- Additionally, 17 input errors were found and corrected (indicated by italicized “*Input*” in the “Error Description” column of Table 3-18). These provided the basis for revealing a number of test specification ambiguities that were then clarified during the simulation trials.

Many of the program improvements listed in Table 3-18 were significant, with greater than 20% effect on total coil loads. For individual programs that applied the current in-depth test cases, some errors had relatively minor (< 1%) effect on cooling load. Where a program had multiple errors of smaller magnitude, such errors did not necessarily compensate each other and may have been cumulative. Therefore, correcting the minor as well as the major errors was important.

The number of input errors listed in Table 3-18 and the resulting test specification clarifications indicate the importance of iterative field trials for vetting a test suite before it is promulgated for wider use as a standard method of test.

After clarifying the test specification, correcting software errors, and making other model improvements using the diagnostic output, the mean of all simulated results of total cooling coil load for the tested programs is, on average, within 1.3% of the QAS results, with average variations among the test cases for a given program of up to 3%. See Table 3-17 (Section 3.5.1) for details of final disagreement range extremes, disaggregated by coil-load outputs and system types.

This shows how the test procedure is used to diagnose and correct or improve algorithms in complex simulation programs. Diagnosis is enhanced by the availability of the QAS results, which may be used as a reference or benchmark against which other software not involved in this project can be tested. Based on results after several simulation trial iterations (“BESTESTing”) and resulting model improvements, all of the tested programs appear reliable for modeling the tested HVAC air distribution systems under the conditions examined, although use of constant system air density by some simulation programs causes modeling disagreements at extreme-low outdoor air temperatures for the specified CV system. Without additional test cases having realistic hourly varying weather data, we do not have a means to scale, relative to typical annual simulations, the effect of disagreements identified in these in-depth diagnostic steady-state analytical verification tests. For future work we recommend developing an airside HVAC comparative test suite with annual hourly varying weather data, analogous to the HVAC BESTEST Volume 2 (Neymark and Judkoff 2004) working-fluid side test suite.

Based on this work, there are a number of recommended areas for further investigation with respect to developing additional validation test cases for airside HVAC equipment modeling. These are described in detail in Section 3.5.2.1. For the longer term, we hope to develop test cases that emphasize special modeling issues associated with more complex building types and HVAC systems (see Section 3.5.2.2).

**Table 3-18. Summary of Software Issues Found Using Airside HVAC Equipment Cases**

Model <sup>a,b</sup>	Issue Description <sup>a</sup>	% Coil Load Effect or Disagreement <sup>a,c</sup>	Outcome <sup>a,b</sup>
DEEAP	Mass flow imbalance	~20% heating: AE101, AE201	Fixed
DEEAP	<i>Input:</i> Fan heat not applied <sup>d</sup>	~ 3% heating, > % for cooling (250 W)	Fixed <sup>d</sup>
DEEAP	Cooling coil outlet W constant at 50%	20% latent cooling: AE206	Fixed
DEEAP	Enthalpy economizer control	18% sensible cooling: AE245	Fixed <sup>e</sup>
DEEAP	<i>Input:</i> supply outlet temperature not set	Up to 300% total cooling: CV system	Fixed
DEEAP	Reheat coil disaggregation	Reheat: Zone1 +100%; Zone2 -100%	Fixed
DEEAP	Cooling coil entering W missing	Results not retained	Fixed
DEEAP	Tighter iteration tolerance on W		Fixed
DEEAP	<i>Input:</i> Reference density at supply fan inlet <sup>d</sup>	Cumulative up to 3% total cooling for these five issues, after above items addressed	Fixed <sup>d</sup>
DEEAP	Moist air cp based on different functions		Fixed
DEEAP	Fan heat based on standard air properties		Fixed
DEEAP	Zone latent load from hfg rather than hg		Fixed
DeST	<i>Input:</i> Return fan position <sup>d</sup>		Negligible
DeST	Constant system air density	Up to 21% pre-heat <sup>f</sup> vs. QAS (AE301); 2%-4% sensible cooling (CV)	A.N. <sup>g</sup>
DOE-2.2 (2)	Zone latent gains calculation, FC system	21% latent (7% total) cooling: AE104	Fixed (2)
DOE-2.2	Detailed output report precision (T,W)	No effect on loads	A.N. <sup>g</sup>
DOE-2.2	<i>Input:</i> Increase number of run days	Cumulative 1% total cooling VAV (4% AE403, at lower load)	Fixed
DOE-2.2	<i>Input:</i> Fan power curve to cube of flow		Fixed
DOE-2.2	<i>Input:</i> Minimum OA setting, CV econo runs	7% tot. clg.: AE326 (0.4% AE345)	Fixed
DOE-2.2	<i>Input:</i> Minimum OA setting, SZ econo runs	0.5% tot. clg.: AE226 (0.2% AE245)	Fixed
DOE-2.2	<i>Input:</i> Return fan efficiency	To 0.9% tot. clg. SZ (0.2% CV ,VAV)	Fixed
DOE-2.2	<i>Input:</i> OWB <sup>d</sup>	Negligible	Fixed <sup>d</sup>
DOE-2.2	System air density at mixed-air location properties	Up to 28% pre-heat <sup>f</sup> vs. QAS (AE301); 3%-5% sensible cooling (CV)	A.N. <sup>g,h</sup>
EnergyPlus	Heating coil auto-size: CV, VAV systems	Results not retained	A.N. <sup>g,i</sup>
EnergyPlus (3)	Latent gains throughout program; other checks initiated by this work	No impact on test cases; affects program elsewhere	Fixed (3)
EnergyPlus	<i>Input:</i> Reheat coil maximum airflow setting	10% Zone1 reheat coil	Fixed
EnergyPlus	<i>Input:</i> Fan power curves to cubic	Cumulative (including reheat coil issue) 1% to 12% total cooling: all cooling cases.	Fixed
EnergyPlus	<i>Input:</i> Supply fan minimum flow fraction <sup>d</sup>		Fixed <sup>d</sup>
EnergyPlus	<i>Input:</i> Return fan airflow rate <sup>d</sup>		Fixed <sup>d</sup>
EnergyPlus	Constant altitude-adjusted standard air density	Up to 21% pre-heat <sup>f</sup> vs. QAS (AE301); 2%-4% sensible cooling (CV)	A.N. <sup>g</sup>
IES-VE	Zone latent load calculation	Negligible (to 0.2% latent cooling)	Fixed
IES-VE	Zone latent load hfg	Up to 6% latent load vs. QAS	A.N. <sup>g,h</sup>
IES-VE	Outdoor air at standard air properties	Up to 24% pre-heat <sup>f</sup> vs. QAS (AE301); 2%-4% sensible clg. (CV)	A.N. <sup>g,h</sup>
LCEM	One coil model failed (other models ok)	Results not reported	A.N. <sup>g</sup>
LCEM	<i>Input:</i> Supply/return fan temperature rise	1% to 4% total cooling: FC, SZ <sup>j</sup>	Fixed
LCEM	Constant system air density	Outside and supply air mass flows up to 3% (FC, SZ) <sup>j</sup>	A.N. <sup>g</sup>
TRNSYS	<i>Inputs:</i> Trial "2a" spec revisions (see Section 3.3): CV/VAV <sup>d</sup>	Up to 7% total cooling: CV, VAV	Fixed <sup>d</sup>
TRNSYS	<i>Input:</i> Constant system pressure <sup>d</sup>	Cumulative 2% to 3% latent (0.3% to 2% total) cooling	Fixed <sup>d</sup>
TRNSYS	Condensate characterization <sup>d</sup>		Fixed <sup>d</sup>
TRNSYS	Zone latent load from hfg		Fixed

<sup>a</sup> Acronyms and abbreviations used in this column are defined in Section 3.7.

<sup>b</sup> Numbers in parenthesis are number of changes associated with the listing, if > 1.

- <sup>c</sup> Specific cases relevant to the described effects are listed. For disagreements indicated as “Fixed”, listed percentage values are the effect of the change for new results versus previous results for a given model. For disagreements indicated as authors notified (“A.N.”), listed percentage values are versus the QAS. See Section 3.4 for further details.
- <sup>d</sup> Per test specification clarification.
- <sup>e</sup> Additional differential dry-bulb check removed for testing; code author recommends not to remove for typical control.
- <sup>f</sup> Accompanying reheat coil load disagreements are consistent with balancing the pre-heat coil load disagreement.
- <sup>g</sup> Authors notified.
- <sup>h</sup> Software developer indicates currently applied assumptions to be updated in future program revision.
- <sup>i</sup> Modelers developed an input work-around.
- <sup>j</sup> Did not run CV and VAV cases.

### 3.6.1.3 Advantages of BESTEST Methodology

Similar to previous test suites that applied the BESTEST method, these new cases have a variety of uses, including:

- Comparing predictions from building-energy simulation programs to the QAS and simulation results in this report, where the QAS results constitute a reliable set of theoretical results given the underlying physical assumptions in the case definitions, and the simulation results represent a reasonable range of disagreement for the current state-of-the-art in simulation modeling
- Comparing several building-energy simulation programs to determine the degree of disagreement among them
- Diagnosing the algorithmic sources of prediction differences
- Checking a program against a previous version of itself after the internal code has been modified, to ensure that only the intended changes actually resulted
- Checking a program against itself after a single algorithmic change to understand the sensitivity between algorithms.

An advantage of the BESTEST methodology is that a program is examined over a broad range of parametric interactions based on a variety of output types, minimizing the possibility of concealing problems by compensating errors. Performance of the tests resulted in quality improvements, or identification of potential improvements, to all but one of the building energy simulation models used in the field trials. Some of the bugs that were found may well have been present for many years. The fact that they have just now been uncovered shows the power of BESTEST and suggests the importance of continuing to develop formalized validation and diagnostic methods. Only after coding bugs and input ambiguities have been eliminated can the assumptions and approximations in the algorithms be evaluated.

Checking a building energy simulation program for the first time with the airside HVAC equipment model test cases requires a few days for an experienced user, not including improvements to the software, if necessary. Subsequent program checks are faster because input files may be reused. Because the simulation programs have taken many years to produce, the new BESTEST cases provide a cost-effective way of testing them. As we continue to develop new test cases, we will adhere to the principle of parsimony so the entire suite of BESTEST cases may be implemented by users with a reasonable time commitment. Several software developers have automated the tests, further reducing the time to rerun them as an internal quality control check after code modifications.



## 3.6.2 Recommendations

### 3.6.2.1 Adaptation of Test Procedures for ANSI/ASHRAE Standard 140

The work presented in this report and the work that has preceded it (discussed below) are important for two reasons:

- The methods have been successful at correcting software errors in advanced building-energy simulation programs throughout the world.
- The methods are finding their way into industry by being adopted as the theoretical basis for formalized standard methods of test and software certification procedures; in this sense, the work may be thought of as pre-normative research.

The overall validation methodology (Judkoff 1988; Judkoff and Neymark 2006, 2009; Judkoff et al. 2008) is included in the *ASHRAE Handbook of Fundamentals* (ASHRAE 2013) and informative Annex B23 of Standard 140 (ANSI/ASHRAE 2014). The following test suites, all but one of which were developed in conjunction with the International Energy Agency (IEA), have been code-language adapted and formally approved as a standard method of test, ANSI/ASHRAE Standard 140-2014, *Standard Method of Test for the Evaluation of Building Energy Analysis Computer Programs* (ANSI/ASHRAE 2014):

- Class I tests (detailed diagnostic tests for simulation software capable of hourly or subhourly simulation time steps)
  - IEA BESTEST, building thermal fabric comparative tests (Judkoff and Neymark 1995a)
  - IEA BESTEST, in-depth diagnostic ground-coupled slab-on-grade heat transfer analytical verification tests (Neymark and Judkoff, et al. 2008a)
  - HVAC BESTEST Volume 1, unitary cooling equipment analytical verification tests (Neymark and Judkoff 2002)
  - HVAC BESTEST Volume 2, unitary cooling equipment comparative tests (Neymark and Judkoff 2004)
  - HVAC BESTEST fuel-fired furnace analytical verification and comparative tests (Purdy and Beausoleil-Morrison 2003)
- Class II tests (for all types of building load calculation methods regardless of time-step granularity)
  - HERS BESTEST, building thermal fabric comparative tests (Judkoff and Neymark 1995b).

Within the BESTEST/Standard 140 structure, there is room to add new test cases. BESTEST is better developed in areas related to energy flows and energy storage in the architectural fabric of the building. BESTEST work related to mechanical equipment is still in its early phases in that there are many kinds and configurations of mechanical systems to test.

The new airside HVAC equipment test cases described in this report are planned for inclusion in Standard 140. Additional test cases either currently being adapted for inclusion in Standard 140 or planned for future inclusion are listed in ASHRAE Standard 140, Informative Annex B23 (ANSI/ASHRAE 2014).

### 3.6.2.2 Closing Remarks

ASHRAE Standard 140 (ANSI/ASHRAE 2014) and the BESTEST reports that comprise the test suites contained therein are being referenced and used worldwide by a growing number of code promulgation authorities. ASHRAE Standard 90.1 (ANSI/ASHRAE/IESNA 2013), which is ASHRAE's consensus

energy code for commercial buildings and for non-low-rise residential buildings, requires that software used for demonstrating performance compliance with Standard 90.1 be tested using the Class I test procedures of ASHRAE Standard 140-2011 (ANSI/ASHRAE 2011). Software used for calculating energy savings for purposes of the energy-efficient commercial building tax deductions in the United States (IRS 2008) must be tested with Standard 140-2007 (ANSI/ASHRAE 2007). As part of their building energy performance assessments under the European Community's *Energy Performance Directive* (European Union 2002), several countries are using software tools that have been checked with BESTEST. Further details of international use of BESTEST, along with growing evidence that the BESTEST procedures are becoming part of software developers' normal in-house quality control efforts, are included elsewhere (Judkoff and Neymark 2006, 2013; Neymark and Judkoff et al. 2008a, Section 2.6.2.2).

Computer scientists universally accept the merits of software testing. A well-known rule of thumb is that in a typical programming project more than 50% of the total cost is expended in testing the program or system being developed (Myers 2004). Of this, about 20% of development time goes toward system testing (McConnell 2004). Because new energy-related technologies are continually being introduced into the buildings market, there will always be a need for further development of simulation models, combined with a substantial program of testing and validation. Such an effort should contain all the elements of an overall validation methodology (ANSI/ASHRAE 2014, Informative Annex B23; ASHRAE 2013, Ch. 19; Judkoff and Neymark 2006), including:

- Analytical verification
- Comparative testing and diagnostics
- Empirical validation.

Future work should therefore:

- Continue to produce analytical verification tests where possible
- Continue to develop a set of diagnostic comparative tests that emphasize the modeling issues important in large commercial buildings, including more tests for heating, ventilating, and air-conditioning systems, and other mechanical equipment, including on-site power generation equipment
- Develop a sequentially ordered series of high-quality diagnostic data sets for empirical validation.

The work described here represents continuing progress in the effort to develop carefully validated building energy simulation tools. Continued development and validation of whole-building energy simulation programs are two of the most important activities meriting the support of national energy research programs and are a beneficial area for international collaboration. The U.S. Department of Energy should consider future International Energy Agency collaborations for this essential research area.

Finally, the authors wish to acknowledge that the expertise available through ASHRAE SSPC 140 and other international participants, and their dedication to the simulation trials, were essential to the success of this project. Over the four-year field-trial effort, there were several revisions to the test specifications and subsequent re-executions of the computer simulations. This iterative process led to the refining of the new BESTEST cases, and the results of the tests led to improving and debugging of the simulation models. This process underscores the importance of the software development industry participation in this project via the ASHRAE Standard 140 project committee (SSPC 140), and via our previous collaborations with International Energy Agency software testing and validation projects. Extensive field trials, and resulting enhancements to the tests were much more comprehensive and cost effective with the participation of the national and international software-development industry experts. Their persistence throughout the project demonstrates their interest in, and the importance of, this work.

### 3.7 Abbreviations and Acronyms for Part III

These abbreviations and acronyms are used in Sections 3.1 through 3.6.

ANSI	American National Standards Institute
Apr	April
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
Aug	August
BESTEST	Building Energy Simulation Test and Diagnostic Method
BF	bypass factor
clg.	cooling
cp	specific heat
CV	constant volume system, see Part I, Section 1.5.5.3
Dec	December
DEEAP	simulation program, see Part IV, Table 4-1
DeST	simulation program, see Part IV, Table 4-1
DOE-2.2	simulation program, see Part IV, Table 4-1
econo.	economizer
EnergyPlus	simulation program, see Part IV, Table 4-1
FC	fan coil system, see Part I, Section 1.5.5.1
Feb	February
Hf	enthalpy of liquid
hfg	heat of vaporization
hg	enthalpy of vapor
HVAC	heating, ventilating, and air conditioning
IEA	International Energy Agency
IES-VE	simulation program, see Part IV, Table 4-1
Jan	January
Jun	June
LCEM	simulation program, see Part IV, Table 4-1
Mar	March
N/A	not applicable
Nov	November
NREL	National Renewable Energy Laboratory
OA	outdoor air
ODB	outdoor dry-bulb temperature
ODP	outdoor dew-point temperature
OWB	outdoor wet-bulb temperature
QAS	quasi-analytical solution – NREL’s final merged version of the two independent spreadsheets of the original RP 865 report
RH	relative humidity
RP 865	Yuill and Haberl (2002)
SSPC 140	ASHRAE Standing Standard Project Committee 140, responsible for ANSI/ASHRAE Standard 140 <i>Standard Method of Test for the Evaluation of Building Energy Analysis Computer Programs</i>
SZ	single zone system, see Part I, Section 1.5.5.2
T	temperature
tot.	total
TRNSYS	simulation program, see Part IV, Table 4-1
VAV	variable air volume system, see Part I, Section 1.5.5.4

VECM            verified external calculation method – NREL’s preliminary merged version of the two independent spreadsheets of the original RP 865 report; also see QAS  
W                humidity ratio

### 3.8 References for Part III

These references are cited in Sections 3.1 through 3.6.

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### 3.9 Appendix III: Simulation Modeler Reports

Here we present reports written by the modelers for each simulation program. The modelers were asked to document:

- Modeling assumptions (required inputs not explicitly described in the test specification)
- Modeling options (alternative modeling techniques)
- Difficulties experienced in developing input files for the test cases with their programs
- Bugs, faulty algorithms, documentation problems, and input errors uncovered during the field trials
- Source code or input modifications made because of the diagnostic results
- Comments on agreement or disagreement of results compared to the quasi-analytical solution and other simulation results
- Any odd results obtained with their programs
- Sensitivity studies conducted to further understand the sources of differences between their programs and the others
- Conclusions and recommendations about their simulation models, the test specification, or both.

Modelers also filled out a pro-forma description that defines allowable ranges of inputs for geometry and material properties related to slab-on-grade construction. The pro-forma description is provided at the end of each individual modeler report. The modeler reports appear as submitted, with minimal reformatting and editing.

Some figures in the modeler reports use case labels from the early simulation trials. The simulation-trial test cases used a prefix of “AET”, and the FC and SZ test cases had a suffix of “a” to identify the zone. For the final version of the test specification, all of the test cases in Part I have a prefix of “AE”, and the “a” suffix is dropped for the FC and SZ test cases.

## **Appendix III-A**

### **Modeler Report for HVAC Air Distribution System Tests AAON Detailed Energy and Economics Analysis Program (DEEAP), Version 1.1.2**

by

**John Gall  
AAON, Inc.  
United States**

**March 2015**

#### **1. Introduction**

The AAON Detailed Energy and Economics Analysis Program (DEEAP) is in version 1.1.2. The software is composed of three main modules. The first module is EnergyPlus, which is used to compute the loads for the analysis. At the time of final results development EnergyPlus was in Version 7.0.0.036 [1]. EnergyPlus is developed in the United States of America (USA) and distributed by the United States Department of Energy. The second module of the program, used to compute the system response to the loads at steady state is derived from the Thermal Storage Analysis program originally produced by Engineering Applications Specialists Inc., in the USA [2], but has been modified from a bin method to accept hour by hour input by AAON Inc., USA. The third module is a user interface which has been wholly created by AAON Inc., USA.

The program modeling of mechanical air distribution systems is derived from the methods utilized in the BLAST program originally developed in the USA, but no longer maintained.

The loads program EnergyPlus can utilize a subhourly timestep; for precision often a 15 minute timestep is used, but reporting frequency is hourly, and the systems solver program accepts hourly loads information.

#### **2. Modeling Assumptions**

The loads program, EnergyPlus, required some additional sensible loads inputs in order to produce the proper load profile in the zone. I was unaware of an input in the EnergyPlus loads section that allowed for an infrared emittance value. Apart from this, generic place hold inputs for required items such as unloading curves for dx unit models, unit capacity, fan unloading curves, generic utility rates were input.

Additional information is included in the pro-forma questionnaire responses; see Appendix A of this supplementary modeler report.

#### **3. Modeling Difficulties**

As mentioned above, some additional loads information was required to achieve the proper load profile in the space, and infrared emittance was not directly input.

In some instances, inputs were not easily taken by the interface, and had to be directly entered as text into the input stream.

The EnergyPlus program used for the loads module utilizes SI units, while the Thermal Storage Analysis program uses English units. Users typically use English units. Inputs to the interface in English units must



be converted to SI, back to English, and back to SI for the output, so some small artifacts of the conversion and rounding process could be carried forward as error.

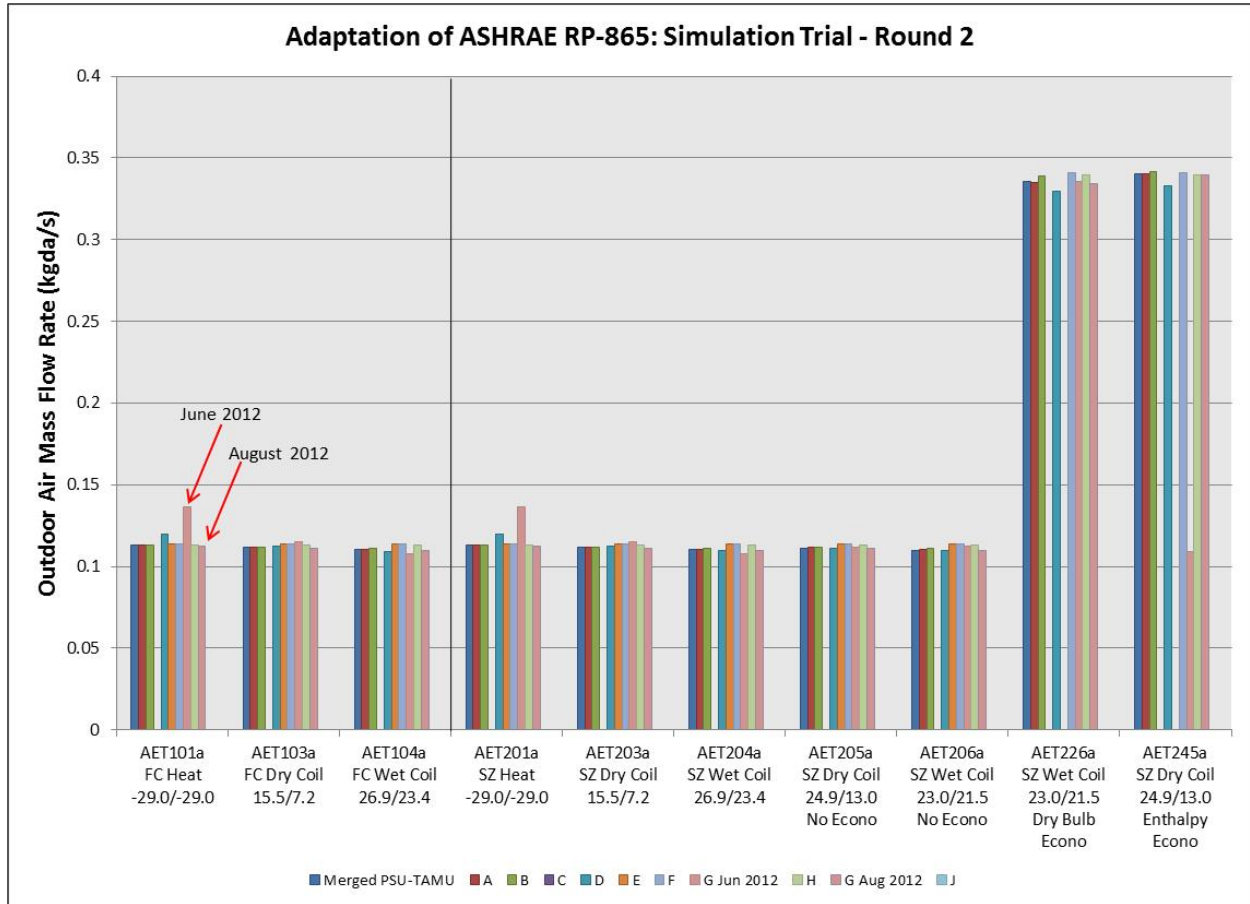
#### **4a. Software Errors Discovered, Program Improvements, Input Corrections, and/or Other Comparisons Between Different Versions of the Same Software**

In the latest revision, there were a few things that were adjusted:

1. Cooling coil entering air humidity ratio was not being fed forward from the mixed air or heating coil leaving air condition, and was modified to reflect proper airflow. For certain cases where the cooling coil was upstream from the heating coil (terminal reheat) an incorrect humidity ratio was propagated to the cooling coil, but in cases where the heating coil was upstream from the cooling coil (single zone and fan coil), the humidity ratio was likely correct. (Fixed program error) Only preliminary results were attempted without making these changes, and were not retained.
2. One order of magnitude tightening of the iteration tolerance that is based upon humidity ratio. (Program improvement)
3. The previous test spec instructions were ambiguous about the definition of the supply air. After spec clarification, adjustments were made to the density calculation to reflect the air entering the supply fan as the reference density. (Program improvement and spec improvement)
4. Calculation of moist air specific heat was adjusted. Two different functions were being used in different places, and it was standardized. (Program improvement)
5. Fan heat calculations were adjusted from the earlier computation that was based upon standard air. Now fan heats are being calculated based upon the mass flow rate through the fan. (Program improvement)
6. The biggest factor in changes to the results is the correction to the moisture load in the space. It was being erroneously computed by hfg (latent heat of vaporization) rather than hg (enthalpy of water vapor at zone air temperature). Also see discussion with Issue 4 below. (Fixed program error)

Upon review of the Adaptation of Air Handling System Test Cases from 865 TRP to Standard 140 Simulation Trials - Round II report from San Antonio June 25, 2012 [3], the result differences in the AAON Energy and Economics Analysis Program Version 1.1 (indicated as Program “G” in the following plots from the early simulation trials) and the Merged PSU-TAMU solution (defined as the quasi-analytical solution [QAS] in the final report) are discussed and sources of those differences are explained. Methods for addressing the issues are also presented. The simulation trials were re-run, and the issues below are accounted for in the methods described and the causes for discrepancy are minimized, as shown in the results labeled “G Aug2012”.

**Issue 1: In AET101a (a) the outdoor air mass flow rate is greater than predicted by the merged solution, and (b) there is a mass imbalance in the system; see Figure 3A-1, June 2012 result for the initial prediction and the August 2012 result for the corrected result. (Program improvement)**

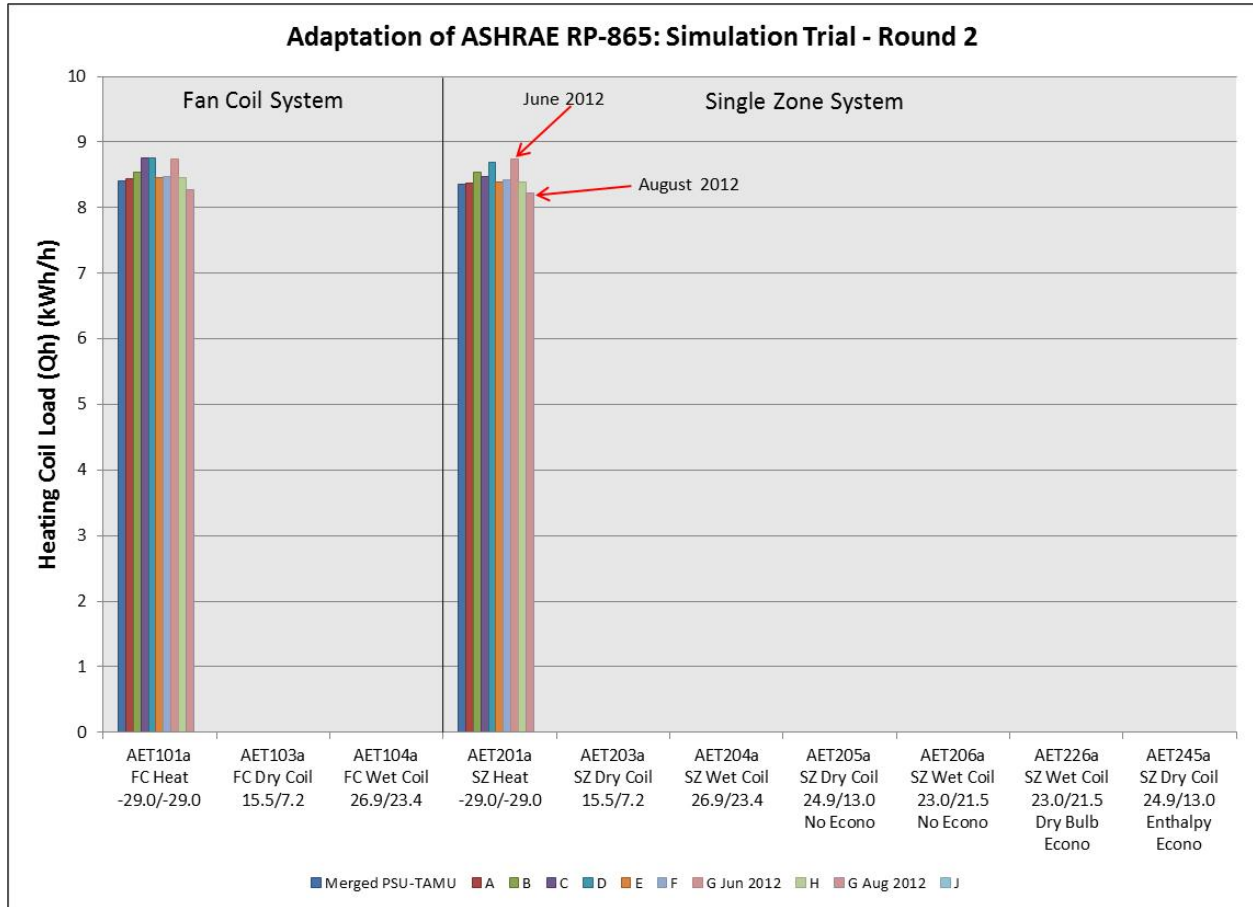


**Figure 3A-1. FC and SZ Outside Air Mass Flow Rate, Simulation Trial Round 2, Jun-Aug 2012**

Response to Issue 1: Both a and b follow directly from the same root cause. The system analysis portion of the program utilizes an SCFM (standard CFM) method of calculation, and carries volumetric flow rate through the various components as the airflow variable. The density calculations were made post-simulation and were based upon the temperature and humidity ratio at each node, and mass flow rate was similarly computed post-simulation without proper consideration for actual CFM.

Steps taken to address issue 1: The algorithm that computes the Fan Coil type single zone draw through system configurations is provisionally adjusted to utilize a mass flow rate variable rather than a volumetric flow rate variable, which directly outputs the mass flow rate without any post-simulation calculations other than conversion to SI (as the program remains in English units).

**Issue 2: In AET101a and AET201a, the program predicts a greater heating load than is predicted by the merged solution; see Figure 3A-2, June 2012 result for the initial prediction and the August 2012 result for the corrected result. (Input error)**

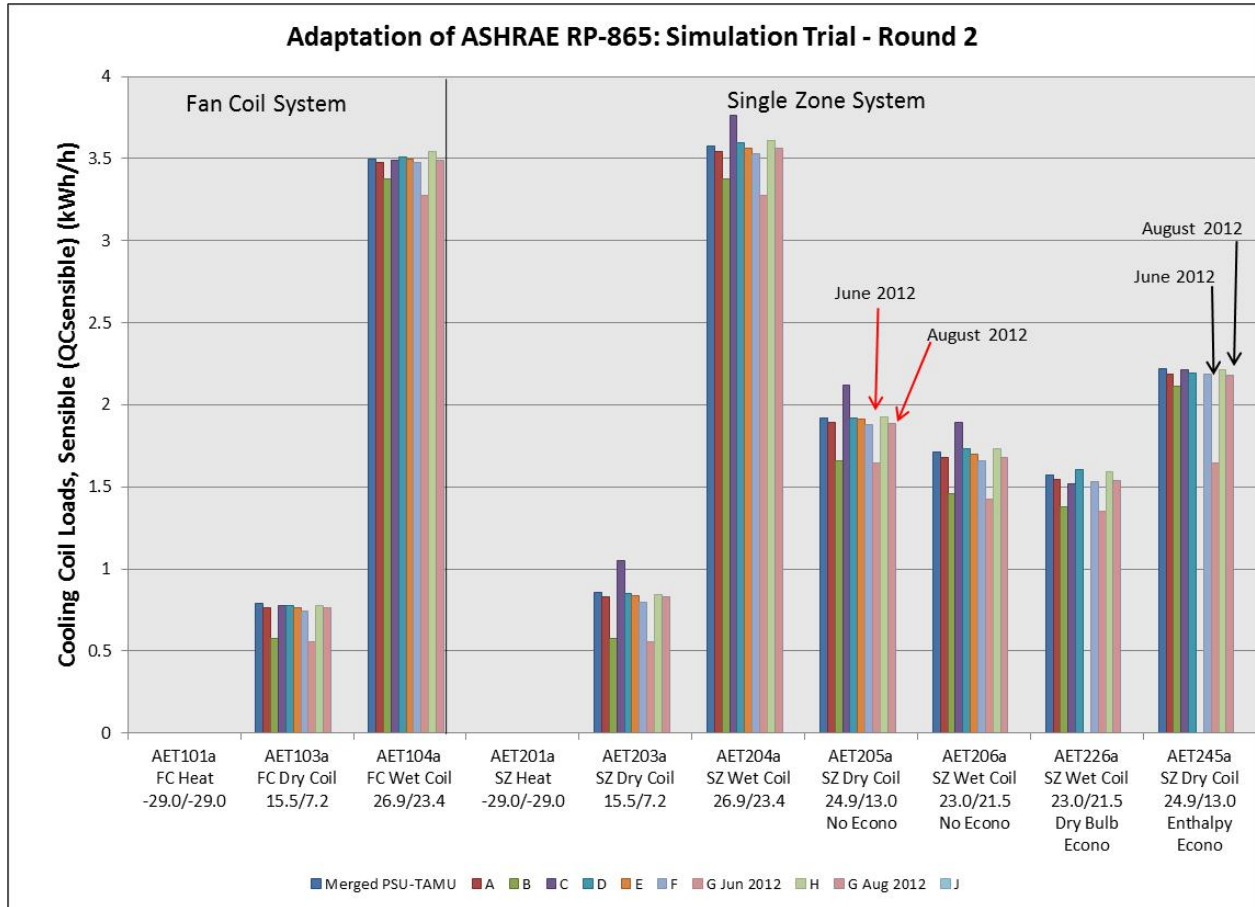


**Figure 3A-2. FC and SZ Heating Coil Load, Simulation Trial Round 2, Jun-Aug 2012**

Response to Issue 2: Part of the cause of greater heating load is attributed to the absence of fan heat. An oversight from a misunderstanding of the instructions led to the fan heat being eliminated from the simulation, and an increase in system heating coil energy as compensation. The high outdoor air mass flow rate (see Figure 3A-1) would also have an impact here. *[Editor's note: In the QAS the presence of supply fan heat reduces heating coil load by 2.4% (matching approximate fan heat manual calculation of  $0.33 \text{ kg/s} * 1006 \text{ J/(kgK)} * 0.6^\circ\text{C} = 200 \text{ W}$ ). Presence of return fan heat in the QAS SZ model reduces system heating coil load by another 0.7% (roughly 60 W).]*

Steps taken to address Issue 2: Fan heat was restored to the simulation for all cases.

**Issue 3: In all FC and SZ system test cases with cooling, the program predicted sensible cooling energy lower than that predicted by the merged PSU-TAMU solution; see Figure 3A-3, June 2012 result for the initial prediction and the August 2012 result for the corrected result. (Input error)**

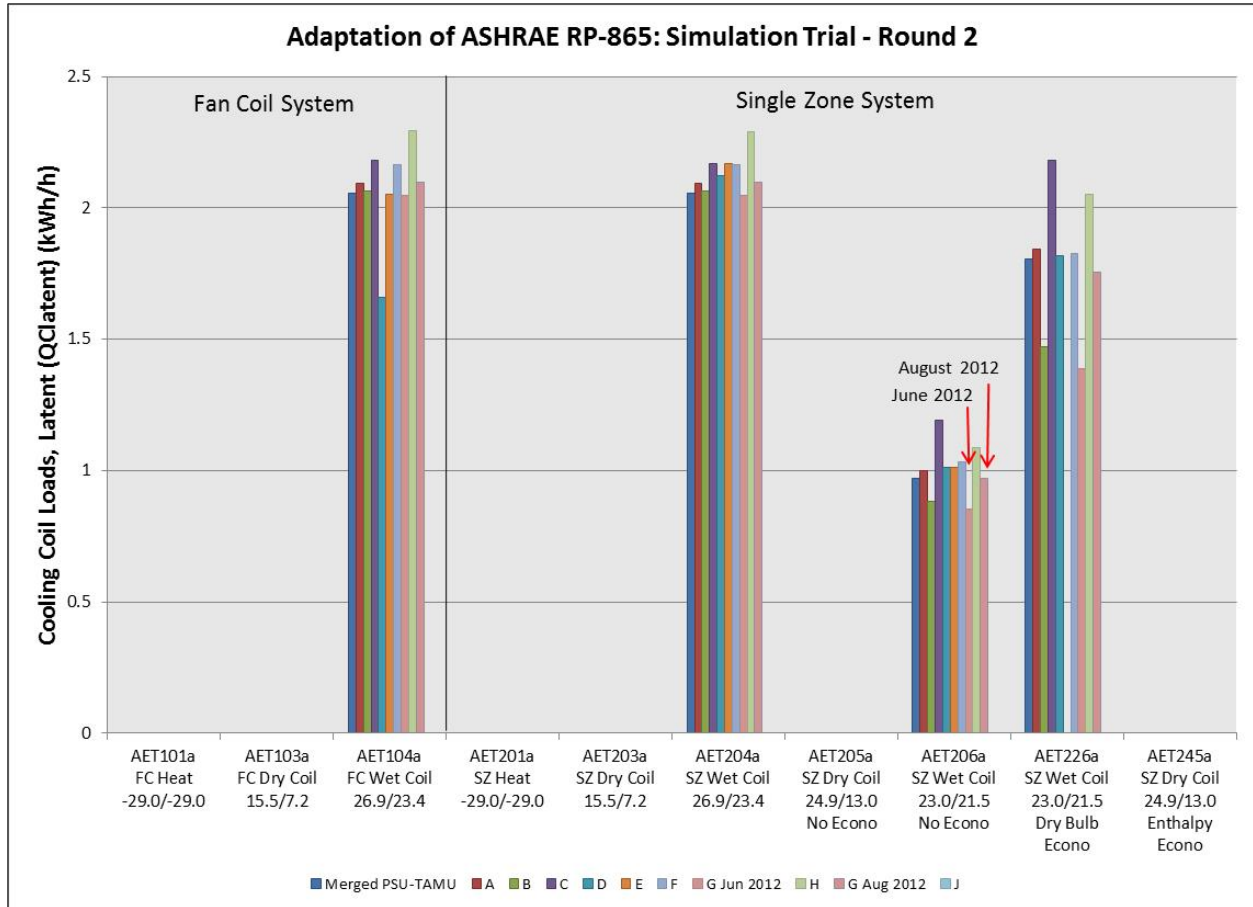


**Figure 3A-3. FC and SZ Sensible Cooling Load, Simulation Trial Round 2, Jun-Aug 2012**

Response to Issue 3: The primary reason for this reduction in cooling effort seems to be the fan heat being neglected from the simulation.

Steps taken to address Issue 3: As in issue 2, fan heat is restored to the simulation.

**Issue 4: Latent load for trials AET206a and AET226a is lower than merged solution prediction; see Figure 3A-4 red arrows, June 2012 result for the initial prediction and the August 2012 result for the corrected result. (Fixed program error)**



**Figure 3A-4. FC and SZ Latent Cooling Load, Simulation Trial Round 2, Jun-Aug 2012**

Response to Issue 4: This is likely a result of the humidity ratio calculations in the algorithm. A simplification in the program's calculation set the space relative humidity at 50% for comfort. A 100% cooling coil leaving air condition was also applied to the coil per the instructions. It is likely that a humidity ratio imbalance occurred in the simulation.

Steps taken to address Issue 4: The 50% humidity ratio in the space simplification is provisionally removed, and the humidity ratio is computed at each node in the system by an iterative technique.

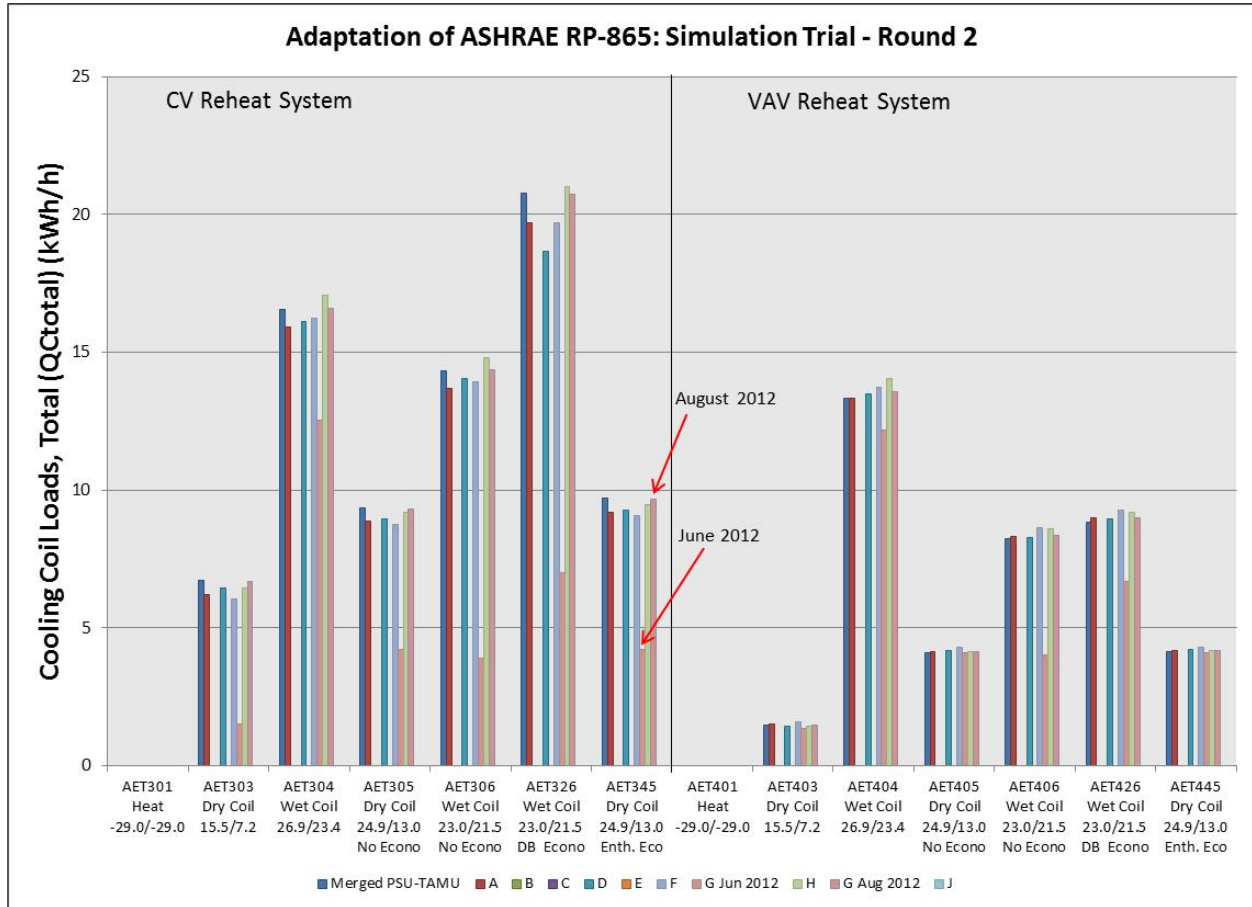
The first attempt at incorporating room latent load properly was done erroneously, utilizing the latent heat of vaporization instead of the latent heat of water vapor at the room condition. This has been corrected, and the moisture values align more closely with the merged solution prediction.

**Issue 5: Sensible Load for AET245a is lower than merged solution prediction; see Figure 3A-3, black arrows, June 2012 result for the initial prediction and the August 2012 result for the corrected result. (Program modification for testing purposes)**

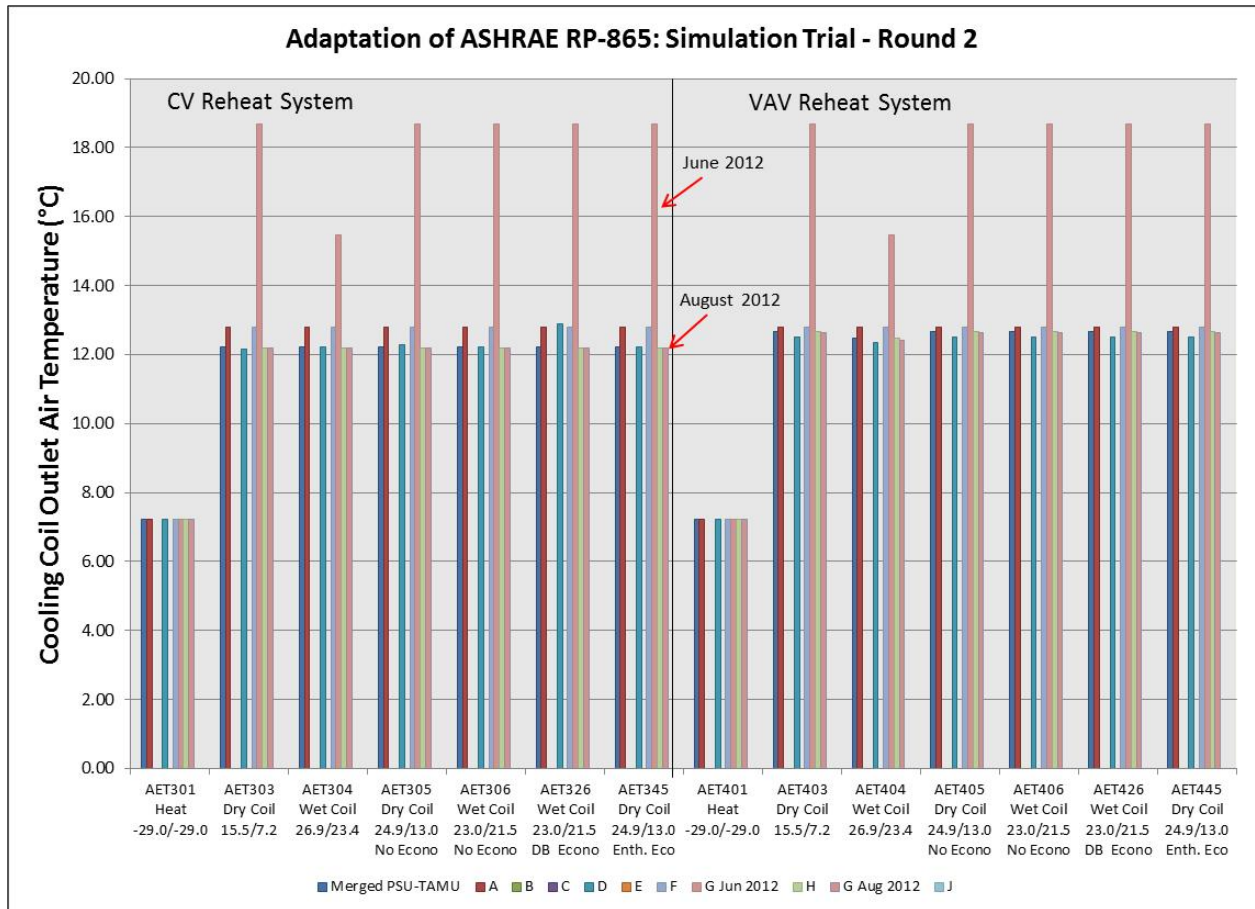
Response to Issue 5: In addition to the fan heat not being added to the case, the enthalpy economizer is not operating in this case. If the enthalpy economizer passes the differential enthalpy criteria for operation, it must then pass a secondary, differential dry bulb test before it operates in an economy mode. In this case, the enthalpy criteria was satisfied, but the dry bulb test was not satisfied, and the system operated based upon its outdoor air schedule rather than in an economizing mode.

Steps taken to address Issue 5: The algorithm was provisionally adjusted to allow the second test criteria to be relaxed for this testing, but it is unlikely that this second criteria for operation will be removed for typical simulation.

**Issue 6: Cooling Coil loads lower than merged solution for all AET300-series cases and AET400-series wet-coil cases (see Figure 3A-5 June 2012 result for the initial prediction and the August 2012 result for the corrected result). Cooling coil leaving air temperature higher than merged solution for AET300- and AET400-series cooling cases; see Figure 3A-6 June 2012 result for the initial prediction and the August 2012 result for the corrected result. (Input error)**



**Figure 3A-5. CV and VAV Total Cooling Load, Simulation Trial Round 2, Jun-Aug 2012**

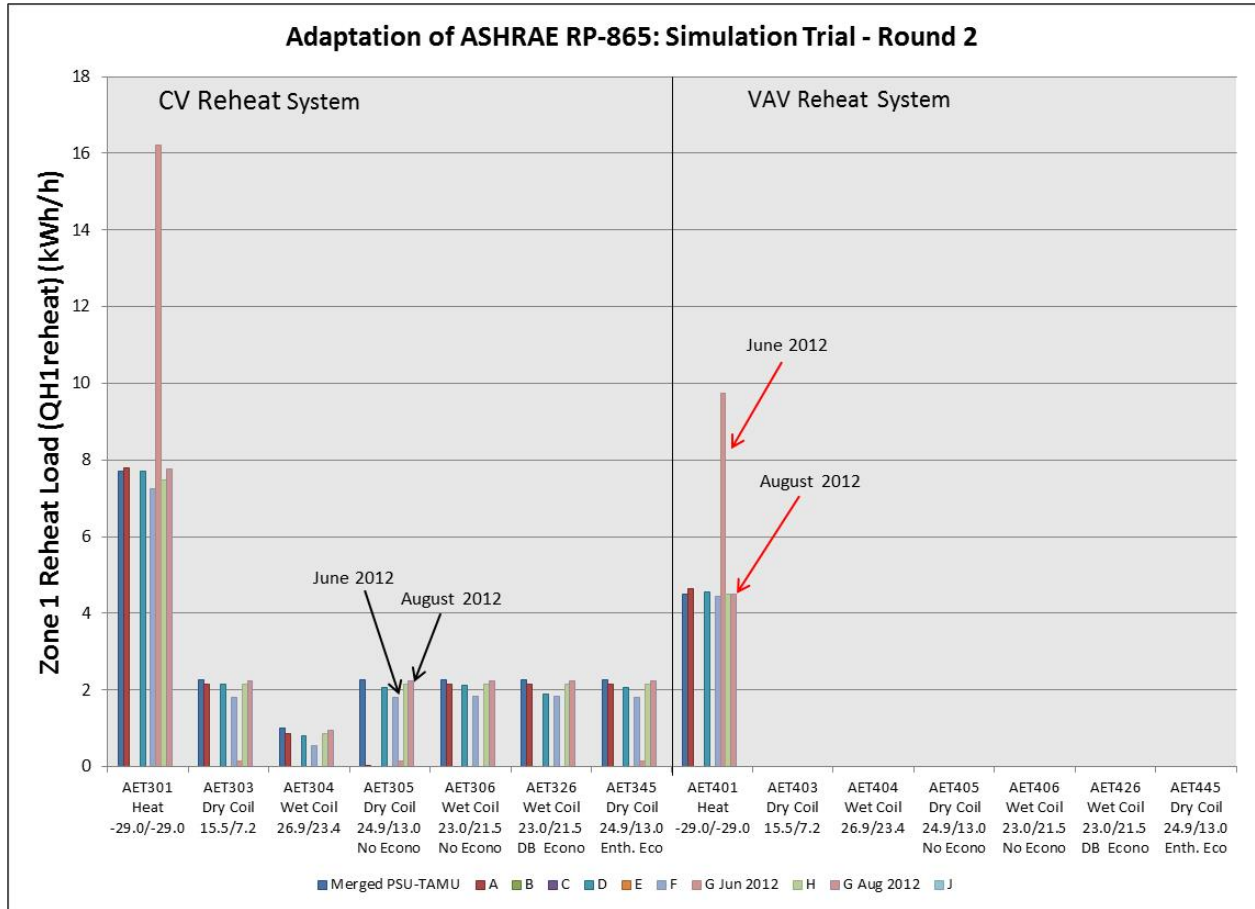


**Figure 3A-6. CV and VAV Cooling Coil Outlet Air Temperature, Simulation Trial Round 2, Jun-Aug 2012**

Response to Issue 6: This issue was caused by oversight. The required system supply dry bulb temperature in the instructions was overlooked, and the leaving air dry bulb was not properly set. This is also true for all cases AET400.

Steps taken to address Issue 6: The proper control was set for the leaving air dry bulb condition. Additionally, the adjustments in fan heat, mass flow rate, humidity ratio, and relaxed dry bulb test for the economizer previously discussed are applied to cases AET300 and AET400.

**Issue 7: Heating coil load for Zone 1 Reheat in Cases AET301 and AET401 are too high; see Figure 3A-7 red arrows, June 2012 result for the initial prediction and the August 2012 result for the corrected result. (Program improvement)**



**Figure 3A-7. CV and VAV Zone 1 Reheat Load, Simulation Trial Round 2, Jun-Aug 2012**

Response to Issue 7: The program computes the total heat required as one variable only, and the reheat required for both Zone 1 and Zone 2. Reheat was reported as reheat for Zone 1 only.

Steps taken to address Issue 7: Reheat coil loads were separated into the separate air streams being simulated for the two zones. The result can now be reported for each stream independent of the other.

**Issue 8: Reheat loads for Cases AET303 - AET345 are lower than merged solution prediction; see Figure 3A-7 black arrows, June 2012 result for the initial prediction and the August 2012 result for the corrected result. (Input error)**

Response to Issue 8: This is caused principally by the oversight on the leaving air condition.

Steps taken to address Issue 8: Same as Issue 6.

**4b. Input / Modeling Changes and Issues Reported in S140outNotes.txt**

See Appendix B of this supplementary modeler report.



## 5. Results

Enthalpy results not included because the program runs in English units. I can provide the enthalpy in English units, but I didn't convert it to SI. I can provide the English units results if desired, as I fill out the results tables (Sec5-5out.xls) with English units and then use conversion factors to convert the entire table. Also see Section E of Appendix B of this modeler report.

### ***Remaining Minor Disagreements with Quasi-Analytical Solution – Fan Heat***

Following is a summary of remaining minor disagreements identified by the NREL team test spec authors for Simulation Trial Round 3 (8/15/13) [4], with logical reasons for differences described by AAON (8/28/13) [5].

Three minor differences, all regarding fan heat, were identified.

#### ***SZ and CV System Return Fan Heat Gain***

Relative to the merged solution, the return fan heat gain, shown in Figures 3A-8 and 3A-9, is high in Case AE201 and low in all other SZ and CV cases. The SZ and CV return fan heat differs from the QAS because the SZ and CV return fans are modeled as constant volume fans whereas the QAS and specification vary the SZ and CV return fan volumetric flow as required to return the supply air mass flow minus zone exhaust air mass flow. These differences appear large in relative terms but are only a 0.03°C to 0.06°C difference in return air temperature rise. Given the small impact on coil loads this seems like a reasonable modeling simplification.

*[Editor's note: Both models approximate, but do not exactly replicate, real constant-volume system behavior. In a real system for given fan speeds the associated volumetric capacities, pressure changes, efficiencies, and therefore mass flows of all system fans would balance according to the individual fan curves.]*

#### ***VAV Supply and Return Fan Heat Gain***

The supply fan heat gain is high for the VAV test cases. The percentage differences appear large but the actual differences in supply air temperature rise are only 0.005°C to 0.03°C as shown in Figure 3A-10. The absolute differences seem reasonable. The issue stands out only because Program G supply fan heat gain is in close agreement for the other system types.

The return fan heat gain is high in all VAV cases (see Figure 3A-9). This ~36% difference is only a 0.002°C temperature difference except in case AE403 where the relative deviation is 600% and the return fan temperature rise difference is 0.03°C.

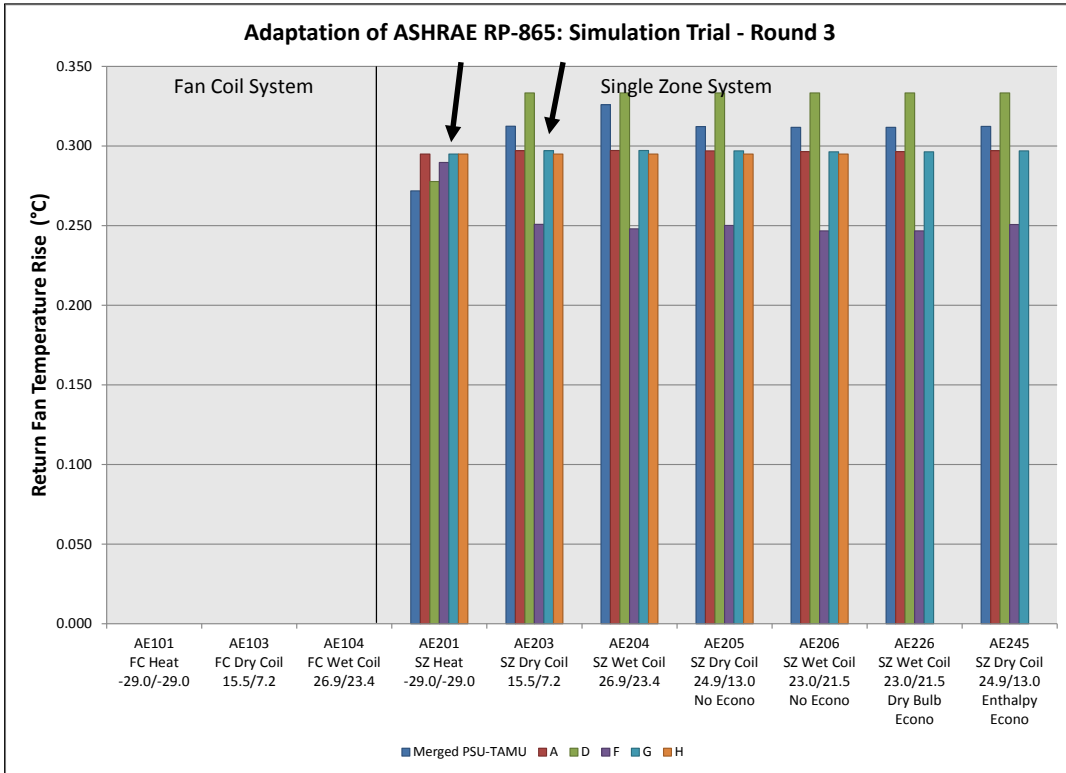


Figure 3A-8. SZ Return Fan Air Temperature Rise

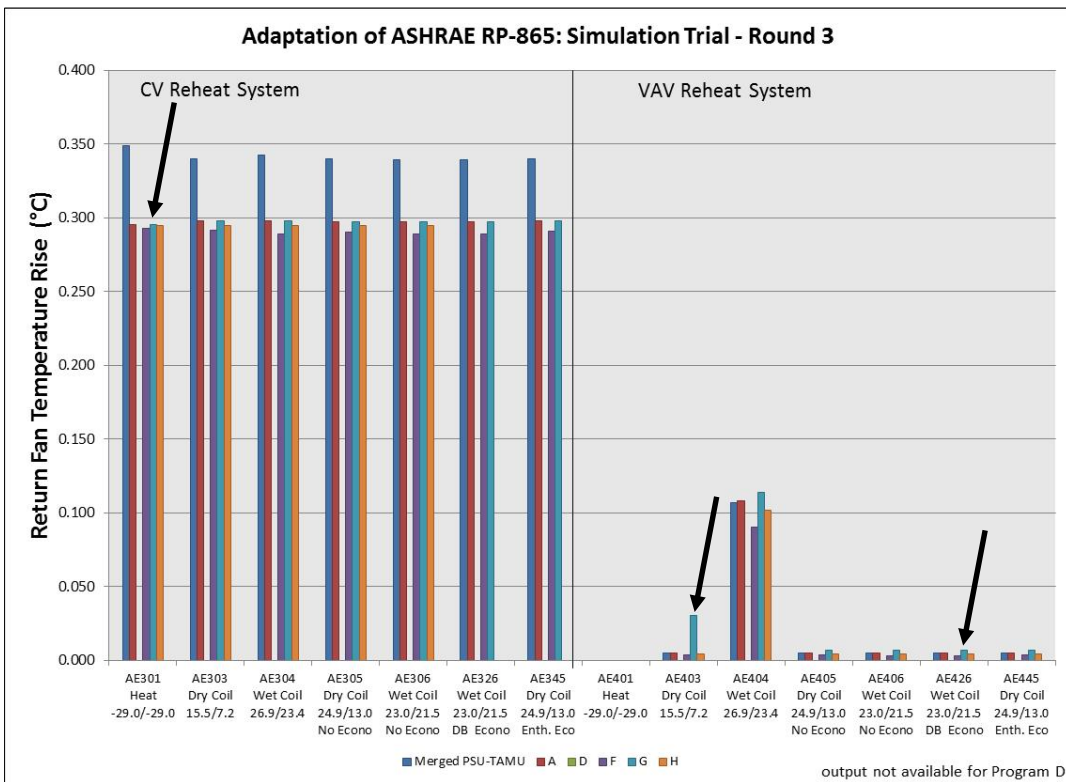
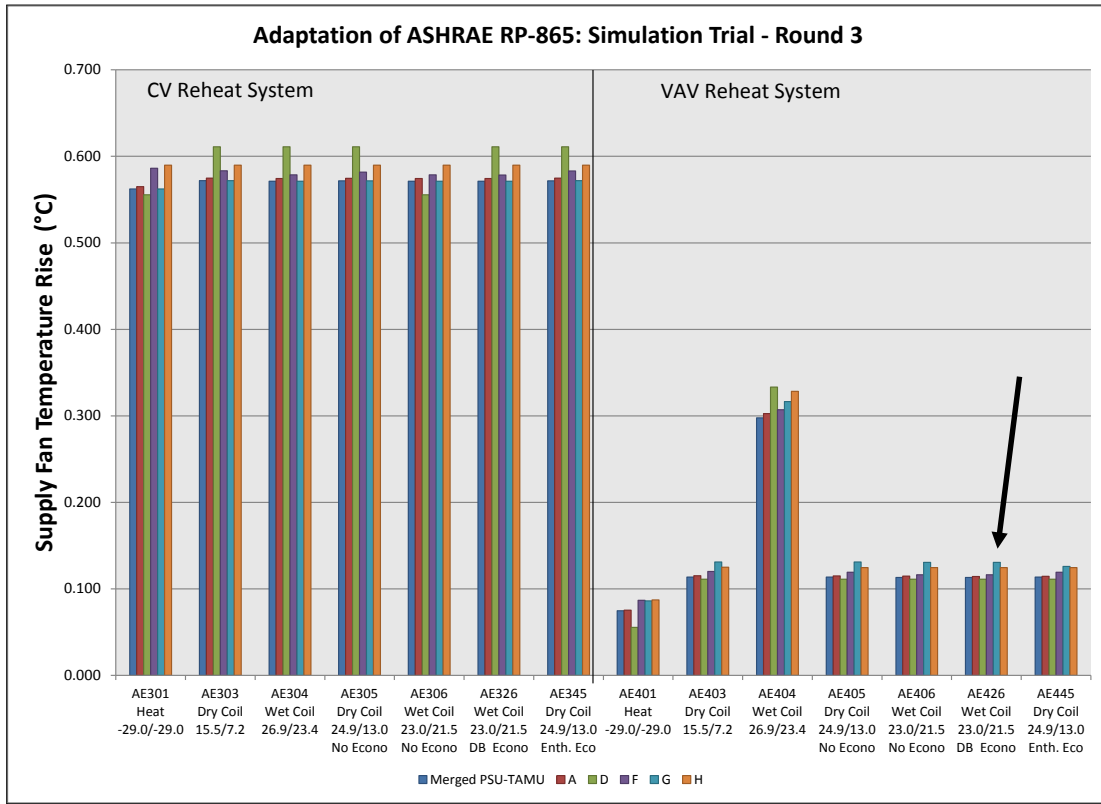


Figure 3A-9. CV/VAV Return Fan Air Temperature Rise

[Editor's Note: Prog A typo for AE403 corrected.]



**Figure 3A-10. CV/VAV Supply Fan Air Temperature Rise**

The reason the VAV supply and return fan heat is different from the quasi-analytical solution (QAS) is that the fan power required at the given condition is determined differently. The QAS uses a squared fan law to determine static pressure rise at the given condition, and fan power follows from that. DEEAP's solution [6]) uses a fan unloading characteristic function associated with a particular unloading methodology (VFD, etc.) to determine fan power as a function of load ratio and minimum static pressure.

It follows that the fan temperature rise will always be different between the two except in the case where the unloading characteristic exactly matches the power given by squared fan law.

Further detail is as follows.

In the QAS, supply fan heat calculation methodology, the supply fan pressure at the 'actual' condition is determined according to the relationship (see Part II, Equation 4):

$$ASFPR = (TSMF \times ACCSV / NSCFM)^2 \times SFPR, \text{ where}$$

ASFPR = the actual supply fan pressure rise at the test conditions (in. wg), assuming fan pressure rises proportionally to the square of the volumetric airflow rate. This is an intermediate value from within a longer spreadsheet equation; this variable name is not used in the QAS models.

TSMF = supply air mass flow ( $lb_{m,da}/min$ ), as defined in Equation 1.

- ACCSV = specific volume of the air entering the supply fan calculated at the supply fan entering air temperature and humidity ratio ( $\text{ft}^3/\text{lb}_{\text{m,da}}$ ), see Part II, Section 2.2.1.27.2 for details.
- NSCFM = design system supply airflow rate per the test specification (cfm); FC and SZ models use a variable named ROS.
- SFPR = design supply fan pressure rise = 2 (in. wg).

From this point the fan heat associated with the pressure rise across the fan is computed.

The methodology applied in DEEAP is similar, yet has a distinct and significant difference. The DEEAP methodology relies upon computation of required fan power to achieve the required pressure.

We can show that this methodology is nearly identical for Case AE403:

1. Compute the pressure with  $5.20233 \text{ (lb/ft}^2\text{)}/(\text{in. wg})$
2. Fan power = pressure \* mdot \* specific volume of air at cooling coil outlet / fan efficiency, where “mdot” is mass flow rate at the cooling coil outlet

$$\begin{aligned} \text{In Case AE403, pressure} &= 0.39777 \text{ inches wg} * 5.20233 \text{ (lb/ft}^2\text{)}/(\text{in. wg}) = 2.0693308 \text{ lb/ft}^2 \\ \text{Fan power} &= (2.0693308 \text{ lb/ft}^2 * 44.46 \text{ lb/min} * 13.04 \text{ ft}^3/\text{lb}) / 0.7 = 1713.874 \text{ ft-lb/min} \end{aligned}$$

$$\text{Convert to hp with } 33000 \text{ ft-lb/min} = 1 \text{ hp, then fan power} = 0.0519356 \text{ hp}$$

3. Convert from hp to Btu/hr,  $1 \text{ hp} = 2544.43357764 \text{ Btu/hr}$   
fan power = 132.14668 Btu/hr

4. Compute  $\Delta T = Q/(\text{mdot} * c_p)$   
 $\Delta T = 132.1466845 \text{ Btu/hr} * (1 \text{ hr}/60 \text{ min}) / (44.46 \text{ lb/min} * 0.241459 \text{ Btu/lb}^\circ\text{F})$

$$\Delta T = 0.20515975 \text{ }^\circ\text{F}$$

In the DEEAP fan heat calculation for VAV at part loads, the fan power is computed by a relationship where fan power = full load power \* FFL, where FFL = f(part load ratio). The functional relationship is input in terms of a lookup table, and a linear interpolation subroutine returns the multiplier FFL.

Upon determination of the multiplier FFL, the computation proceeds forward from step 1 above. The outcome is highly dependent upon the multiplier FFL, which correlates the part load ratio to the fan power via the unloading mechanism.

In case AE403 for example, the multiplier FFL = 0.10206

Full load horsepower is based upon:

$$\begin{aligned} 2 \text{ inches of total static pressure} &\Rightarrow 10.40466 \text{ lb/ft}^2 \text{ pressure} \times 1300 \text{ cfm} = 13526.058 \text{ lb-ft/min} / 0.7(\text{eff}) \\ &= 19322.94 \text{ lb-ft/min} \end{aligned}$$

In horsepower this is 0.58554 hp, which ultimately gives supply fan temperature rise = 1.0312°F.

But, taking the multiplier FFL by the full load hp = 0.58554 hp × .10206 = 0.0597602124 hp.

This is greater than the 0.0519356 hp determined by the QAS methodology, and ultimately gives

supply fan temperature rise = 0.23609°F, which is greater than the 0.20516°F given by the QAS.

## 6. Other

Included are the inputs to the system calculation module, as well as the final input file generated that goes to EnergyPlus (E+). DEEAP utilizes an interface that a user can input information for buildings, system and plant. The building information is passed to EnergyPlus and it creates an output file. The E+ output file is processed and a number of zone data files and environment files are created that are then passed to the DEEAP system calculator. Also included are the EnergyPlus files (.idf) that were finally used to calculate EnergyPlus (there can be input files that are generated before this that contain certain EnergyPlus template systems that are then parsed by a preprocessor macro to generate the actual input file). After EnergyPlus runs, there is a segment that reads the E+ output and generates the environment and zone loads data files. That data then is utilized to create data files that are sent to the DEEAP system/plant calculation module. The calculation module contains all of the psychrometric and system information that is of interest, and it can be run independently (it is just a .exe). The overall program is designed to run with an interface, but some of the things here required modification outside the scope of the interface, so interface files may not be included for all of the examples. Most of the system and plant models were run just through text (batch processor) files ultimately, because once the loads were generated, only the system section required simulation. So, included are the system and plant input files that can be used to run the calculation as well as the environment data and zone data. Unfortunately, the program was designed for 8760 hours only, so even though the loads and environmental information do not change, there are 8760 hours of data.

## 7. Conclusions and Recommendations

The test suite is useful in calibrating the system simulation for the basic system types included in the specification. It is beneficial to gauge the impact of simulation differences on the overall solution as well as air state at various nodal positions in the supply air loop. There is potential for impact from varying functions utilized to calculate psychrometric properties. Perhaps a tested, verified, and recommended set of algorithms to compute psychrometric functions that are utilized by all practitioners would eliminate psychrometrics calculations as a source of differences between models.

## 8. References

- [1] U.S Department of Energy. (2012). *EnergyPlus Version 7.0.0.036* (computer program) [apps1.eere.energy.gov/buildings/energyplus/](http://apps1.eere.energy.gov/buildings/energyplus/) [Editor's note: This is the version of EnergyPlus that was current as of completion of AAON's simulations for this project.]
- [2] Knebel, D.; Linton, R. (1995). *Thermal Storage Analysis* (computer program). Madison, Wisconsin: Engineering Applications Specialists Inc.
- [3] Neymark, J. (2012). Personal communication with J. Gall and D. Knebel, AAON, Tulsa, Oklahoma. Golden, CO: J. Neymark & Associates, June 22, 2012.
- [4] Neymark, J. (2013). Personal communication with J. Gall and D. Knebel, AAON, Tulsa, Oklahoma. Golden, CO: J. Neymark & Associates, August 19, 2013.
- [5] Gall, J. (2013). Personal communication with J. Neymark, J. Neymark & Associates, Golden, Colorado. Tulsa, Oklahoma: AAON, August 28, 2013.
- [6] Knebel, D. 1983. *Simplified Energy Analysis Using the Modified Bin Method*. ASHRAE, Atlanta GA. (pp. 4-15 through 4-18).

## Appendix A: Pro-Forma Questionnaire for Airside HVAC Tests, October 2014

### *Instructions:*

- *Replace empty boxes with symbols just below, as appropriate.*
  - Possible to use
  - Used to develop simulation trial results
- *Provide additional information and comments as appropriate; add lines as needed.*
- *If specific responses require disclosure of proprietary data not normally available to users in the program's engineering manual, leave the response blank, and include a note to the effect that including a response would require disclosure of proprietary information.*
- *If modeling varies among systems (and/or test cases), specify which systems (and/or test cases) are associated with each response; recopy a query as needed for this purpose (e.g., create 14a, 14b, etc.).*

### **Program name, including version number**

Detailed Energy and Economic Analysis Program, Version 1.1.2

---

### **Your name and organization**

John Gall, AAON Inc.

---

**1. Program status**

- Public domain
- Commercial
- Other (please specify): \_\_\_\_\_

**2. What weather inputs are used by the program for these tests?**

- Provided TMY2 weather data
- Directly entered DB and DP
- Directly entered DB and RH
- Directly entered DB and W
- Directly entered DB and WB

Note: DB = outdoor dry-bulb temperature; DP = outdoor dew-point temperature; RH = outdoor relative humidity; W = outdoor humidity ratio; WB = outdoor wet-bulb temperature.

**2a. If weather inputs provided from TMY2 data, which data are used for calculating outdoor air humidity ratio?**

- DB and DP
- DB and RH
- Other (please specify): Outdoor air humidity ratio is fed forward to the system simulator from EnergyPlus, which is the program responsible for reading the input file. \_\_\_\_\_

**3. How frequently is the HVAC system air density calculated for each system?**

- Once per run, same value applied throughout system
- Once every time step, same value applied throughout system
- Once every time step, varies by location throughout system
- Other (please specify): Moist air density is calculated at multiple locations throughout the system. These densities are computed for each iteration at each time step. \_\_\_\_\_

**4. What air property basis is applied for HVAC system air density calculations for each system?**

- Single fixed air condition (please specify): \_\_\_\_\_
- Multiple fixed air conditions depending upon operating mode or system type (please specify): \_\_\_\_\_
- Air condition at each time step at a single specific system location \_\_\_\_\_ (please specify location): \_\_\_\_\_
- Local air conditions at each time step at each system location
- Other (please specify): \_\_\_\_\_

**5. What air pressure is used in HVAC system air density calculation?**

- Altitude adjusted standard pressure
- Weather file barometric pressure
- Other (please specify): Weather file barometric pressure, but if this is unavailable, altitude adjusted standard pressure. \_\_\_\_\_

**6. Are fan/duct/system pressure differences accounted for in the HVAC air density calculations?**

- Yes
- No

**7. Are system models used for test cases the same as those commonly used by modelers to model similar systems?**

- Fan Coil:         Yes    No  
Single Zone:     Yes    No  
CV Reheat:      Yes    No  
VAV:             Yes    No

**8. How are zone loads defined?**

- Zone loads specified directly from input tables (e.g., Part I, Table 1-2)  
 Using alternative zone definition (e.g., Part I, Section 1.5.5.1.1.5)

**9. Thermostat Control**

- Perfect control  
 On/Off thermostatic control  
 On/Off thermostatic control with deadband  
 Proportional control  
 Other (please specify): \_\_\_\_\_

**10. Were adjustments to zone temperature set points required to attain test case specified zone temperature?**

- Yes, consistent adjustment for all cases  
 Yes, different adjustment for each case  
 No  
 Other, or additional discussion (please specify): \_\_\_\_\_

**11. Were adjustments to CV and VAV system supply air temperature set points required to attain test case specified system supply air temperature?**

- Yes, consistent adjustment for all cases  
 Yes, different adjustment for each case  
 No  
 Other, or additional discussion (please specify): \_\_\_\_\_

**12. How was the zero bypass factor idealization modeled in FC/SZ test cases?**

- Directly, using the program's standard input scheme  
 Using custom supply air delivery temperature inputs for each test case  
 Other, or additional discussion (*describe*): Adjustment of leaving air relative humidity.  
 Was not successful

**13. How was the zero bypass factor idealization modeled in CV/VAV test cases?**

- Directly, using the program's standard input scheme  
 Using custom supply air delivery temperature inputs for each test case  
 Other, or additional discussion (*describe*): Adjustment of the leaving air relative humidity.  
 Was not successful



**14. Which Section 1.6.5.n.3 specified detailed output variables were reported, and at which specified locations?**

Temperatures:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:* \_\_\_\_\_

Humidity Ratios:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:* \_\_\_\_\_

Specific Volumes:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:* \_\_\_\_\_

Enthalpies:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:* \_\_\_\_\_

Mass Flows:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:* Zones \_\_\_\_\_

**15. For reported detailed variables, how were outputs generated?**

Temperatures:  Direct from model  Required unit conversion  Other post processing\*

Humidity Ratios:  Direct from model  Required unit conversion  Other post processing\*

Specific Volumes:  Direct from model  Required unit conversion  Other post processing\*

Enthalpies:  Direct from model  Required unit conversion  Other post processing\*

Mass Flows:  Direct from model  Required unit conversion  Other post processing\*

\* “Other post processing” includes calculations outside of the model, and/or output that is inferred as equivalent or approximately equivalent from another location, e.g., for the SZ system return air fan inlet conditions and zone air conditions might be assumed to be equal or approximately so.

Describe any post-processing (outside-the-model) calculations and/or assumptions used to produce detailed outputs: \_\_\_\_\_

## Appendix B: Modeling Notes for ASHRAE Standard 140

*[Editor's note: Content here summarized from S140outNotes.txt form submitted during the simulation trials.]*

### A. Software Information

1. Vendor: [AAON Inc.](#)
2. Software Name: [AAON Detailed Energy and Economics Analysis Program](#)
3. Version (unique software version identifier): [Version 1.1.2](#)
4. Operating System Requirements:  
[Windows XP, VISTA, 7 w/ .NET Framework 2.0](#)
5. Approximate Hard Disk Space Required for Installation:  
[With all components except weather files 182 MB](#)
6. Minimum RAM Required for Software Operation:  
[128 MB](#)

### B. Alternative Modeling Methods

[NONE](#)

### C. Equivalent Modeling Methods

[NONE](#)

### D. Use of Non-Specified Inputs

[NOTE 1 - Supplementary sensible gain](#)

#### 1.1 Simulated Effect:

[Space sensible gain](#)

#### 1.2 Relevant Sections of the Test Specification:

[Pursuant to section 1.5.5.1.1.5.3 \[see Part I of this document\], adjustment of the supplementary internal gains shall be permitted.](#)

#### 1.3 Non-Specified Input Used:

[The only changes related to the adjustment of the supplemental gains, because there was no apparent control \(in the DEEAP\) over radiative emissivity as set forth in the specification.](#)

#### 1.4 Physical, Mathematical or Logical Justification for use of the Non-Specified Input

[Provision of the appropriate space sensible gains to the system solver routine.](#)

## **E. Omitted Test Cases and Results**

### NOTE 1 - <Omitted Results>

1.1 Cases where Results Were Omitted, and which Results Were Omitted:

Omission of the enthalpy calculations. All cases.

1.2 Explanation for Omitting Test Case Results:

Redundant, and not critical to the analysis routine.

## **F. Changes to Source Code for the Purpose of Running the Tests, Where Such Changes are Not Available in Publicly Released Versions of the Software**

### NOTE 1 - 100% cooling coil leaving relative humidity

1.1 Changes to the source code:

-0% bypass factor, 100% RH

1.2 Test Cases Relevant to Changes in the Source Code:

-All of them

1.3. Explanation of Why the Change Is Not Included in the Publicly Released Version of the Software:

-Typical wet coil assumption is 95% cooling coil leaving RH.

### NOTE 2 – 45°F leaving preheat coil dry bulb temp

2.1 Changes to the source code:

45°F leaving preheat coil dry bulb temperature.

2.2 Test Cases Relevant to Changes in the Source Code:

-All of them

2.3. Explanation of Why the Change Is Not Included in the Publicly Released Version of the Software:

-The typical preheat calculation attempts to meet the coil leaving air temperature setpoint, not an arbitrary 45°F.

### NOTE 3 - Enthalpy Economizer Test

3.1 Changes to the source code:

-Ordinarily the Enthalpy Economizer logic includes a backup dry bulb test. The Economizer will limit outdoor air when enthalpy is less than return air (passes enthalpy test) but outdoor dry bulb temperature is greater than return (fails dry bulb test). The additional dry bulb test was removed for enthalpy economizer testing.

3.2 Test Cases Relevant to the Changes in the Source Code:

-Relevant to cases with Enthalpy Economizer: AE245, AE345, AE445.

3.3. Explanation of Why the Change Is Not Included in the Publicly Released Version of the Software:

-The backup dry bulb test is reflective of economizer control in production HVAC equipment, and better approximates performance.

## **G. Anomalous Results**

NONE

## **Appendix III-B**

### **Modeler Report for HVAC Air Distribution System Tests Designer's Simulation Toolkit (DeST), Version 2.0**

by

**Xin Zhou and Da Yan  
Tsinghua University  
China**

**Tianzhen Hong  
Lawrence Berkeley National Laboratory  
United States**

**Ruhong Wu, Ye Zhang, Xiaoliang Zhang and Ye Liu  
Tsinghua University  
China**

**April 2015**

#### **1. Introduction**

**SOFTWARE NAME: DeST;  
SOFTWARE VERSION (unique software version identifier): 2.0  
SOFTWARE VENDOR: Tsinghua University, China**

**SCHEME program to calculate the control information, HVAC program to calculate the performance and output of each component  
- Timestep: 1 hour**

#### **2. Modeling Assumptions**

- 2.1. The density of air is constant to  $1.2 \text{ kg/m}^3$  and will not change as the temperature.**
- 2.2. The heating exchange efficiency method is used to calculate the coils in DeST.**

#### **3. Modeling Difficulties**

- 3.1. The indoor sensible and latent load cannot be input directly, and the internal heat and wall material should be adjusted to reach the specified sensible and latent load.
- 3.2. For the modeling method of coil, in some cases, the outlet air temperature can only be close to the wanted temperature, so in these test cases, the room temperature will have a little deviation.
- 3.3. In the coil calculation process, DeST uses the heating efficiency method and the default coil type, so the outlet air relative humidity cannot be set to 100% as the test case specified, so the latent load of coil would be larger under the same outlet air temperature.

In the AE104, AE204, AE206 and AE226 cases, it seems results from DeST reflect an overestimation about the latent cooling load. As shown in Figure 3B-1, compared with other building energy modeling programs, results from DeST (indicated as “H” in Figure 3B-1) present a larger latent load.

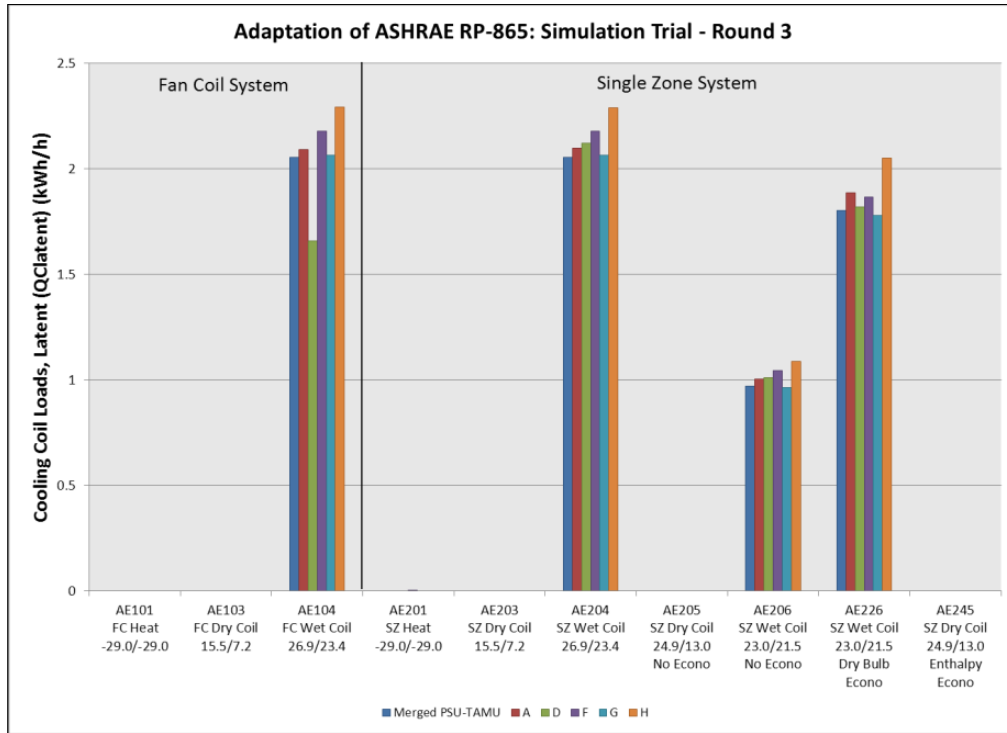


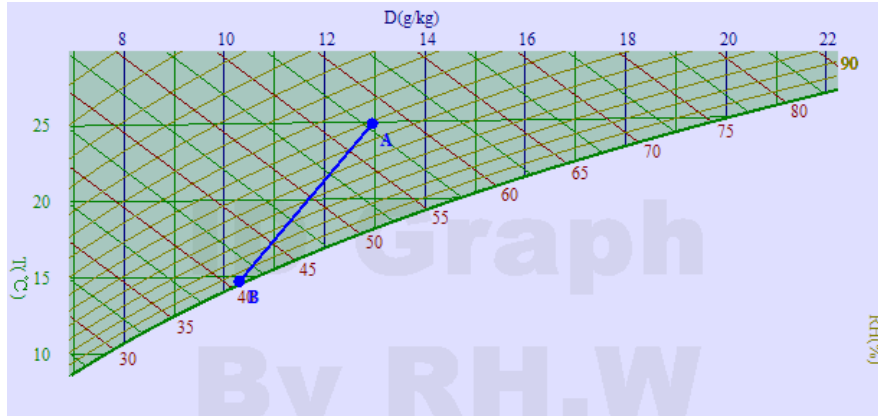
Figure 3B-1. Simulation – Latent Cooling, Round 3

The main reason for this phenomenon is the calculation method in DeST cannot fit the requirement about the wet coil in the test cases exactly. In DeST, the coil type is chosen automatically or by the users, and the heat efficiency method is used to complete the calculation. Under these assumptions, the outlet air relative humidity in DeST cannot be 100%, which means that the specified zero bypass condition cannot be satisfied. Per the situation above, the latent load that the wet coil needs to deal with is larger in DeST.

Taking the case AE104 as an example, in DeST, the inlet air condition is 24.89°C, 0.01295 g/gda, and the outlet air condition is 14.70°C, 0.01030 g/gda. The air handling process (from A to B) is shown in Figure 3B-2. During the calculation process, the heat exchange efficiency ( $E_g$ ) is 0.57, and the contact efficiency ( $E'$ ) is 0.97; these are calculated from:

$$E_g = (T_A - T_B) / (T_A - T_{W1}), \text{ where } T_A \text{ and } T_B \text{ are the temperatures at conditions A and B, respectively, and } T_{W1} \text{ is the water inlet temperature (} ^\circ\text{C)}.$$

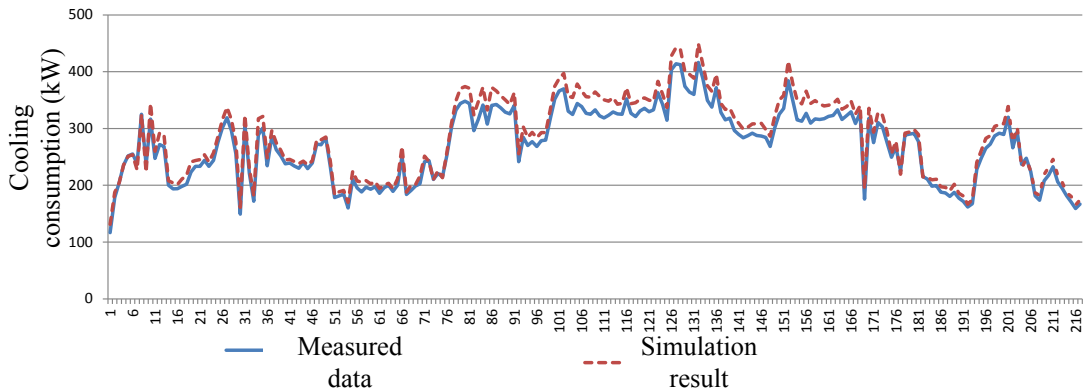
$$E' = 1 - (T_B - T_{BS}) / (T_A - T_{AS}), \text{ where } T_{AS} \text{ and } T_{BS} \text{ are the wet-bulb temperatures at conditions A and B, respectively.}$$



**Figure 3B-2. Air handling process of the wet coil in DeST in case AE103**

In reality, the outlet air relative humidity tends to be less than 100%. The calculation method in DeST is commonly used in real projects and committed to reflect the real situation. Figure 3B-3 shows the comparison between simulation result of DeST and the measured data in a single coil. The simulation result fits the real data quite well, and the average difference is within 5%.

Based on these discussions, we think the zero bypass requirements in test cases simplify the problem. It is better to make some extra explanation about this part.



**Figure 3B-3. Comparison between simulation result of DeST with the measured data from a single coil**

#### **4. Software Errors Discovered, Program Improvements, Input Corrections, and/or Other Comparisons Between Different Versions of the Same Software**

No software errors were found.

An input error was found and fixed in the SZ and CV system economizer tests (AE226, AE245, AE326, AE345). There was a misunderstanding about the systems, which were modeled with only recirculated air passing through the return air fan in the SZ and CV cases. This resulted in no return fan temperature rise in the economizer cases and erroneous return air mass flows which had a small impact of return fan temperature rise. We have replaced the fan position, and updated the results. The effect on total cooling coil load from correcting the input error is negligible when comparing DeST results submitted Nov 21, 2013 versus Jun 12, 2012.

## 5. Results

### *Remaining Minor Disagreements with Quasi-Analytical Solution*

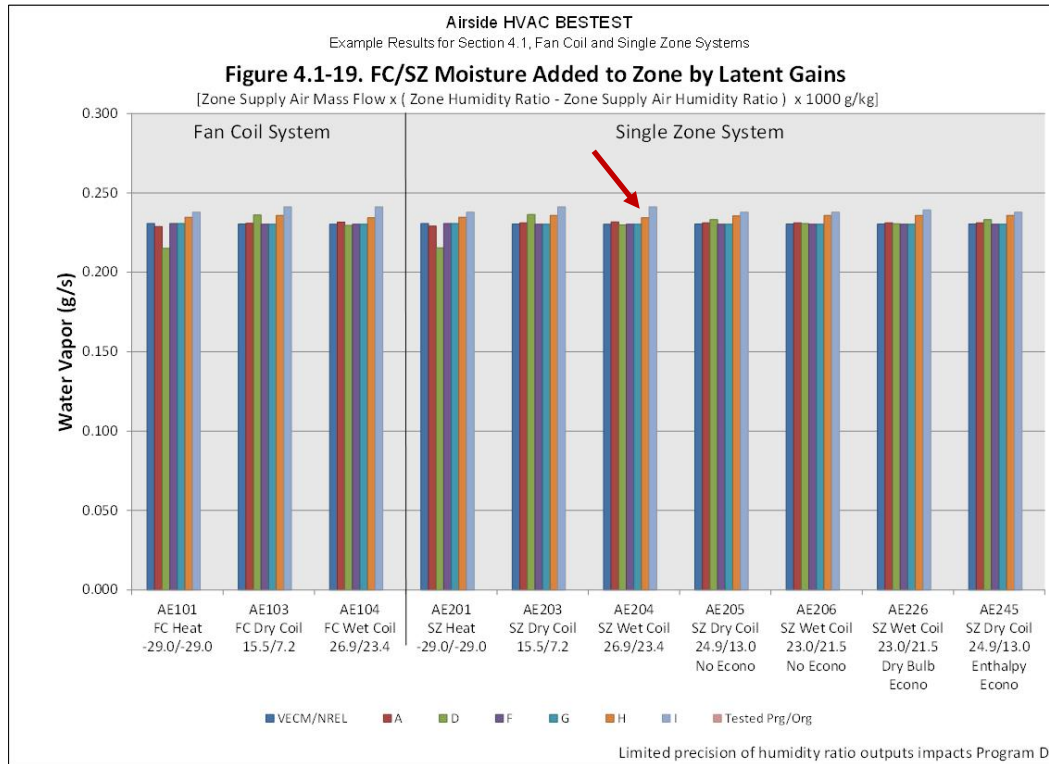
Following is revised summary of remaining minor disagreements identified by the NREL team test spec authors for Simulation Trial Round 3 (8/15/13), with logical reasons for differences described.

### *Latent Cooling Loads*

The predicted latent cooling loads are high by ~10%; see Part IV example results and accompanying electronic files (Results-FCSZ.xlsm and Results-CVVV.xlsm). As discussed in Section 3.3 of this modeler report, this is caused by DeST not modeling the zero bypass factor idealization of the test cases. Another contributing factor to this difference is the moisture added to the zones is about 3% higher than the quasi-analytical solution (identified as “VECM/NREL” in Figure 3B-4). Figure 3B-4 (DeST is “H” in this figure) shows the mass of water vapor added to the zone in the FC/SZ cases. For developing the figure, this has been calculated from other reported results as:

$$(\text{zone supply air mass flow rate}) \times ((\text{zone air humidity ratio}) - (\text{supply air humidity ratio}))$$

To the problem about a higher mass of water vapor added to zone, the main reason may be that DeST uses the enthalpy of saturated vapor to convert the latent gains to moisture, and the value is constant to 2500 kJ/kg. The reason is that it is used to present the property of moist air, not the transfer process.



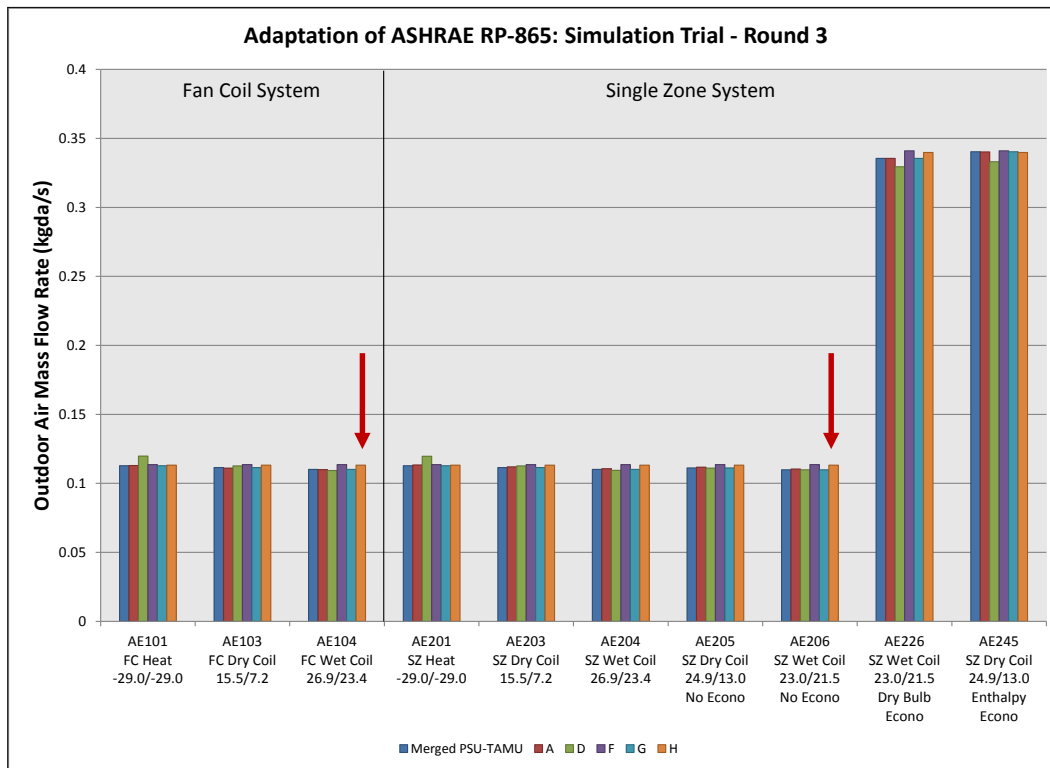
**Figure 3B-4. Moisture added to zone by latent gains**

### Mass Flow Differences

Outdoor air mass flow, presented in Figures 3B-5 and 3B-6, is 1%-3% higher than the quasi-analytical solution (labeled as “Merged PSU-TAMU” in these figures) in all non-economizer cases. Supply air mass flow is shown in Figures 3B-7 and 3B-8. Supply air mass flow is 3% high in the FC and SZ system heating cases, 0.5%-2% high in the VAV system cases, and 2%-4% low in the CV system heating and cooling cases. These differences result from DeST (Program H) calculating supply, exhaust, and outdoor air mass flow rates at a single set of standard air conditions for all test cases, which results in mass flow rates that do not change between test cases. This issue impacts all coil loads, and makes diagnosis of any other issues within these range of differences difficult.

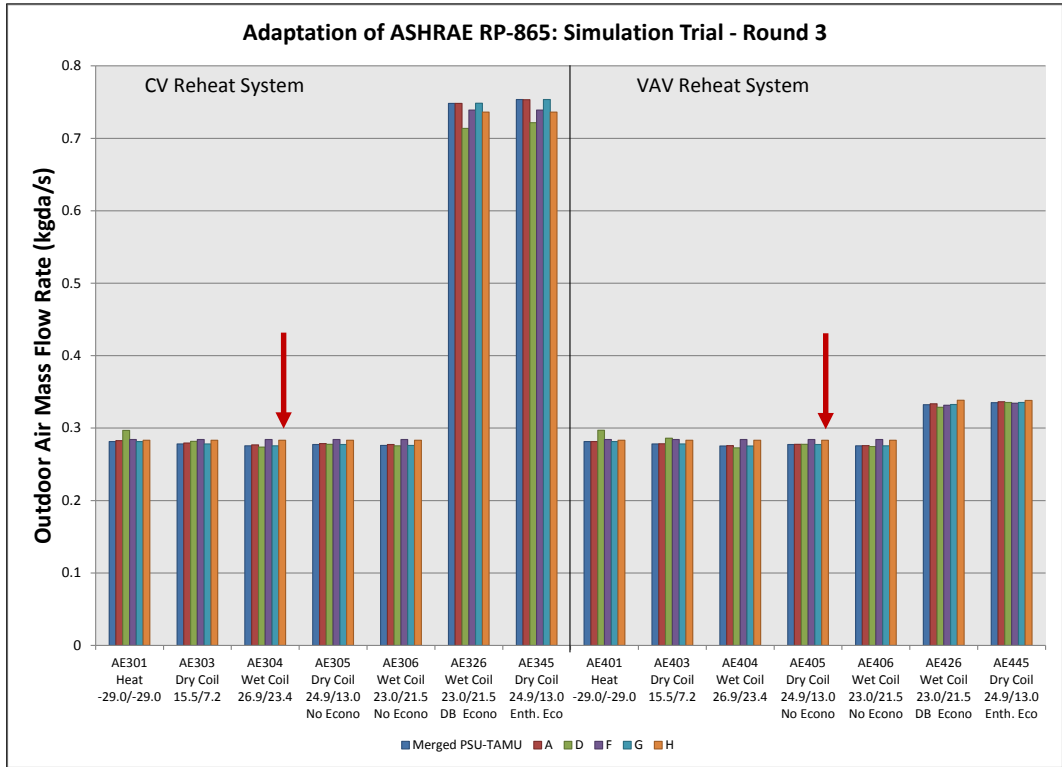
Result differences versus the quasi-analytical solutions (see final results of Part IV and accompanying final results files) that likely stem from this include:

- The 21% high CV system preheat coil load in Case AE301. This may be a result of the 1% higher mass flow rate of cold outdoor air combined with a system supply air mass flow rate that is 4% lower, which results in the mixed air temperature being lower and preheat coil loads larger.
- The 2%-4% low CV system sensible cooling load in all cooling cases. This may be in part the result of the 2% low system mass flow rate.
- The 3%-13% low CV system zone reheat coil load in all cases. This may be the result of the ~2%-4% low system mass flow rate.

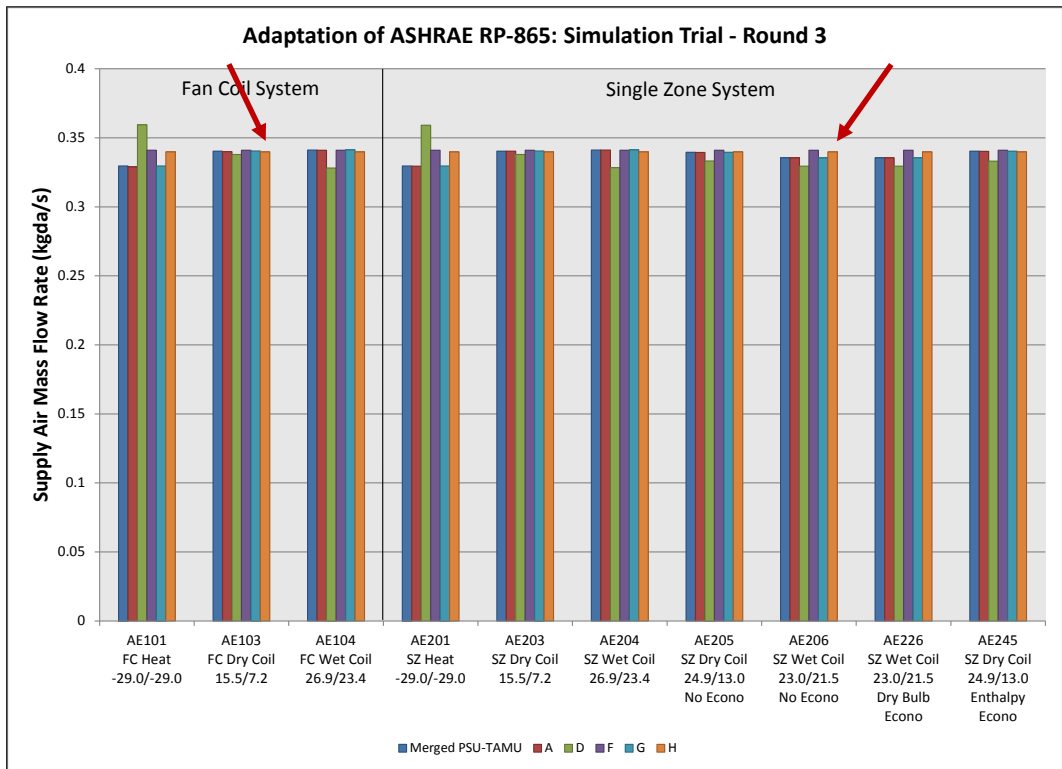


**Figure 3B-5. FC/SZ Outdoor Air Mass Flow Rate**





**Figure 3B-6. CV/VAV Outdoor Air Mass Flow Rate**



**Figure 3B-7. FC/SZ Supply Air Mass Flow Rate**

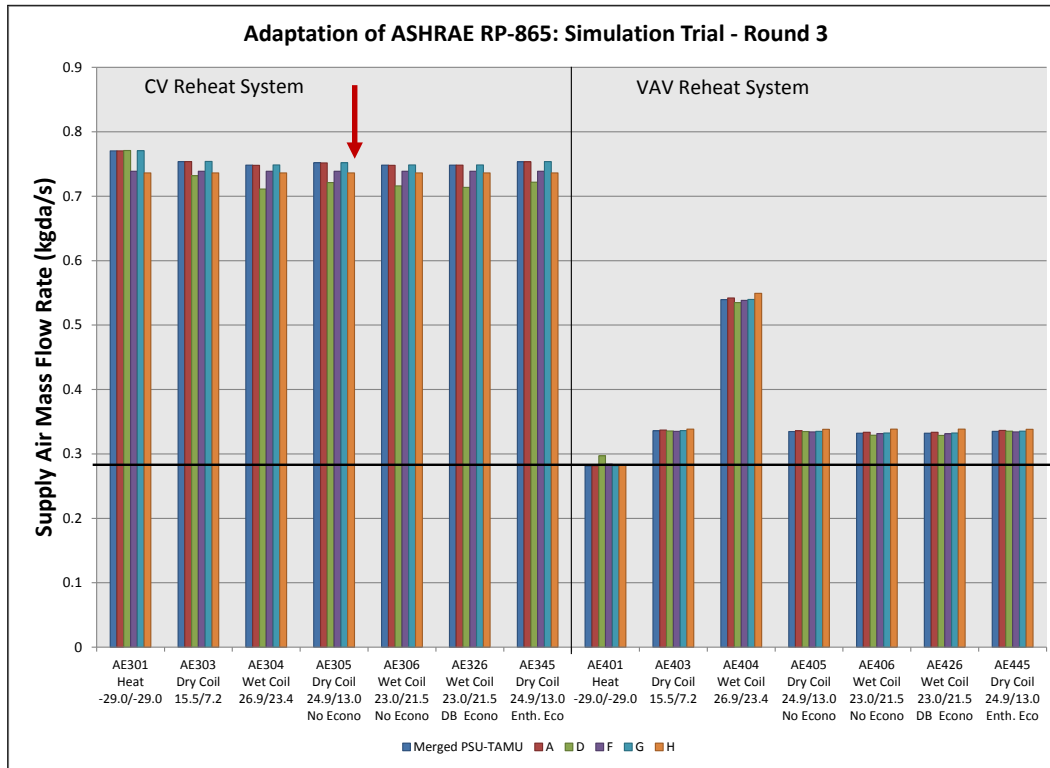


Figure 3B-8. CV/VAV Supply Air Mass Flow Rate

### *SZ and CV System Return Fan Heat Gain*

Compared with the quasi-analytical solution (QAS) results (see final results of Part IV and accompanying results files), the SZ and CV return fan heat gain is 8% high for case AE201 and 6%-16% low for all other SZ and CV cases. This stems from the SZ and CV return fans being modeled as constant volume fans whereas the QAS and specification vary the SZ and CV return fan volumetric flow as required to return the supply air mass flow minus zone exhaust air mass flow. These differences are large in relative terms but are only a 0.03°C to 0.06°C difference in return air temperature rise. Given the small impact on coil loads this seems like a reasonable modeling simplification.

*[Editor's note: Both models discussed here approximate, but do not exactly replicate, real constant-volume system behavior. In a real system for given fan speeds the associated volumetric capacities, pressure changes, efficiencies, and therefore mass flows of all system fans would balance according to the individual fan curves.]*

#### 6. Other (optional)

- None

#### 7. Conclusions and Recommendations

The test cases can include more component model in one test series and focus both air and water side, which may make it more practical.

## 8. References

- [1] DeST team of Tsinghua University. Simulation Method for Building Thermal Environment - DeST. China Architecture and Building Press, 2006. (In Chinese)
- [2] Yan Da and Yi Jiang. An Overview of an Integrated Building Simulation Tool – Designer’s Simulation Toolkit (DeST), Ninth International IBPSA Conference.
- [3] Yan Da, Xia Jianjun, Tang Waiyin, etc. DeST-An Integrated Building Simulation Toolkit Part I: Fundamentals, BUILDING SIMULATION 1(2), 2007,6: 95-110
- [4] Xiaoliang Zhang, Jianjun Xia, ZIyan Jiang, etc. DeST—An Integrated Building Simulation Toolkit Part II: Applications, Building Simulation, 2008, 1(3): 193-209
- [5] Zhu D, Yan D, Wang C, and Hong T (2013), Comparison of building energy modeling programs: DeST, EnergyPlus and DOE-2. Building Simulation, 2013(6): 323-335
- [6] Zhou X, Hong T, and Yan D (2014), Comparison of HVAC system modelling in EnergyPlus, DeST and DOE-2.1E. Building Simulation, 2014 (7):21-33

## Appendix A: Pro-Forma Questionnaire for Airside HVAC Tests, April 2015

### *Instructions:*

- *Replace empty boxes with symbols just below, as appropriate.*
  - Possible to use
  - Used to develop simulation trial results
- *Provide additional information and comments as appropriate; add lines as needed.*
- *If specific responses require disclosure of proprietary data not normally available to users in the program's engineering manual, leave the response blank, and include a note to the effect that including a response would require disclosure of proprietary information.*
- *If modeling varies among systems (and/or test cases), specify which systems (and/or test cases) are associated with each response; recopy a query as needed for this purpose (e.g., create 14a, 14b, etc.).*

### **Program name, including version number**

DeST 2.0

---

### **Your name and organization**

Da Yan, Tsinghua University, China

---

**1. Program status**

- Public domain
- Commercial
- Other (please specify): \_\_\_\_\_

**2. What weather inputs are used by the program for these tests?**

- Provided TMY2 weather data
- Directly entered DB and DP
- Directly entered DB and RH
- Directly entered DB and W
- Directly entered DB and WB

Note: DB = outdoor dry-bulb temperature; DP = outdoor dew-point temperature; RH = outdoor relative humidity; W = outdoor humidity ratio; WB = outdoor wet-bulb temperature.

**2a. If weather inputs provided from TMY2 data, which data are used for calculating outdoor air humidity ratio?**

- DB and DP
- DB and RH
- Other (please specify): \_\_\_\_\_

**3. How frequently is the HVAC system air density calculated for each system?**

- Once per run, same value applied throughout system
- Once every time step, same value applied throughout system
- Once every time step, varies by location throughout system
- Other (please specify): \_\_\_\_\_

**4. What air property basis is applied for HVAC system air density calculations for each system?**

- Single fixed air condition (please specify): \_\_\_\_\_
- Multiple fixed air conditions depending upon operating mode or system type (please specify): \_\_\_\_\_
- Air condition at each time step at a single specific system location \_\_\_\_\_ (please specify location): \_\_\_\_\_
- Local air conditions at each time step at each system location
- Other (please specify): Air density fixed to 1.2 (kg/m<sup>3</sup>).

**5. What air pressure is used in HVAC system air density calculation?**

- Altitude adjusted standard pressure
- Weather file barometric pressure
- Other (please specify): \_\_\_\_\_

**6. Are fan/duct/system pressure differences accounted for in the HVAC air density calculations?**

- Yes
- No

**7. Are system models used for test cases the same as those commonly used by modelers to model similar systems?**

- Fan Coil:         Yes    No  
Single Zone:     Yes    No  
CV Reheat:      Yes    No  
VAV:             Yes    No

**8. How are zone loads defined?**

- Zone loads specified directly from input tables (e.g., Part I, Table 1-2)  
 Using alternative zone definition (e.g., Part I, Section 1.5.5.1.1.5)

**9. Thermostat Control**

- Perfect control  
 On/Off thermostatic control  
 On/Off thermostatic control with deadband  
 Proportional control  
 Other (please specify): \_\_\_\_\_

**10. Were adjustments to zone temperature set points required to attain test case specified zone temperature?**

- Yes, consistent adjustment for all cases  
 Yes, different adjustment for each case  
 No  
 Other, or additional discussion (please specify): \_\_\_\_\_

**11. Were adjustments to CV and VAV system supply air temperature set points required to attain test case specified system supply air temperature?**

- Yes, consistent adjustment for all cases  
 Yes, different adjustment for each case  
 No  
 Other, or additional discussion (please specify): \_\_\_\_\_

**12. How was the zero bypass factor idealization modeled in FC/SZ test cases?**

- Directly, using the program's standard input scheme  
 Using custom supply air delivery temperature inputs for each test case  
 Other, or additional discussion (*describe*): the coil type is chosen automatically or by the users, and the heat efficiency method is used to complete the calculation  
 Was not successful

**13. How was the zero bypass factor idealization modeled in CV/VAV test cases?**

- Directly, using the program's standard input scheme  
 Using custom supply air delivery temperature inputs for each test case  
 Other, or additional discussion (*describe*): the coil type is chosen automatically or by the users, and the heat efficiency method is used to complete the calculation  
 Was not successful

**14. Which Section 1.6.5.n.3 specified detailed output variables were reported, and at which specified locations?**

Temperatures:      All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:*\_\_\_\_\_

Humidity Ratios:    All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:*\_\_\_\_\_

Specific Volumes:    All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:*\_\_\_\_\_

Enthalpies:          All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:*\_\_\_\_\_

Mass Flows:          All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:*\_\_\_\_\_

**15. For reported detailed variables, how were outputs generated?**

Temperatures:      Direct from model    Required unit conversion    Other post processing\*  
Humidity Ratios:    Direct from model    Required unit conversion    Other post processing\*  
Specific Volumes:    Direct from model    Required unit conversion    Other post processing\*  
Enthalpies:         Direct from model    Required unit conversion    Other post processing\*  
Mass Flows:         Direct from model    Required unit conversion    Other post processing\*

\* “Other post processing” includes calculations outside of the model, and/or output that is inferred as equivalent or approximately equivalent from another location, e.g., for the SZ system return air fan inlet conditions and zone air conditions might be assumed to be equal or approximately so.

Describe any post-processing (outside-the-model) calculations and/or assumptions used to produce detailed outputs:\_\_\_\_\_

## Appendix B. Modeling Notes for ASHRAE Standard 140

*[Editor's note: Content here summarized from S140outNotes.txt form submitted during the simulation trials.]*

### A. Software Information

1. Vendor: [Tsinghua University](#)
2. Software Name: [DeST](#)
3. Version (unique software version identifier): [2.0](#)
4. Operating System Requirements:  
[All](#)
5. Approximate Hard Disk Space Required for Installation:  
[1GB](#)
6. Minimum RAM Required for Software Operation:  
[128 MB](#)
7. Minimum Display Monitor Requirements:  
[VGA with 600x800 resolution and 256 colors](#)
8. Other Hardware or Software-Related Requirements:  
[\(none\)](#)

### B. Alternative Modeling Methods

NOTE 1 - [Four Pipe Fan Coil \(FC\) System Cases \(Case AE101: Base Case, High Heating 1\) \(section 1.5.5.1.1 \[see Part I of this document\]\)](#)

#### 1.1 Simulated Effect:

[System air flow, and heat and mass balance in a four pipe fan coil system, base case, high heating test](#)

#### 1.2 Optional Settings or Modeling Capabilities

##### 1.2.1 Alternate Zone Specification

[The zone identification method in section 1.5.5.1.1.5 \[see Part I of this document\] is used to specify the zone and model all test cases.](#)

#### 1.3 Setting or Capability Used:

[The other settings mentioned in the document are used.](#)

### C. Equivalent Modeling Methods

[NONE](#)



**D. Non-Specified Inputs**

NONE

**E. Omitted Test Cases and Results**

NONE

**F. Changes to Source Code for the Purpose of Running the Tests, Where Such Changes are Not Available in Publicly Released Versions of the Software.**

NONE

**G. Anomalous Results**

NONE

## **Appendix III-C**

### **Modeler Report for HVAC Air Distribution System Tests Program: DOE-2.2-48I**

by

**M. Kennedy,  
Mike D. Kennedy Inc., United States**

**April 2015**

#### **1. Introduction**

**Software:** JJ Hirsch DOE-2.2, Version 48I  
**Authoring Organization:** James J. Hirsch & Associates  
12185 Presilla Road  
Camarillo, California  
**Authoring Country:** USA

This report describes the modeling methodology used to produce results for the Adaptation of ASHRAE RP 865 Airside Mechanical Equipment Modeling Test Cases. Modeling was conducted using the JJ Hirsch DOE-2.2 software developed by LBNL/University of California, and James J. Hirsch & Associates. The specifications for the test suite are described in *Adaptation of ASHRAE RP 865 Airside HVAC Equipment Modeling Test Cases for Inclusion in ASHRAE Standard 140, Final Draft Test Specification, Sep 2014*.

The JJ Hirsch DOE-2.2 software is developed by James J. Hirsch & Associates based upon the DOE-2.1 building energy simulation program developed by LBNL and the University of California. The software operates as a stand-alone program and as the calculation engine used in the eQUEST building energy use analysis tool.

#### **2. Modeling Assumptions**

The test case specification required several adjustments to the program inputs to produce the required simulation behavior. Major assumptions and choices made in order to implement the test cases are as follows.

##### ***System Type***

The fan coil system (AE100 series) tests were modeled using the DOE-2.2 Fan Coil (FC) system type. The FC model forces the fan to be the first element of the system and also forces the fan and motor to be in the air-stream so it was not possible to exactly model the test specification of a draw-through fan. As a result the latent coil loads and detailed output temperatures deviate slightly versus what would occur if a blow-through fan were modeled. Since the FC model assumes fan and motor are both in the air-stream, the combined fan and motor efficiency was set to 70%, so the resulting fan and motor heat imparted to the air stream was equivalent to the 70% fan mechanical efficiency specified by the test case. The DOE-2.2 Packaged Single Zone (PSZ) system would have allowed closer adherence to the test specification but there was interest in testing the FC system type.

Single-zone system (AE200 series) test cases were modeled using the DOE-2.2 Packaged Single Zone (PSZ) system type. This system type allowed the test case configuration to be modeled as specified.

The constant volume system test cases (AE300 series) were modeled using the DOE-2.2 Reheat Fan System (RHFS). This system allowed these cases to be modeled as specified and is the most suitable model to capture this system type.

The variable volume system test cases (AE400 series) were modeled using the DOE-2.2 Variable Air Volume (VAVS) system. This system allowed the test cases to be modeled as specified and is the most suitable model to capture this system type. In the future the Packaged Variable Air Volume (PVAVS) system type should be tested too.

### ***Weather Data***

The test case specification specifies fixed outdoor air conditions for each test case and as an alternate provides TMY2 weather data. DOE-2.2 weather files were created from the provided TMY2 weather but it was determined that the DOE-2.2 packed weather file format limits precision of the dry-bulb to the nearest whole degree F, which for the moist air cases can impact the resulting humidity ratio and resulting latent loads. To minimize impacts resulting from deviations in the weather data, test cases were simulated using the model design-day facility, which allowed the test case specified dry- and wet-bulb temperatures to be entered directly with full precision. This is a non-standard method of generating model results, but appears to work well, except when economizer operation is specified (see Modeling Difficulties).

### ***Zone Loads***

The test cases specify zone loads as fixed heating or cooling loads, but do not necessarily require the modeling of physical zones. For programs that cannot model zones as pure heating or cooling loads without describing a physical zone, Part I, Section 1.5.5.1.1.5 specifies an alternate zone definition with areas and constructions. Per that section, zone surfaces and internal gains were defined and then the sensible internal gain input(s) were varied for each test case until the zone load(s), as reported by the DOE-2.2, LS-A output report, agreed with the specified zone heating or cooling load for the test case.

### ***Set point Offsets***

DOE-2.2 requires some amount of thermostat throttling range for stable operation. To achieve the test case specified zone air temperature, the input zone air temperature set points were adjusted to achieve zone air temperatures, as reported by the hourly zone report variable 31 (TAVE), which agreed with the test case requirements. The required temperature adjustments (offset) are different for each test case and system type. They have been determined using a trial and error process so that reported zone temperatures are exactly as specified in each test case. The precision of this adjustment was constrained by the DOE-2.2 zone air temperature outputs, which are limited to a tenth of a degree Fahrenheit.

### ***Bypass Factor***

The test cases require that bypass factor equals zero. DOE-2.2 has a coil bypass factor input that allows a minimum entry of 0.001. However, the DOE-2 FC and SZ models utilize a fixed set point for the supply air temperature during the cooling portion of the hour and cycle as required to meet the zone loads. Several control options were tried. Setting "COOL-CONTROL=WARMEST" did not seem to change the delivered temperature to the extent needed. A trial and error process was implemented to adjust the supply air temperature in each test case (using the MIN-SUPPLY-T keyword) to achieve a bypass factor less than 0.01 (as reported by system hourly report variable number 56 – system bypass) while still maintaining the zone temperature (as reported by the zone hourly report variable number 31 – average

zone temperature). The adjustments were only made to the cases with latent cooling load and worked well, though as the supply air temperature increased (effective increase to coil temperature) to get fully saturated leaving air at the required leaving air temperature, the coil's spare capacity decreased and the run period needed to be extended to 4 days from 1 to reach steady state conditions.

### ***Fan Power***

The test specification includes variable volume return fans for the single zone (SZ) and constant volume zone reheat system (RHFS), and variable volume supply and return fans for the VAV system. A custom fit fan power curve was created which varies the fan power with the cube of the flow ratio (actual to design). This curve was assigned to the return fan in the CV test cases and to the supply and return fans in the VAV test cases. For the SZ cases, the DOE-2.2 PSZ system model does not allow the return fan power to be specified based upon a curve; however, because DOE-2.2 uses the same air density for all locations in the system, the SZ return fan volumetric flow does not vary from design and the custom curve would have no effect. For the CV test cases, the DOE-2.2 RHFS model allows a fan power curve to be entered for the return fan, but always returns unity since the return fan is assumed to always operate at design flow.

DOE-2.2 does not have a full set of inputs for return fan efficiency. When specifying static pressure and return fan efficiency it only allows the specification of total efficiency. In these runs, DOE-2.2 assumed electrical efficiency (~90%) to calculate a mechanical efficiency, as reported by the DOE-2.2 SV-A report, from the total efficiency. Initially we assumed that DOE-2.2 placed the return fan motor outside the airstream, since the supply fan was specified as such. To get the proper return fan mechanical efficiency as reported in the DOE-2.2 SV-A report, the return fan total efficiency was adjusted (~63%) yielding a return fan mechanical efficiency reported on the SV-A report of 70%. In Round 3b, it was determined DOE-2.2 assumes that when the return fan is specified in this way the return motor is assumed to be in the air stream. So the return fan total efficiency was set to 70% and SV-A now reports a mechanical efficiency of 78% but total efficiency is 70%.

### ***Detailed Output Calculations***

Generating the required detailed output involved using DOE-2.2 hourly report variables directly where available and in combination with manual analysis to calculate other required values. There are detailed hourly output variables that directly output most of the required temperature and humidity ratio results. Temperature and humidity ratio at some system locations and air mass flow rates at all locations required manual calculations to get the required values. Details of the calculations used follow.

### ***Mixed Air Temperature***

The temperature of air entering the system is an available output (system hourly report variable 3, "TM") and is the same as the mixed air temperature if the fan is configured as a draw-through system per the specification and there is no preheat. The coil entering air temperature was used directly for all SZ system test cases, and all CV and VAV system cooling test cases. For the CV and VAV heating cases, AE301 and AE401, the preheat coil operates. The program-reported air entering the cooling coil is after the preheat coil and therefore is not the same as the mixed air temperature, so the mixed air temperature is not reported for these test cases.

The DOE-2.2 fan coil system model used in the FC cases (AE101, AE103, and AE104) forces the fan to have a blow-through configuration, and the air temperature entering the coil is the mixed air temperature plus the supply fan temperature rise. A separate output (Report SV-A, Fan-Delta-T) is available, which provides the delta-T across the supply fan that was subtracted from the entering coil temperature to get the mixed air temperature.

### *Mass Flow Rates*

Mass flow rate is not directly available from the program and the required calculations here were not verified with the program authors, so should be considered tentative. The program provides the mixed air density for each hourly time step. From looking at the provided DOE-2.2 source code, this mixed air density is calculated each hour. The system keeps a volumetric balance and uses the mixed air density in combination with the volumetric flow for all energy calculations. The mixed air, heating and cooling coil leaving air, and supply air flow all are equal to the supply CFM, and the mass flow rate is assumed to be the product of the mixed air density and the supply CFM. The program calculates mixed air conditions as the CFM weighted average of the return and outdoor conditions with the outdoor CFM being equal to the exhaust CFM. The implicit outdoor air mass flow rate is therefore the product of the outdoor air CFM ratio, the supply CFM, and the mixed air density. This calculation was used to calculate the detailed mass flow results.

### ***Other***

Additional information is included in the pro-forma questionnaire responses; see Appendix A of this supplementary modeler report.

## **3. Modeling Difficulties**

### *FC System Model*

The DOE-2.2 Fan Coil (FC) model does not allow the fan to be located after the heating and cooling coils. Only a blow-through configuration with the fan before the coils is allowed. The main impact of this is to slightly decrease latent cooling loads as the coil only needs to cool the supply air to the required supply air temperature whereas with the fan after the coils, the air must be cooled below the required supply air temperature so that when fan heat is added the supply air is the correct temperature. The FC test cases could have been modeled better with the DOE-2.2 Package Single Zone (PSZ) model, but it was decided testing both the FC and PSZ models was worthwhile. The model author stated they recommend not using the fan coil (FC) or other DOE-2.2 zonal systems, but the model documentation does not include this comment.

### *RHFS System Model*

There was difficulty in maintaining system supply air temperature precisely at 55°F when running a dry bulb economizer with wet ambient air. This issue is a coil capacity issue rather than a control issue, and arises because design-day runs were used (see “Weather Data” above). At the conditions of the AE326 test case (ODB = 23°C, OWB = 21.525°C), the dry bulb economizer should save a small amount of sensible cooling but increase latent cooling coil load substantially. This was not an issue with the SZ or VAV systems.

As explained by the program author (Hirsch 2013): “... the sizing calculations do not take economizer operation into account when calculating the coil capacity; the capacity is calculated using the design OA quantity. The design sizing calculations assumed an outside air ratio of 0.385 as reported in SV-A. Given this outside air ratio, the run without an economizer held 55°F supply T. In the run with an economizer, the drybulb was sufficient to open the economizer 100%, but the outdoor wetbulb was forced to be high (70°F), so that the mixed air enthalpy was higher than with the economizer at minimum position. This overloaded the cooling coil, and the supply air temperature rose to 56.3°F. Because the sizing calculations assume the economizer is at minimum position in both cases, the coil size is the same in both cases. If the SYSTEM command keyword MIN-OUTSIDE-AIR = 1.0 is specified, then the sizing calculations will

upsized the cooling coil for the 100% outside air fraction, and the supply T is maintained at 55°F in the economizer run case. However, most designers would likely not utilize a 100% OA sizing condition in humid climates.”

To generate coil loads reflecting 100% outdoor air, the minimum outside air fraction was set to 100% for the economizer runs. This solution would not work in a simulation that included non-economizer hours with cooling, but the capacity issue would not arise in a typical annual simulation either.

#### 4. Software Errors Discovered

##### *FC System Model*

Zone latent gains in the Round 3 and earlier results were not properly accounted for in the FC system model type. The rise in humidity ratio between the supply (variable name WCOIL) and zone air or return air (variable name WR) is near zero even with large latent gains specified in the zone; see Program D in Figure 3C-1. The single zone system (PSZ) results for the same conditions showed a rise in humidity ratio across the zone, similar to other programs (figure not shown). Figure 3C-2 shows the resulting difference in Program D predicted cooling coil latent load for the FC system (Case 104), while the latent coil loads for the PSZ system (AE200-series cases) have good agreement with the other program results.

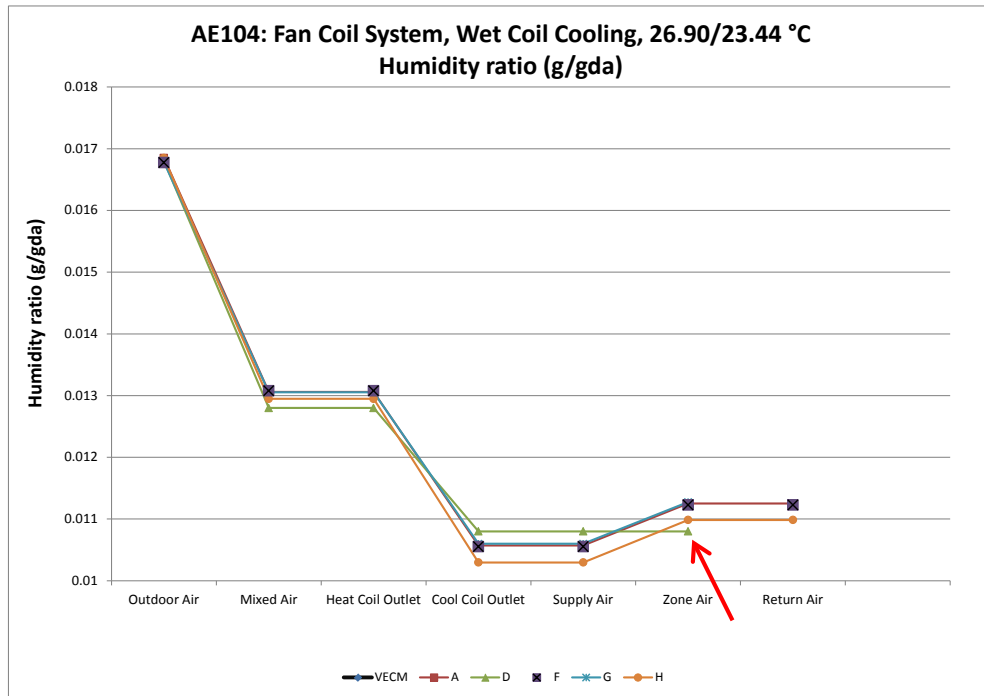
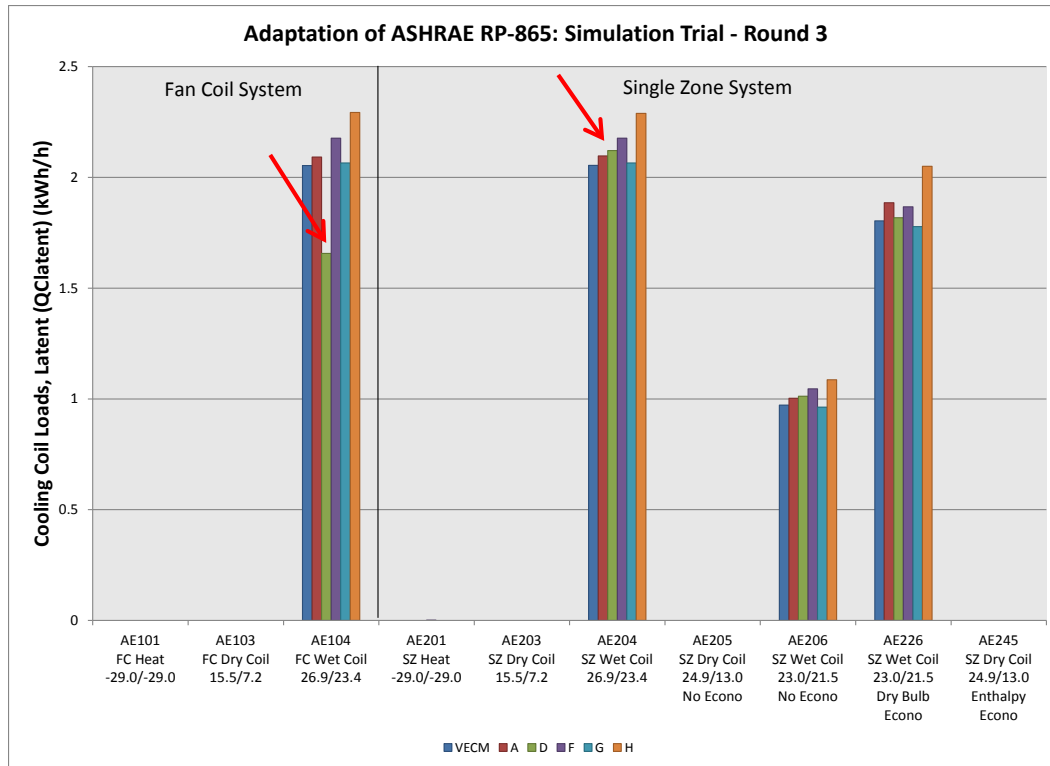


Figure 3C-1. Humidity Ratio at Various System Locations, Case AE104, Simulation Trial Round 3 (June 2013)



**Figure 3C-2. Cooling Coil Latent Loads for the FC and SZ System Cases, Simulation Trial Round 3 (June 2013)**

After reporting this issue the software developer discovered two bugs related to zone latent gains, which impact the FC system model, one of which was primarily responsible for the results disagreements. The primary bug caused the zone latent gain to be low by a factor of 60 due to a conversion from hours to minutes being skipped. The other bug did not impact this case, but was found while trying to explain the issue. According to the code author, the air-handler model used by the FC system was using the sum of the system zone volumes rather than the specific zone volume when calculating the overall rise in fan coil unit humidity due to zone gains. This error is minimal or zero for an FC system with a single zone with the zone multiplier equal to 1, but impacted building descriptions with more than one zone that used this routine. (Hirsch 2013, 2014)

## 5. Input / Modeling Changes

The simulation field trials were conducted in several rounds. The initial FC and SZ case models were created in round 1. The initial CV and VAV case models were created in round 2. Subsequent rounds involved improvement of the initial models. Changes were limited to input changes and updating the program for the bug discussed in Section 4 of this modeler report. No changes to the simulation program were made that have not been included in the distributed program. Changes made to the inputs were:

### 5.1 Changes Made for Production of Round 3 Results (May 2013)

- The number of days specified to calculate the results increased from 1 to 4.
- Changed fan power curve function from square of flow to cube of flow.

### 5.2 Changes Made for Production of Round 3a Results (January 2014)

- Updated simulation program to version DOE-2.2-481 from DOE-2.2-47h2

- Set minimum outside air to 1 for CV system economizer runs (AE326 and AE345) so that coil loads would be correct (see Modeling Difficulties for details).

### 5.3 Changes Made for Production of Round 3b Results (March 2014)

- Set minimum outside air to 1 for SZ system economizer runs (AE226 and AE245) so that coil loads would be correct (see Modeling Difficulties for details). Adjusted system supply air temperature (see Bypass Factor in Modeling Assumptions for details)
- Changed return fan total efficiency from ~63% to 70%. SV-A now reports mechanical efficiency of 78% but return fan delta T is now correct. (see Fan Power in Modeling Assumptions for details)

### 5.4 Changes Made for Production of Final Round Results (March 2015)

- Changed test case input wet-bulb temperatures to agree with new test specification. This variation is a maximum of 0.002°C in the test cases and had negligible effect on results.

## 6. Results

Updated results for the specified test cases are provided with the file Sec5-5out-DOE22-12Mar2015.xls [this data appears on tab “DOE22” in the Results-FCSZ/CVVV.xlsm files]. Additional results using TMY2 weather file inputs are provided with the file Sec5-5out-DOE22-TMY-14Apr2015.xls [this file is included with the accompanying files within subfolder “\Part IV\InpFiles-Airside-HVAC\DOE22\AdditionalOutput].

DOE-2.2 results are comparable to results from other programs and as shown in Figures 3C-3 through 3C-16 [see after Section 8 of this modeler report]. The final DOE-2.2 results are labeled “DOE-2.2/NREL” and have been calculated using the DOE-2.2 design-day runs. “DOE-2.2/TMY” labeled results are DOE-2.2 results using TMY2 weather inputs.

Figures 3C-3 and 3C-4 present the outdoor air temperature and humidity ratio results. The predicted outdoor air temperature and humidity ratios from the design-day versus TMY2 weather file inputs are clearly different and affect other system results. The effect is mostly small; however, the higher outdoor humidity ratio when using weather file AE106 (DOE-2.2/TMY results) noticeably increases latent coil load for cases AE206, AE226, AE306, AE326, AE406, and AE426 over the design-day results (DOE-2.2/NREL) as shown in Figures 3C-9 and 3C-10.

Figures 3C-5 through 3C-11 present the heating and cooling coil load results. There are small persistent differences in heating and cooling coil loads that appear to primarily result from differences in the calculation of system supply air and outdoor air mass flow. DOE-2.2 calculates the air density at the mixed air node before the heating and cooling coils (but after the preheat coil) and uses this density to calculate mass flow rates throughout the system. In Figure 3C-10, the DOE-2.2/NREL latent coil load for Case AE326 appears low. This is likely a direct consequence of the low supply air mass flow and low outdoor air mass flow predicted by the program in this case as seen in Figures 3C-13 and 3C-15. The DOE-2.2/TMY results do not show this same low latent coil load prediction due to the higher case humidity ratio, which is an offsetting issue.

Figures 3C-12 and 3C-13 summarize the supply air mass flows. The system supply air mass flows for the FC, SZ, and CV test cases trend higher than the other example results in the heating test cases, and trend lower than the other example results in the cooling test cases. This is consistent with DOE-2.2 using the mixed air density with the supply air volumetric flow to calculate the system supply mass flow rate. In the heating case results, the FC and SZ cases (AE101, AE201), have mixed air that is colder (~39°F) and



denser than air entering the draw-through supply fan (~85°F) and using the mixed air density results in a higher mass flow of supply air than the QAS<sup>1</sup> model. In contrast, in the CV heating case (AE301), the mixed air temperature is 38°F and the air entering the supply fan is 45°F, and the resulting mass flow is very close to the QAS model. The heating case supply air mass flow variation does not impact heating coil loads significantly; the heating coil load differences apparent in Figures 3C-5 and 3C-6 are the result of the outdoor air mass flow as discussed below.

In the cooling test cases, the mixed air is warmer and less dense than the air entering the supply fan (having just exited the cooling coil), so the system supply air mass flow rates are lower than the other programs for the FC, SZ, and CV cooling test cases. In the CV cooling test cases the lower supply air mass flow combined with a fixed supply air temperature results in smaller cooling and reheat coil loads.

Figures 3C-14 and 3C-15 summarize the outdoor air mass flow. In DOE-2.2 the outdoor air mass flow is determined from the zone exhaust CFM and the air density at the mixed air node. This does not match the assumption for the current specification (and the QAS), which specifies (calculates) the outdoor air as the mass needed to replace 200 cfm of room exhaust air calculated at the room air density.

In FC, SZ and CV heating cases (AE101, AE201, AE301), the mixed air is colder (~39°F) and denser than the zone air entering the exhaust fan (70°F) so the DOE-2.2 predicted outdoor air mass flows are greater than for other programs. The greater outdoor air mass flow leads to increased heating coil (or pre-heating coil) loads. One interesting, remaining uncertainty is in the VAV heating case (AE401), where the outdoor air is much colder than 39°F due to the lack of return air, yet the outdoor air mass flow rate is the same as in Case AE301 (see Figure 3C-15). It may be that DOE-2.2 is calculating the air density not at the mixed air node but after the preheat coil.

In the cooling cases, the mixed air is at nearly the same temperature as the air entering the exhaust fan, so the DOE-2.2 predicted outdoor air mass flow rates are similar to other programs. The result is that cooling loads are similar to the other programs for the FC, SZ, and VAV systems. In the CV system cooling test cases, the variation in DOE-2.2 cooling energy results versus other program results is the result of the supply mass flow variation versus other program results, and not from differences in outdoor air mass flow.

Figure 3C-16 presents the delta coil loads between the 200 series (SZ) test cases and the 100 series (FC) test cases. The difference should reflect the heat added by the return fan which is the sole difference in the test cases. DOE-2.2 shows a larger difference than the other programs, of which, some or all is attributable to different fan placement used in the DOE-2.2 models. The test specification calls for a draw-through fan located after the heating and cooling coils in all test cases. The DOE-2.2 Fan Coil (FC) does not allow this fan configuration, so the fan is being modeled before the coils while the Packaged Single Zone (PSZ) model follows the test specification with the fan after the coils. With the fan before the coils the cooling coil can cool the supply air just to the required temperature. If the fan were modeled as specified, with the fan after the coil, the supply air would need to be cooled below the required supply air temperature so that when the fan heat is added, the supply air temperature is adequate. This lower supply temperature condenses more moisture from the supply air leading to higher latent loads. Figure 3C-9 shows this effect with Case AE104 (DOE-2.2 FC system) having a relatively lower latent load than Case AE204 (DOE-2.2 PSZ system).

## **7. Modeling Notes for ASHRAE Standard 140 (S140outNotes.txt)**

See Appendix B of this modeler report.

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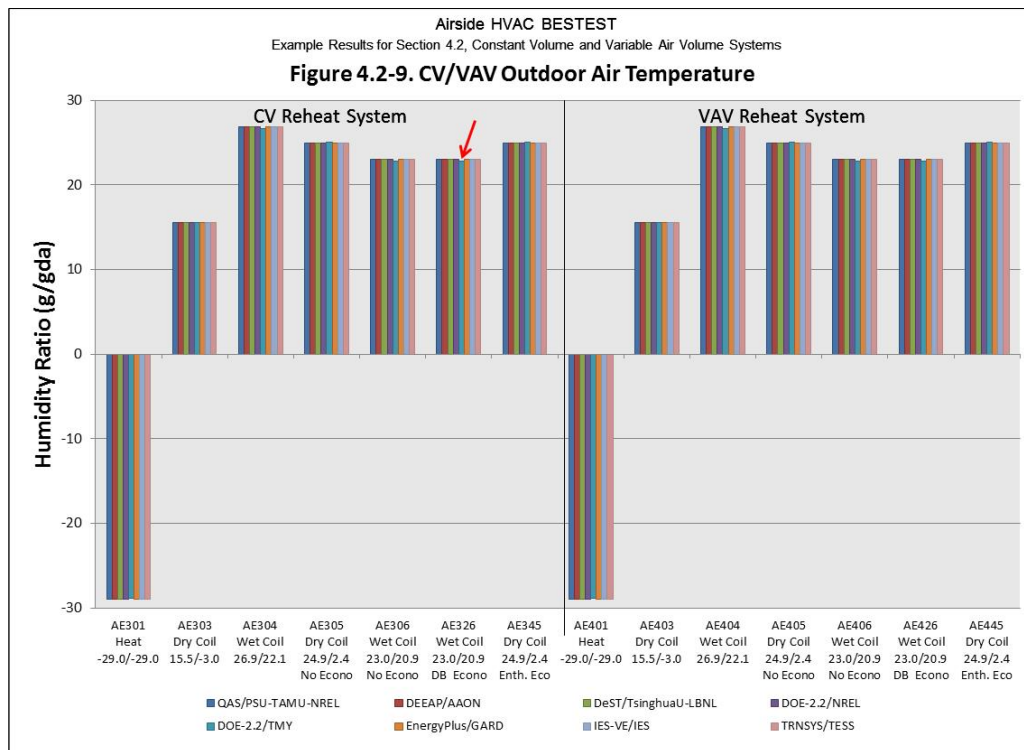
<sup>1</sup> Quasi-analytical solution (QAS).

## 8. Conclusions and Recommendations

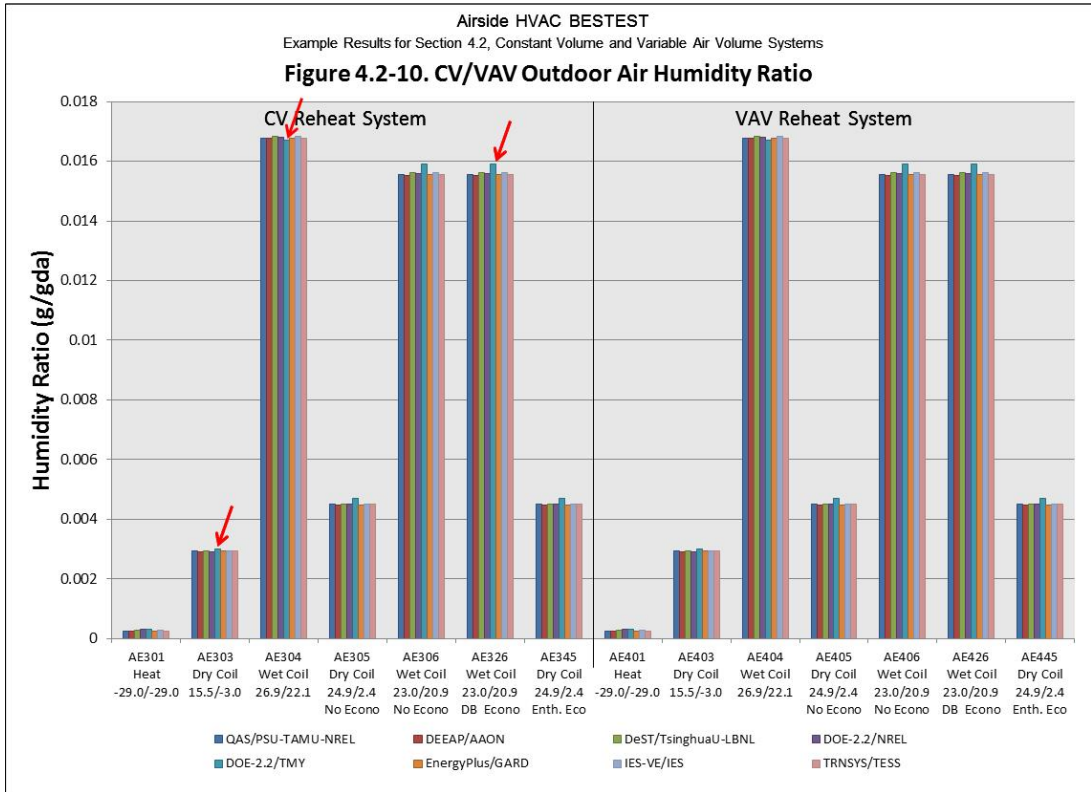
The test cases look at a single fan placement (draw-through) and a single method of specifying outdoor air (replacement of zone exhaust mass flow). Many simulation programs make simplifying assumptions related to mass flow that will align better or worse with this fan placement and outdoor air specification. For DOE-2.2, the use of the hourly mixed air density to calculate mass flows throughout the system better represents conditions in systems with a blow-through fan (fan before coils, common in fan coil and single zone) and where outdoor air is specified as a percent of outdoor air (common in smaller buildings). Use of the mixed air density performs less well with the current test case configurations – i.e., with draw-through fan, and with outdoor air specified as replacing a fixed volumetric flow of zone air exhaust. Programs that calculate a fixed air density for all hours based upon standard conditions perform well in the current test case system configurations and outdoor air specification, but would not do as well if the supply fan entering air were before the coils, or if outdoor air were specified directly as a volume of outdoor air.

For this reason, it is important that test cases be developed to explore differing fan placement and outdoor air specifications to test program abilities over a range of configurations rather than a single fan and outdoor air specification, which consistently favors one class of mass flow simplifying assumptions.

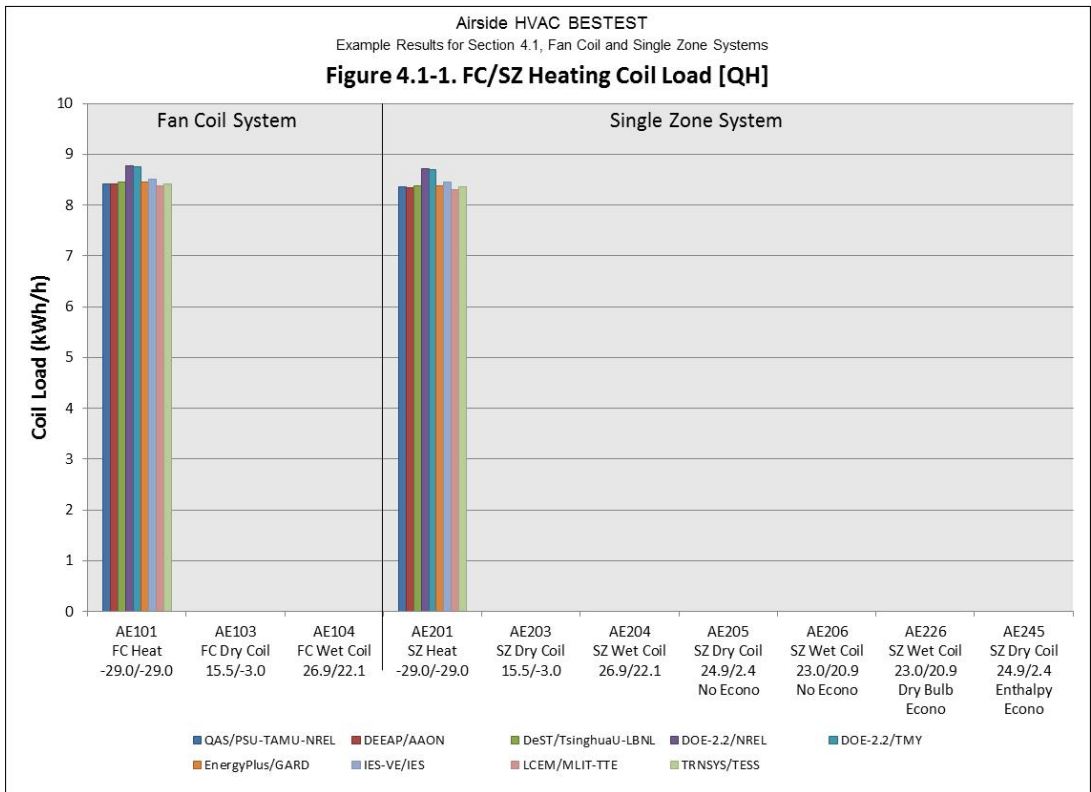
DOE-2.3 (not tested because it was in early development when we began initial simulation work) handles mass flow differently and utilizes a constant air density similar to other programs in this trial. This may address many of the differences between DOE-2.2 and other programs applying constant system air density in these test cases. Though like such other programs, the results may not agree as well with a quasi-analytical solution (or with programs allowing variation of air density within the system) in a test case with a blow-through fan or in a test case with outdoor airflow rate defined directly (rather than as replacing exhaust airflow).



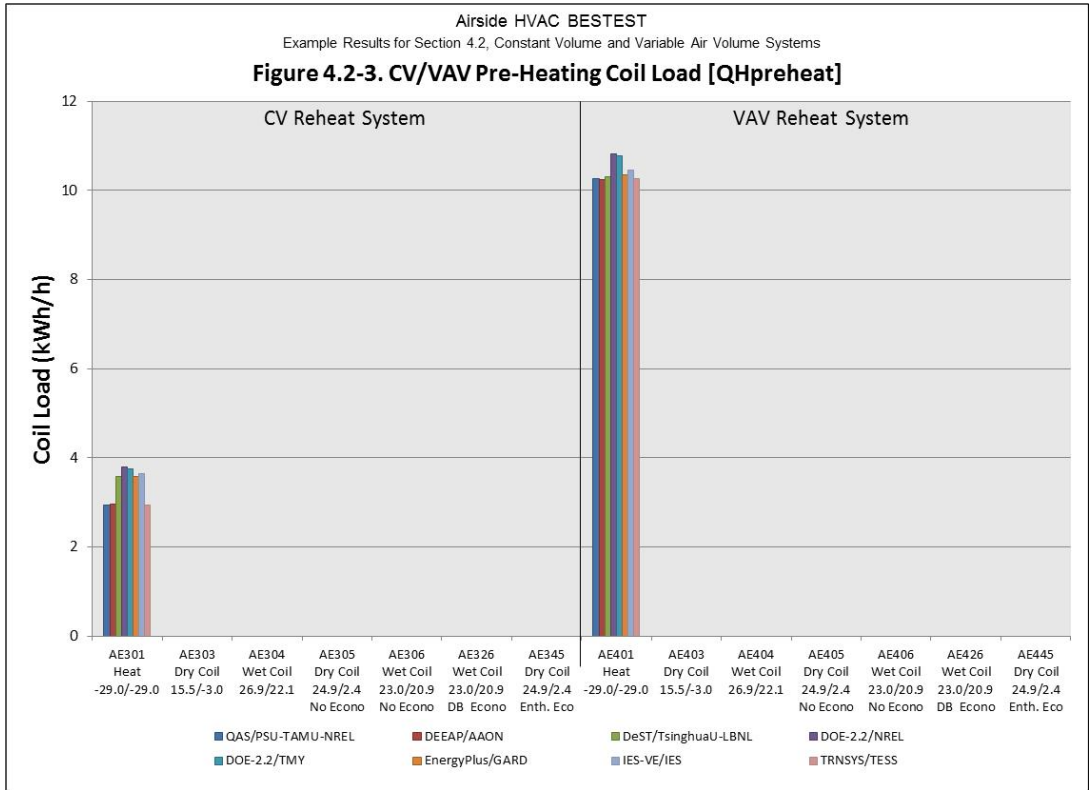
**Figure 3C-3. Outdoor Air Temperature**



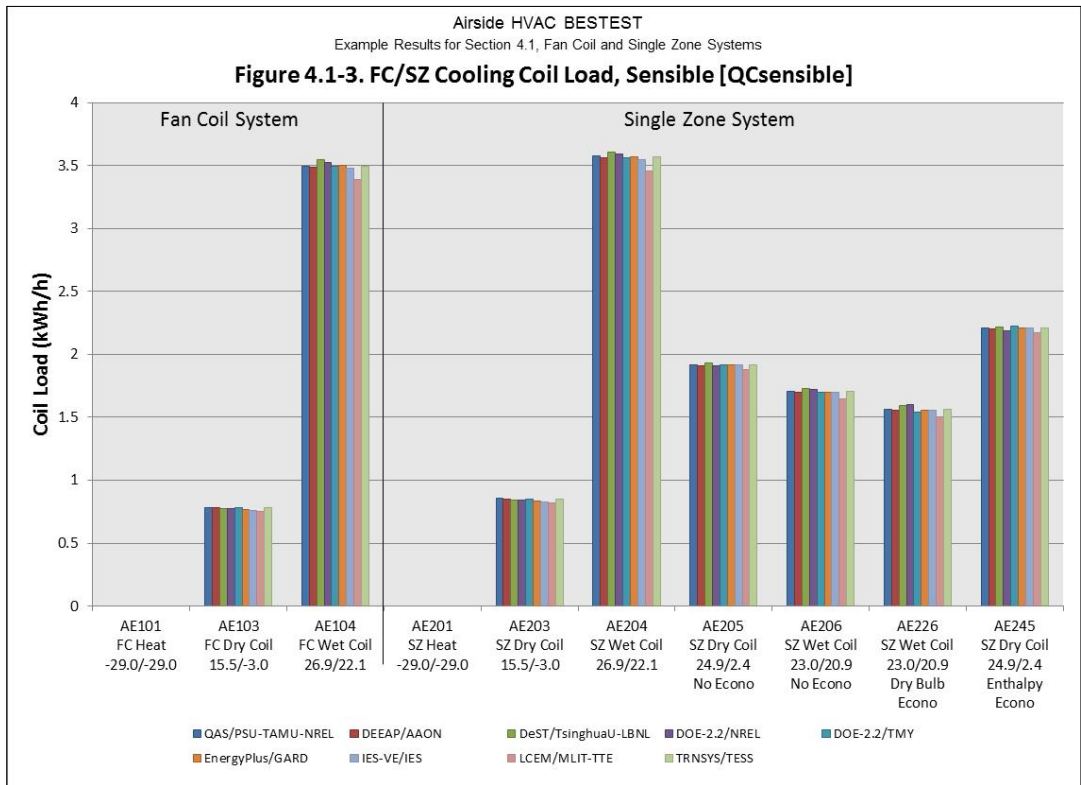
**Figure 3C-4. Outdoor Air Humidity Ratio**



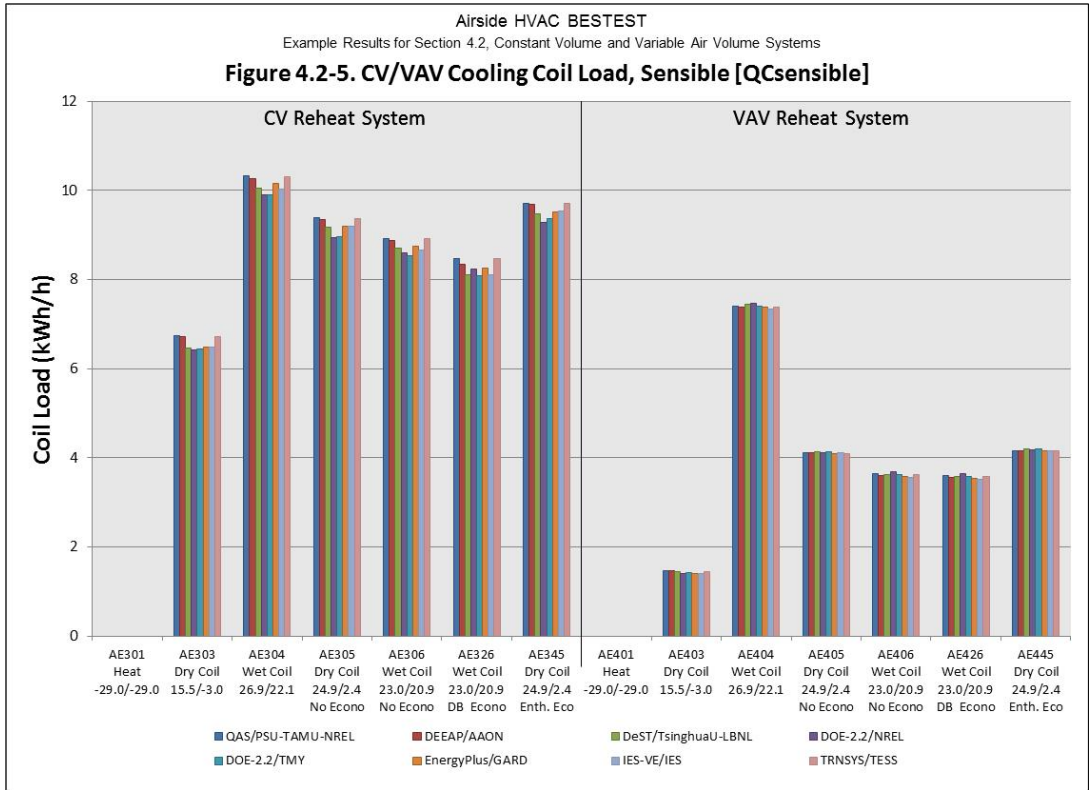
**Figure 3C-5. FC and SZ Test Case Heat Coil Load**



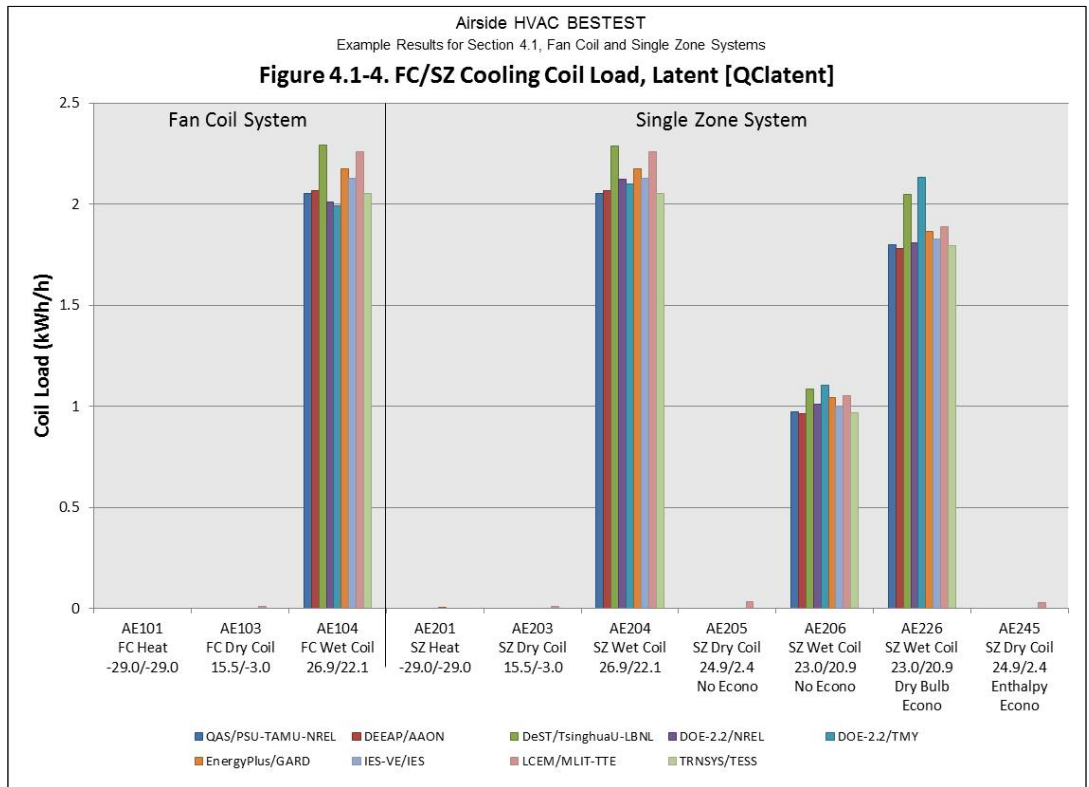
**Figure 3C-6. CV and VAV Test Case Heat Coil Load**



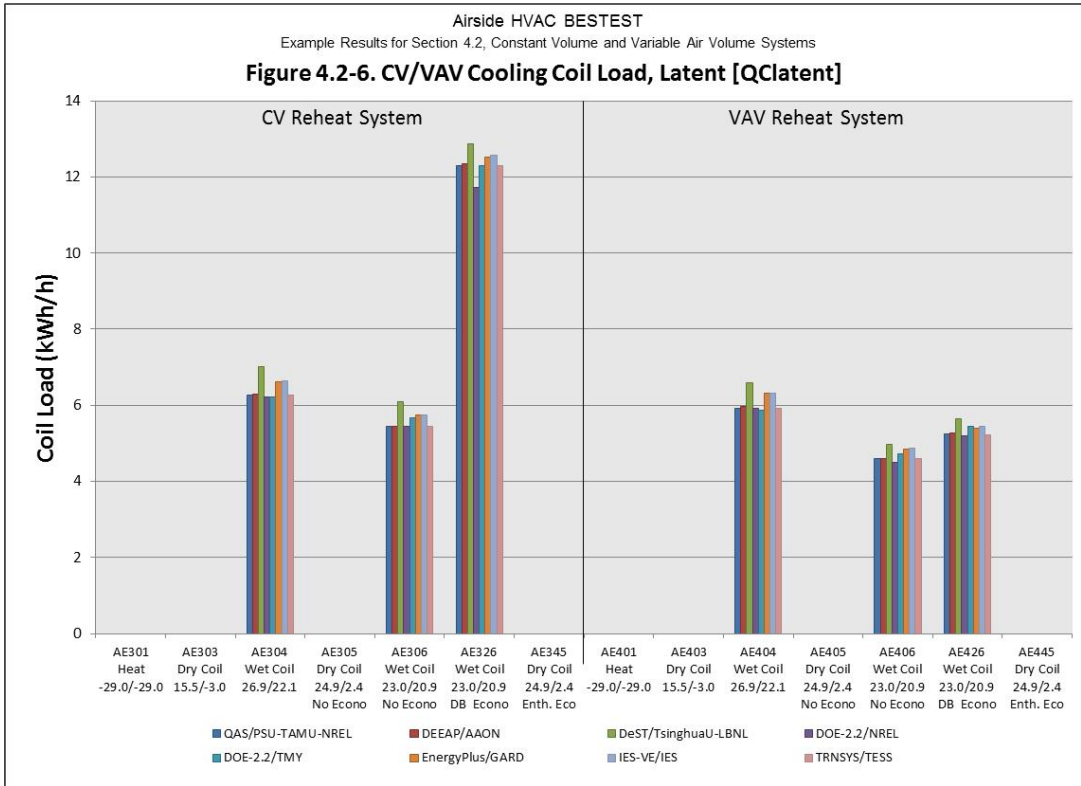
**Figure 3C-7. FC and SZ Test Case Cooling Coil Sensible Load**



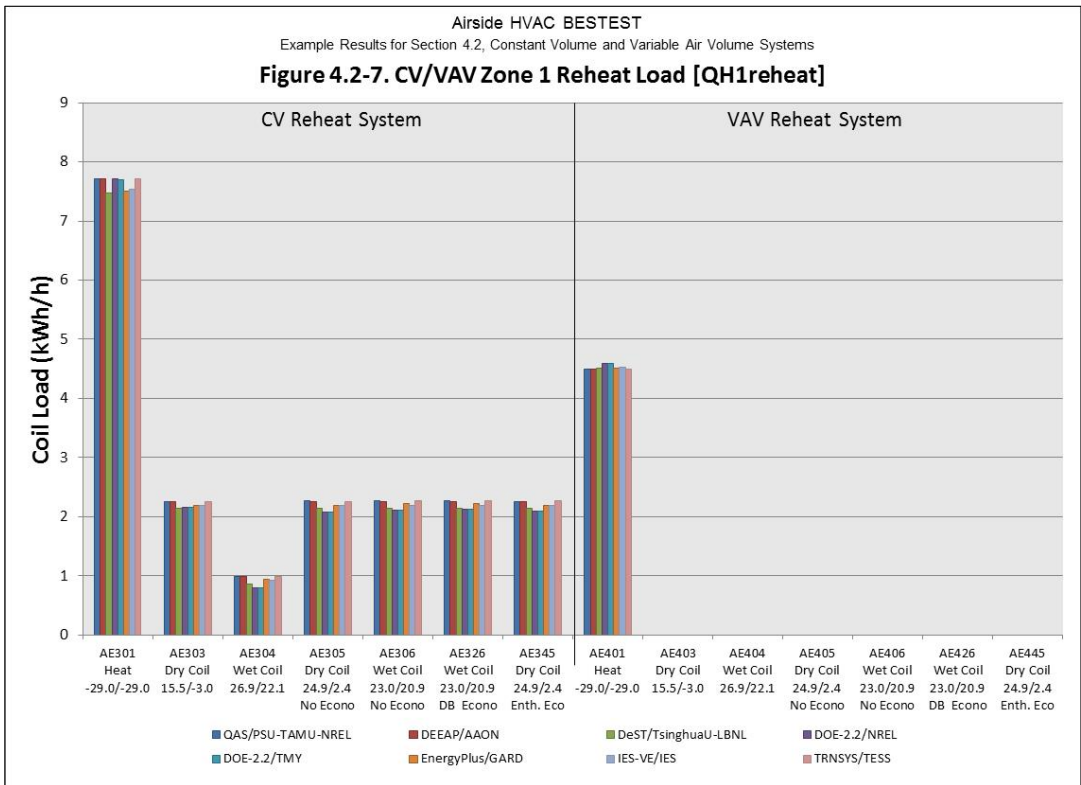
**Figure 3C-8. CV and VAV Test Case Cooling Coil Sensible Load**



**Figure 3C-9. FC and SZ Test Case Cooling Coil Latent Load**



**Figure 3C-10. CV and VAV Test Case Cooling Coil Latent Load**



**Figure 3C-11. CV and VAV Test Case Zone 1 Reheat Coil Load**

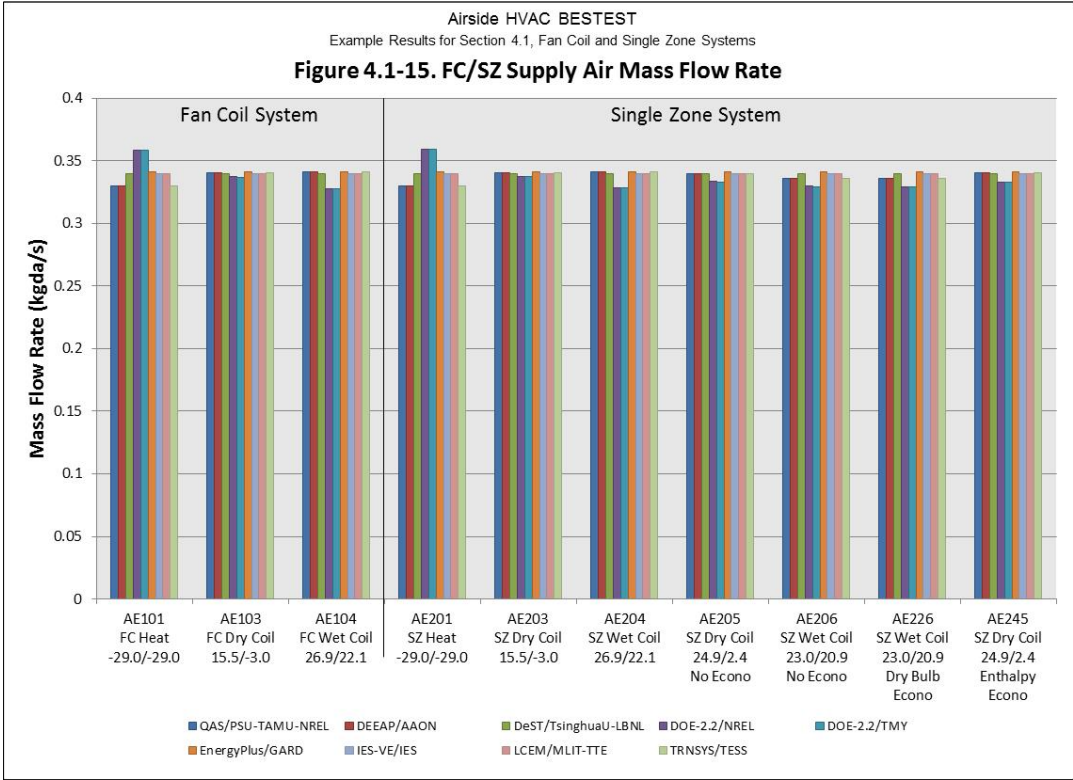


Figure 3C-12. FC and SZ Test Case Supply Air Mass Flow Rate

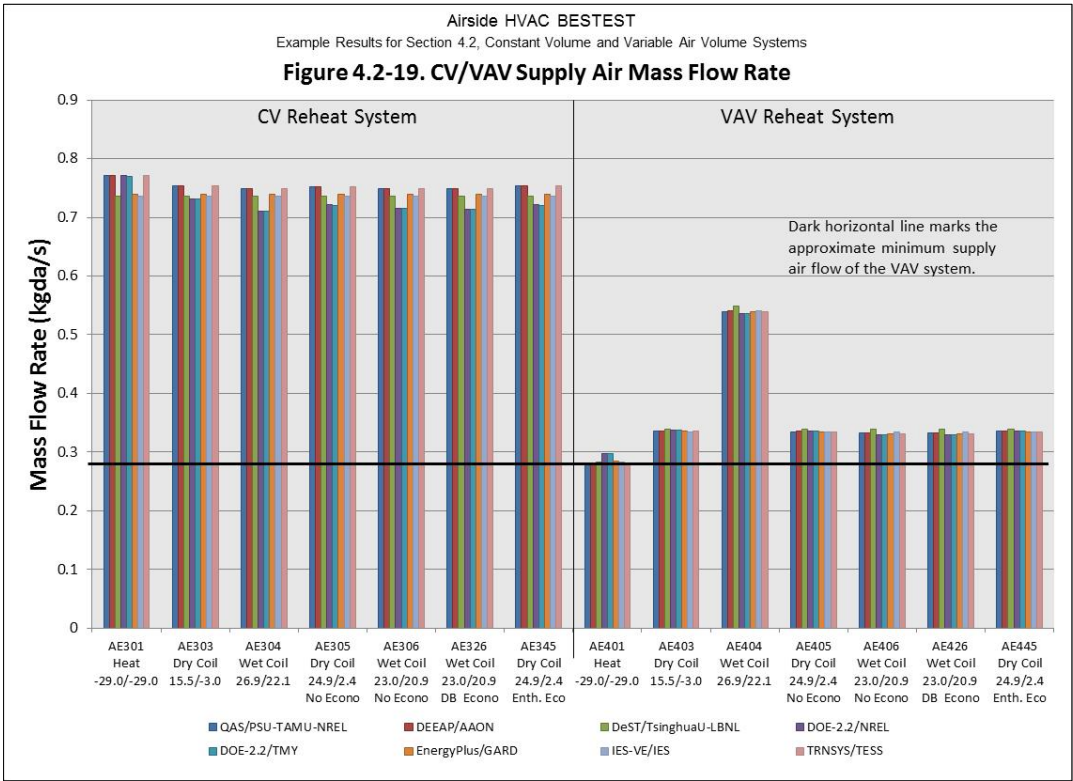
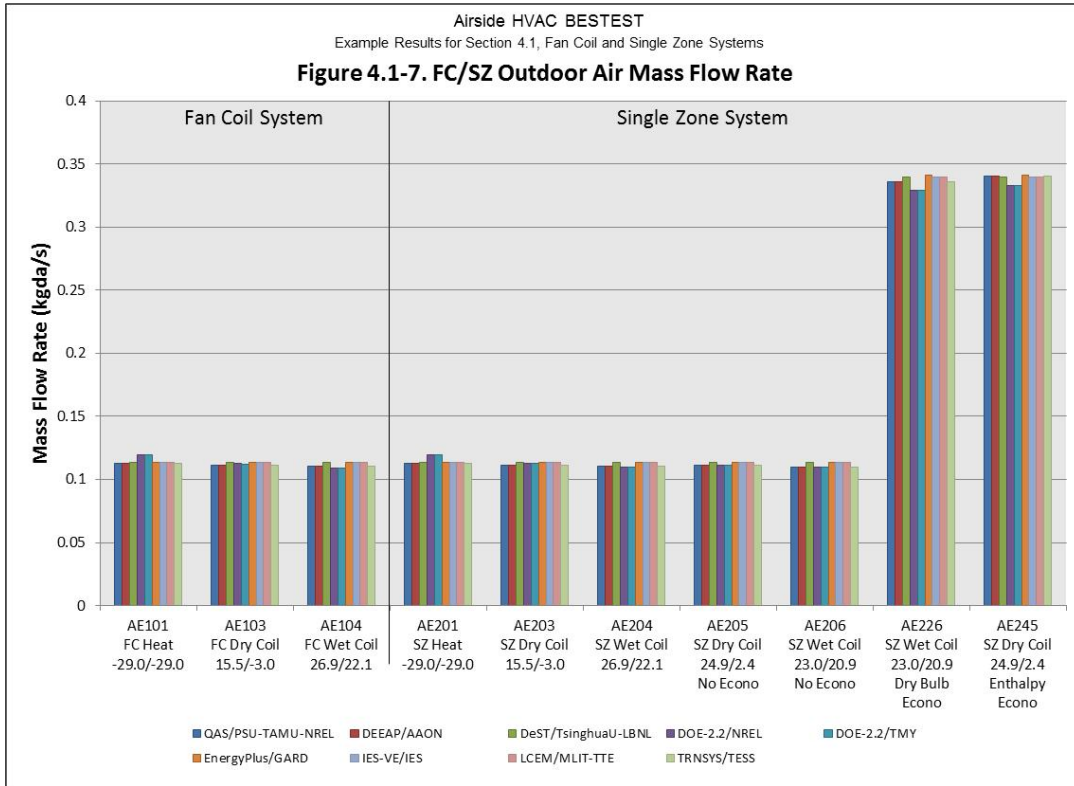
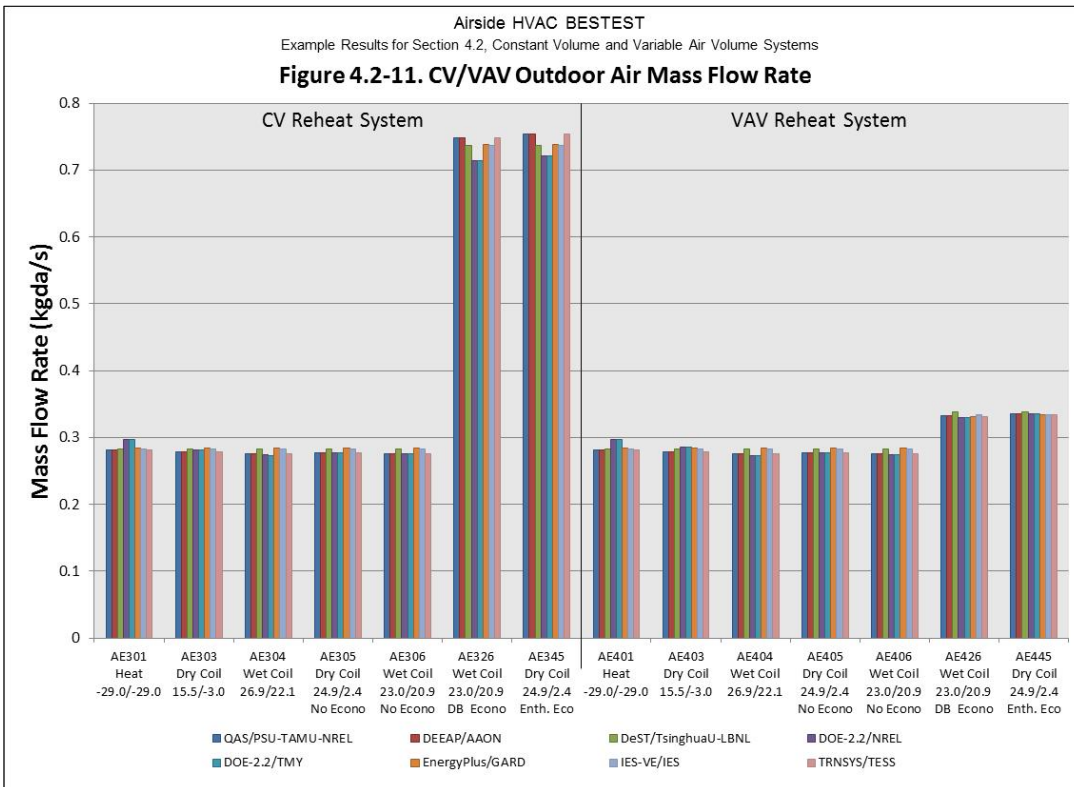


Figure 3C-13. CV and VAV Test Case Supply Air Mass Flow Rate



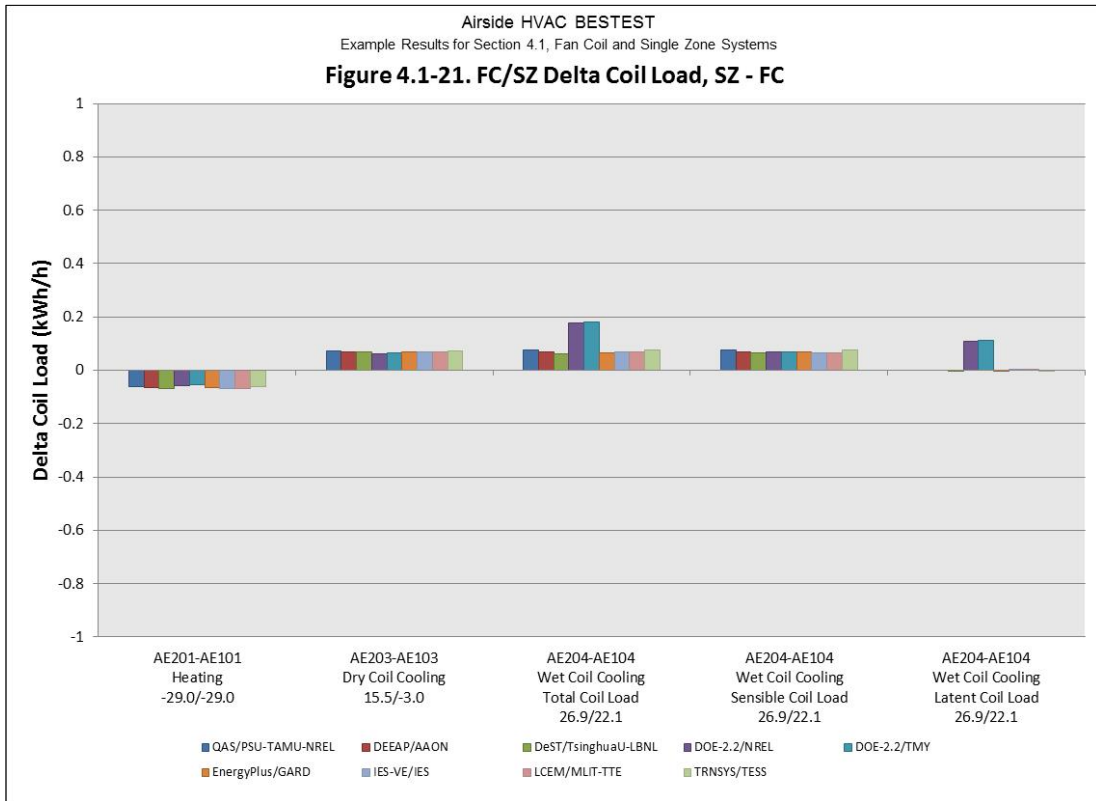


**Figure 3C-14. FC and SZ Test Case Outdoor Air Mass Flow Rate**



**Figure 3C-15. CV and VAV Test Case Outdoor Air Mass Flow Rate**





**Figure 3C-16. FC to SZ Test Case Delta Coil Load**

## 9. References

*DOE-2.2 Building Energy Use and Cost Analysis Program Volume 2: Dictionary.* Lawrence Berkeley National Laboratory / James J. Hirsch & Associates.

*DOE-2.2 Building Energy Use and Cost Analysis Program Volume 4: Libraries & Reports.* Lawrence Berkeley National Laboratory / James J. Hirsch & Associates.

*Adaptation of ASHRAE RP 865 Airside HVAC Equipment Modeling Test Cases for Inclusion in ASHRAE Standard 140, Final Draft Test Specification, Sep 2014.*

Hirsch, J.J. (2013). James J. Hirsch & Associates. Personal communications with J. Neymark, Joel Neymark & Associates, Golden, CO, August 2013.

Hirsch, J.J. (2014). James J. Hirsch & Associates. Personal communications with J. Neymark, Joel Neymark & Associates, Golden, CO, March 2014.

## Appendix A. Pro-Forma Questionnaire for Airside HVAC Tests, October 2014

### *Instructions:*

- *Replace empty boxes with symbols just below, as appropriate.*
  - Possible to use
  - Used to develop simulation trial results
- *Provide additional information and comments as appropriate; add lines as needed.*
- *If specific responses require disclosure of proprietary data not normally available to users in the program's engineering manual, leave the response blank, and include a note to the effect that including a response would require disclosure of proprietary information.*
- *If modeling varies among systems (and/or test cases), specify which systems (and/or test cases) are associated with each response; recopy a query as needed for this purpose (e.g., create 14a, 14b, etc.).*

**Program name, including version number**

DOE 2.2 V48L

---

**Your name and organization**

Mike Kennedy, Mike D. Kennedy Inc.

---

**1. Program status**

- Public domain
- Commercial
- Other (please specify): \_\_\_\_\_

**2. What weather inputs are used by the program for these tests?**

- Provided TMY2 weather data
- Directly entered DB and DP
- Directly entered DB and RH
- Directly entered DB and W
- Directly entered DB and WB

Note: DB = outdoor dry-bulb temperature; DP = outdoor dew-point temperature; RH = outdoor relative humidity; W = outdoor humidity ratio; WB = outdoor wet-bulb temperature.

**2a. If weather inputs provided from TMY2 data, which data are used for calculating outdoor air humidity ratio?**

- DB and DP
- DB and RH
- Other (please specify): \_\_\_\_\_

**3. How frequently is the HVAC system air density calculated for each system?**

- Once per run, same value applied throughout system
- Once every time step, same value applied throughout system
- Once every time step, varies by location throughout system
- Other (please specify): \_\_\_\_\_

**4. What air property basis is applied for HVAC system air density calculations for each system?**

- Single fixed air condition (please specify): \_\_\_\_\_
- Multiple fixed air conditions depending upon operating mode or system type (please specify): \_\_\_\_\_
- Air condition at each time step at a single specific system location \_\_\_\_\_ (please specify location): [mixed air node](#)
- Local air conditions at each time step at each system location
- Other (please specify): \_\_\_\_\_

**5. What air pressure is used in HVAC system air density calculation?**

- Altitude adjusted standard pressure
- Weather file barometric pressure
- Other (please specify): \_\_\_\_\_

**6. Are fan/duct/system pressure differences accounted for in the HVAC air density calculations?**

- Yes
- No

-

**7. Are system models used for test cases the same as those commonly used by modelers to model similar systems?**

- Fan Coil:       Yes  No  
Single Zone:     Yes  No  
CV Reheat:      Yes  No  
VAV:             Yes  No

**8. How are zone loads defined?**

- Zone loads specified directly from input tables (e.g., Part I, Table 1-2)  
 Using alternative zone definition (e.g., Part I, Section 1.5.5.1.1.5)

**9. Thermostat Control**

- Perfect control  
 On/Off thermostatic control  
 On/Off thermostatic control with deadband  
 Proportional control  
 Other (please specify): [on/off thermostat but program implementation of on/off thermostat still involves a proportional throttling range of 0.5°F.](#)

**10. Were adjustments to zone temperature set points required to attain test case specified zone temperature?**

- Yes, consistent adjustment for all cases  
 Yes, different adjustment for each case  
 No  
 Other, or additional discussion (please specify): \_\_\_\_\_

**11. Were adjustments to CV and VAV system supply air temperature set points required to attain test case specified system supply air temperature?**

- Yes, consistent adjustment for all cases  
 Yes, different adjustment for each case  
 No  
 Other, or additional discussion (please specify): [system supply air temperature throttling range \(COOL-CTRL-RANGE\) was set to 0.01°F.](#)

**12. How was the zero bypass factor idealization modeled in FC/SZ test cases?**

- Directly, using the program's standard input scheme  
 Using custom supply air delivery temperature inputs for each test case [For cases AE104, AE204, AE206, AE226 the program supply air temperature was set for each test case to be very close to the temperature required to just meet cooling load.](#)  
 Other, or additional discussion (*describe*): \_\_\_\_\_  
 Was not successful

**13. How was the zero bypass factor idealization modeled in CV/VAV test cases?**

- Directly, using the program's standard input scheme  
 Using custom supply air delivery temperature inputs for each test case  
 Other, or additional discussion (*describe*): \_\_\_\_\_  
 Was not successful

**14. Which Section 1.6.5.n.3 specified detailed output variables were reported, and at which specified locations?**

Temperatures:  All locations reported  Some locations reported  Not reported

Describe specified output locations not reported:\_\_\_

Zone supply air temperature is not reported for Cases AE201-AE445

---

Return fan inlet temperature is not reported for Cases AE301-AE445

---

Mixed air temperature is not reported for Cases AE301 and AE401

---

Humidity Ratios:  All locations reported  Some locations reported  Not reported

Describe specified output locations not reported:\_\_\_\_\_

Specific Volumes:  All locations reported  Some locations reported  Not reported

Describe specified output locations not reported:\_\_\_Specific volume is not reported for any locations by the program and hand calculations from other outputs were not done. \_\_\_

Enthalpies:  All locations reported  Some locations reported  Not reported

Describe specified output locations not reported:\_\_\_Enthalpies in SI units are not reported for any locations by the program and hand calculations from other outputs were not done.

---

Mass Flows:  All locations reported  Some locations reported  Not reported

Describe specified output locations not reported:\_\_\_Mass flows are reported by the program for all system locations but not for the zones.

---

**15. For reported detailed variables, how were outputs generated?**

Temperatures:  Direct from model  Required unit conversion  Other post processing\*

Humidity Ratios:  Direct from model  Required unit conversion  Other post processing\*

Specific Volumes:  Direct from model  Required unit conversion  Other post processing\*

Enthalpies:  Direct from model  Required unit conversion  Other post processing\*

Mass Flows:  Direct from model  Required unit conversion  Other post processing\*

\* “Other post processing” includes calculations outside of the model, and/or output that is inferred as equivalent or approximately equivalent from another location, e.g., for the SZ system return air fan inlet conditions and zone air conditions might be assumed to be equal or approximately so.

Describe any post-processing (outside-the-model) calculations and/or assumptions used to produce detailed outputs:\_\_\_The program temperature outputs were in degrees Fahrenheit and converted to degrees Celsius.

Mass flow rates are not directly available from the program. The program does provide the mixed air density for each hourly time step. From looking at the provided DOE2.2 source code, this mixed air density is calculated each hour. The system keeps a volumetric balance and uses the density from this single point (mixed air) in combination with the volumetric flow for energy calculations at all system locations. The supply air mass flow rate was therefore assumed to be (and calculated as) the product of the mixed air density and the supply CFM. The mixed air, and the heating and cooling coil leaving air, and supply air volumetric and mass flows are assumed to be equal to the supply air flows.

The return air mass flow was calculated as the return CFM times the mixed air density.

The program calculates mixed air conditions as the CFM weighted average of the return and outdoor conditions with the outdoor CFM being equal to the exhaust CFM. The implicit outdoor air mass flow rate is therefore the product of the outdoor air CFM ratio, the supply CFM, and the mixed air density.

These calculations were not verified with the program authors so should be considered tentative. The mass flows were also converted assuming 1 lb equals 0.453592 kg.

Additional output note. Output or specific outputs were not directly available from the program for many system locations (e.g., return fan inlet). However, a majority of unavailable values have been filled in from adjacent system locations which were deemed comparable. Mass flow of the mixed air (kg/s dry air) was assumed to be the same as system supply air. The humidity ratio is assumed to be the same before and after the return fan. There is some potential for error in this process but the example results were carefully reviewed.

---

## Appendix B. Modeling Notes for ASHRAE Standard 140

[Editor's note: Content here summarized from S140outNotes.txt form submitted during the simulation trials.]

### A. Software Information

1. Vendor: [James J. Hirsch & Associates](#)
2. Software Name: [JJ Hirsch DOE-2.2, Version 48l](#)
3. Version (unique software version identifier): [2.2, Version 48l](#)
4. Operating System Requirements: [Windows XP](#)
5. Approximate Hard Disk Space Required for Installation: [not available](#)
6. Minimum RAM Required for Software Operation: [not available](#)
7. Minimum Display Monitor Requirements: [None](#)
8. Other Hardware or Software-Related Requirements: [None](#)

### B. Alternative Modeling Methods

#### NOTE 1 – [System Selection](#)

1.1 Simulated Effect: [System Selection](#). DOE-2.2 supplies more than 20 different system models, many of which are flexible enough to model more than one of the specified test case systems.

#### 1.2 Optional Settings or Modeling Capabilities

The optional models that are capable of modeling the specified test case systems are:

Test Case System	Available models	Selected Model
FC	FC, PSZ, PVVT, RESYS2, RESVVT	FC
SZ	PSZ, PVVT, RESYS2, RESVVT	PSZ
CV	RHFS, VAVS, PVAV	RHFS
VAV	VAVS, PVAV	VAVS

Some of these models have substantial portions of computer code in common with each other while others are completely separate sections of computer code.

PVVT is capable of modeling single zone variable volume, variable temperature systems. It is the most up-to-date model and some, well informed, modelers use it to model all single zone systems. However, the DOE2.2 manual does not characterize PVVT as a general single zone system and the eQuest wizard assigns the more targeted models (e.g., FC and PSZ) in many situations.

### 1.3 Setting or Capability Used:

The model most similar to the test case system was selected. Untested systems that would be worthwhile inclusions in future work are PVAV and PVVT.

### NOTE 2 – Weather Input

2.1 Simulated Effect: Test case weather input.

### 2.2 Optional Settings or Modeling Capabilities

#### 2.2.1 Weather input = Design day

Physical Meaning: weather specified as design-day dry-bulb and wet-bulb temperatures. Does not appear to limit the precision of specified numbers.

#### 2.2.2 Weather input = weather file

Physical Meaning: weather data supplied using a weather file. The DOE2 weather file format limits the input precision to the nearest whole degree.

2.3 Setting or Capability Used: design-day

## C. Equivalent Modeling Methods

### NOTE 1 - Bypass Factor in FC and SZ systems

1.1 Simulated Effect: Bypass Factor = 0. This is an input in DOE-2.2 but the FC and SZ models utilize a fixed set point for the supply air temperature during the cooling portion of the hour and cycle as required to meet the zone loads.

1.2 Sections of the Test Specification where Relevant Inputs are Specified: 1.5.5.1.1.3, item d and 1.5.5.3.1.3, item f [see Part I of this document]

1.3 Equivalent Inputs Used: A trial and error process was implemented to adjust the supply air temperature in each test case (using the MIN-SUPPLY-T keyword) to achieve a bypass factor less than 0.01 (as reported by system hourly report variable number 56 – system bypass) while still maintaining the zone temperature (as reported by the zone hourly report variable number 31 – average zone temperature). The adjustments were only made to the cases with latent cooling load.

1.4 Physical, Mathematical or Logical Justification of the Equivalent Inputs: Method results in bypass factor of zero.

### NOTE 2 – Return fan heat

2.1 Simulated Effect: Return fan and motor heat and efficiency. DOE-2.2 does not have inputs for specifying return fan static pressure and motor fan efficiency, it only allows the specification of total efficiency. In the systems utilized in these runs, DOE-2.2 assumed electrical efficiency (~90%) to calculate a mechanical efficiency, as reported by the DOE-2.2 SV-A report, from the total efficiency. Initially we assumed that DOE-2.2 placed the return fan motor outside the airstream, since the supply fan was specified as such. To get the proper return fan mechanical efficiency as reported in the DOE-2.2 SV-A report, the return fan total efficiency was adjusted (~63%) yielding a return fan mechanical efficiency reported on the SV-A report of 70%. In Round 3b, it was determined DOE-2.2 assumes that when the return fan is specified in this way, the return motor is assumed to be in the air stream. So the



return fan total efficiency was set to 70% and SV-A now reports a mechanical efficiency of 78% but total efficiency is 70%.

2.2 Sections of the Test Specification where Relevant Inputs are Specified: 1.5.5.2.1.1.2, 1.5.5.3.1.1.2, 1.5.5.4.1.1.2 [see Part I of this document]

2.3 Equivalent Inputs Used: DOE-2.2 assumes electrical efficiency (~90%) to calculate a mechanical efficiency, as reported by the DOE-2.2 SV-A report, from the total efficiency. However, DOE-2.2 also assumes that the return motor is in the air stream in these system types so the return fan total efficiency was set to 70% and SV-A now reports a mechanical efficiency of 78%, but total efficiency is 70% and the proper amount of heat is imparted to the air.

2.4 Physical, Mathematical or Logical Justification of Equivalent Input: Correct fan heat is crucial to the test cases and fan energy is not a required output.

NOTE 3 – CV system sizing issue in economizer runs

3.1 Simulated Effect: There was difficulty in maintaining system supply air temperature precisely at 55°F when running a dry bulb economizer with wet ambient air. This issue is a coil capacity issue rather than a control issue, and arises because design-day runs were used (see Report Block B, note 2). At the conditions of the AE326 test case (ODB = 23°C, OWB = 21.525°C), the dry bulb economizer should save a small amount of sensible cooling but increases latent cooling coil load substantially.

As explained by the program author (Hirsch 2013): "... the sizing calculations do not take economizer operation into account when calculating the coil capacity; the capacity is calculated using the design OA quantity. The design sizing calculations assumed an outside air ratio of 0.385 as reported in SV-A. Given this outside air ratio, the run without an economizer held 55°F supply air temperature. In the run with an economizer, the drybulb was sufficient to open the economizer 100%, but the outdoor wetbulb was forced to be high (70°F), so that the mixed air enthalpy was higher than with the economizer at minimum position. This overloaded the cooling coil, and the supply air temperature rose to 56.3°F. Because the sizing calculations assume the economizer is at minimum position in both cases, the coil size is the same in both cases. If the SYSTEM command keyword MIN-OUTSIDE-AIR = 1.0 is specified, then the sizing calculations will upsize the cooling coil for the 100% outside air fraction, and the supply air temperature is maintained at 55°F in the economizer run case. However, most designers would likely not utilize a 100% OA sizing condition in humid climates."

3.2 Section of the Test Specification where Relevant Inputs are Specified: 1.5.5.3.1.3 [see Part I of this document]

3.3 Equivalent Input Used: To generate coil loads reflecting 100% outdoor air, the minimum outside air fraction was set to 100% for the economizer runs. This solution would not work in a simulation that included non-economizer hours with cooling, but the capacity issue would not arise in a typical annual simulation either.

3.4 Physical, Mathematical or Logical Justification of Equivalent Input: Maintain system supply air at 55°F.

#### **D. Non-Specified Inputs**

None

#### **E. Omitted Test Cases and Results**

NOTE 1 – Detailed outputs described in Appendix A (pro-forma portion of this modeler report), Item 14

1.1 Cases where Results Were Omitted, and which Results Were Omitted: See 1.2

1.2 Explanation for Omitting Test Case Results:

These outputs are not provided by the program:

- Zone supply air temperature for Cases AE201-AE445
- Return fan inlet temperature for Cases AE301-AE445
- Mixed air temperature for Cases AE301 and AE401
- Specific volume for any locations. Hand calculations from other outputs would be possible but were not done.
- Enthalpies in SI units for any locations. Hand calculations from other outputs would be possible but were not done.

#### **F. Changes to Source Code for the Purpose of Running the Tests, Where Such Changes are Not Available in Publicly Released Versions of the Software**

None

#### **G. Anomalous Results**

None

## **Appendix III-D**

### **Modeler Report for HVAC Air Distribution System Tests EnergyPlus Version 8.2.0**

by

**R. Henninger and M. Witte  
GARD Analytics, Inc., USA**

**March 2015**

#### **1. Introduction**

**Software:** EnergyPlus Version 8.2.0. e53a815bb2  
**Authoring Organization:** U.S. Department of Energy  
Energy Efficiency and Renewable Energy  
Office of Building Technologies  
**Authoring Country:** USA

This report describes the modeling methodology and results for follow-up testing done subsequent to Simulation Trial Round 2a results submitted in January 2013 for the Adaptation of ASHRAE RP 865 Airside HVAC Equipment Modeling Test Cases using the EnergyPlus software. The specifications for the test suite are described in *DRAFT Adaptation of ASHRAE RP 865 Airside HVAC Equipment Modeling Test Cases for Inclusion in ASHRAE Standard 140, December 2012*.

#### **2. Modeling Assumptions**

The following comments are provided in regards to user inputs that were used with EnergyPlus to model each of the cases described in the test suite specification. Except where discussed below, all other requirements of the specification were met.

- a) The requirement for an adiabatic building shell was modeled for all surfaces.
- b) All exterior surfaces were locked out from seeing any solar radiation by specifying the NoSun option for each surface.
- c) Number of timesteps per hour was set to 6.
- d) The following EnergyPlus HVACTemplate objects were used to setup the required air distribution systems. The expanded idf (expidf) files were then customized to meet the test specifications.
  - For the single zone constant volume Fan Coil Cases AET101a to AET104a and constant volume Single Zone Cases AET201a to AET206a, the EnergyPlus HVACTemplate:Zone:VAV and HVACTemplate:System:VAV objects were used with changes to force operation as a constant volume system with the heating coil placed before the cooling coil in the main air flow stream. *[Editor's note: This modeler report references preliminary test case numbering, which is similar to final numbering – e.g., Case AET101a became AE101, and similar for other cases.]*

- Alternatively, the EnergyPlus HVACTemplate:Zone:Fancoil object was used to model the four-pipe fan coil system for Cases AET101a, AET103a and AET104a with the following limitations: 1) the EnergyPlus 4-Pipe Fan Coil model only allows a blow-through configuration with the heating coil placed after the cooling coil, and 2) the outdoor air and exhaust air interchange takes place automatically within the fan coil unit without the aid of an exhaust fan and therefore the fan coil return air flow rate is always equal to the supply air flow rate.
  - For the Constant Volume Terminal Reheat Cases AET301 to AET345, the EnergyPlus HVACTemplate:Zone:VAV and HVACTemplate:System:VAV objects were used with zone reheat coils activated and changes to force operation as a constant volume system with the preheat coil located in the mixed air stream.
  - For the Variable Air Volume Cases AET401 to AET445, EnergyPlus HVACTemplate:Zone:VAV and HVACTemplate:System:VAV objects were used with zone reheat coils activated and allowed to operate as a fully operational variable air volume system with the preheat coil located in the mixed air stream.
- e) Since the specification did not indicate what type of coils were to be used, the following assumptions were made:
- The preheat coil and reheat coils were modeled as electric coils.
  - The heating coil in the main air stream was modeled as a hot water coil with hot water supplied by a natural gas fired boiler.
  - The cooling coil was modeled as a chilled water coil with a constant entering chilled water temperature of 44°F provided by an electric chiller.
- f) Since the capacities of the preheat, heating, reheat and cooling coils were not specified, they were allowed to be autosized by the program.

Additional information is included in the pro-forma questionnaire responses; see Appendix A of this supplementary modeler report.

### 3. Modeling Difficulties

The only difficulties encountered were those which arose when attempting to use EnergyPlus HVAC templates to model the air distribution configurations that were different than those modeled in EnergyPlus. These included:

- a) Moving the EnergyPlus placement of the preheat coil from the outdoor air stream to the mixed air stream.
- b) Moving the EnergyPlus placement of the heating coil in the main air stream from after the cooling coil to before the heating coil for cases AET100a to AET104a and cases AET201a to AET206a.
- c) Unable to model the four-pipe fan coil configuration for Cases AET101a, AET103a and AET104a as specified as a draw-through system with the cooling coil after the heating coil using

the EnergyPlus HVACTemplate:Zone:Fancoil object. Also unable to separately model the zone exhaust air fan using EnergyPlus fan coil model.

It should also be noted that EnergyPlus HVAC airside models, except for the VAV system, assume constant supply air mass air flow among the test cases for a given system (applying altitude adjusted standard air density; see Appendix A pro-forma responses, Item 4). Whereas for a given system (except VAV), the ASHRAE HVAC Airside Modeling specification indicates constant volume air flow at the supply fan, which allows different supply air mass flow rates to be calculated among cases for a given system. It is unclear what impact this difference has on the reported results.

#### **4. Software Errors Discovered**

A problem arose when trying to use the EnergyPlus HVACTemplates to model the air distribution configuration where there was a preheat coil located in the mixed air stream and there was no main heating coil. When using the HVACTemplate:System:VAV object to model a VAV reheat system with Heating Coil Type=None, Heating Coil Design Setpoint=50°C and Preheat Coil Design Setpoint=7.22°C, the reheat coils autosized to zero capacity, because the sizing assumed that the 50°C heating coil design setpoint would be supplied to the reheat coils. This gave incorrect results (not saved). It is not clear yet if this is an HVACTemplate error or EnergyPlus sizing error. This problem was resolved by setting the heating coil setpoint to 7.22°C.

#### **5. Changes Made Subsequent to Round 2a Results**

Following the meeting of the ASHRAE SSPC 140 Airside HVAC Working Group in January 2013, an Excel xls file was distributed showing the preliminary simulation results from various programs for all test cases described in the latest specification. A comparison of the EnergyPlus results for Trial Round 2a to other program results prompted a review of the EnergyPlus models for certain cases. As a result, the following changes were made and a revised set of simulations were performed.

##### **a) Series 400 Test Cases Supply Fan Temperature Rise**

The variable volume supply fan modeled by EnergyPlus for the Series 400 test cases assumed a quadratic fan power curve, which put the EnergyPlus supply fan temperature rise much higher than other programs. The specification required a cubic fan power curve. When the EnergyPlus models for the Series 400 test cases were changed to a cubic fan power curve the EnergyPlus results for fan temperature rise were within the bounds of the other program results.

##### **b) Series 400 Test Cases Return Fan Power Curve**

The variable volume return fan modeled by EnergyPlus for the Series 400 test cases assumed a quadratic fan power curve. The specification required a cubic fan power curve. When the EnergyPlus models for the Series 400 test cases were changed to a cubic fan power curve, the EnergyPlus results for VAV economizer test case AET405, AET406, AET426 and AET445 improved such that the Delta Coil Loads for the enthalpy economizer tests (AET445 – AET405) and dry-bulb economizer tests (AET426 – AET406) improved to come more in line with expected results. But the Delta Latent Coil Loads for the dry-bulb economizer tests are still somewhat lower than other program results, which it is believed to be due to the EnergyPlus assumption of altitude-adjusted standard air density during simulations.

c) Case 401 Supply Air Temperature

The EnergyPlus supply air temperature for Case 401 was lower than it should be. This problem was traced back to the value specified for the Supply Fan Minimum Flow Fraction in the Sizing:System object. It had been entered as 0.38 when it should have been 0.384615. When this change was made, the EnergyPlus supply air temperature for Case 401 came within range of the other program results.

d) Series 400 Test Cases Preheat Coil Outlet Temperature

The preheat coil outlet temperature reported for EnergyPlus in the spreadsheet results was incorrect because the value in the spreadsheet had been linked to the wrong location in the EnergyPlus csv output file. The value reported came from column BC of the EnergyPlus csv output file when it should have been from column AW. This change did not affect any other results as it was only a reporting issue with this single variable.

e) Series 300 Test Cases Reheat Coil Load

The EnergyPlus reheat coil loads for Zone 1 were lower than should be and reheat coil loads for Zone 2 were higher than should be. The problem was traced to the AirTerminal:SingleDuct:VAV:Reheat object where the Maximum Air Flow Rate had been set to Autosize when it should have been hard set to 0.28317 m<sup>3</sup>/s for Zone 1 and 0.33036 m<sup>3</sup>/s for Zone 2. Once these were changed and the models rerun, the zone reheat coil loads moved within range of the other program results except for Zone 1 and Zone 2 of Case 301 where results, although within range of other programs, were still somewhat low. Comparing the total heating for Case 301, i.e., Preheat Coil Load + Zone 1 Reheat Coil Load + Zone 2 Reheat Coil Load = 19.212 kWh/h, was within 1% of the Merged PSU-TAMU [now defined as the quasi-analytical solution] results of 19.041 kWh/h.

f) Series 200 Test Cases Return Fan Temperature Rise

The constant volume return fan modeled by EnergyPlus for the Series 200 test cases was specified with the wrong air flow rate. The supply fan air flow rate of 283.17 L/s<sup>2</sup> was mistakenly specified when it should have been the supply air flow rate less the zone exhaust air flow rate, i.e., 283.17 L/s<sup>2</sup> – 94.39 L/s<sup>2</sup> = 188.78 L/s<sup>2</sup>. Making this correction brought the EnergyPlus return fan heat results for the Series 200 test cases more in line with other program results.

## 6. Modeling Notes for ASHRAE Standard 140 (S140outNotes.txt)

See Appendix B of this supplementary modeler report.

## 7. Results

Updated results for each test simulated with EnergyPlus version 8.2.0 are presented in an Excel spreadsheet that was provided with the test suite and is provided separately with the results reported as for Round 3A. The alternative results for cases AET101a, AET103a and AET104a using the EnergyPlus 4-Pipe Fan Coil object are shown separately at the end of the spreadsheet beginning with Row 227 with notes to indicate the limitations of the EnergyPlus 4-Pipe Fan Coil model; see Sec5-5out-Results14Jan2015EnergyPlus8.2.0.xlsx [this data is provided in tab “Eplus” with the Results-FCSZ/CVVV.xlsm files].

## **8. Conclusions and Recommendations**

EnergyPlus results appear to compare favorably to the initial set of results from other programs that were distributed to participants in December 2012.

The following recommendations listed below, which were suggested after the initial round of testing with the May 2012 version of the specification, were implemented in the December 2012 version of the specification.

- The tables on pages 4 and 5 of the draft specification [now Part I, Appendix B], which summarizes pertinent information in a succinct manner for all of the test cases is presented in I-P units only. It would be helpful to have a similar set of tables in SI units.
- The specification is silent regarding the types of coils which should be used to model the various air distribution system configurations. If the modeler chooses to use a chilled water coil for the cooling coil, the split of the cooling load between sensible and latent cooling will change based on what entering chilled water temperature is chosen for the cooling coil. This needs to be addressed in the specification.

## **9. References**

EnergyPlus 2014. U.S. Department of Energy, Energy Efficiency and Renewable Energy, Office of Building Technologies. [www.energyplus.gov](http://www.energyplus.gov)

DRAFT Adaptation of ASHRAE RP 865 Airside HVAC Equipment Modeling Test Cases for Inclusion in ASHRAE Standard 140, Simulation Trial Round 2a, December 2012.

## Appendix A: Pro-Forma Questionnaire for Airside HVAC Tests, January 2015

### **Instructions:**

- *Replace empty boxes with symbols just below, as appropriate.*
  - Possible to use
  - Used to develop simulation trial results
- *Provide additional information and comments as appropriate; add lines as needed.*
- *If specific responses require disclosure of proprietary data not normally available to users in the program's engineering manual, leave the response blank, and include a note to the effect that including a response would require disclosure of proprietary information.*
- *If modeling varies among systems (and/or test cases), specify which systems (and/or test cases) are associated with each response; recopy a query as needed for this purpose (e.g., create 14a, 14b, etc.).*

### **Program name, including version number**

EnergyPlus 8.2.0

---

### **Your name and organization**

R. Henninger & M. J. Witte — GARD Analytics, Inc.

---



**1. Program status**

- Public domain
- Commercial
- Other (please specify): Open source

**2. What weather inputs are used by the program for these tests?**

- Provided TMY2 weather data
- Directly entered DB and DP
- Directly entered DB and RH
- Directly entered DB and W
- Directly entered DB and WB

Note: DB = outdoor dry-bulb temperature; DP = outdoor dew-point temperature; RH = outdoor relative humidity; W = outdoor humidity ratio; WB = outdoor wet-bulb temperature.

**2a. If weather inputs provided from TMY2 data, which data are used for calculating outdoor air humidity ratio?**

- DB and DP
- DB and RH
- Other (please specify): \_\_\_\_\_

**3. How frequently is the HVAC system air density calculated for each system?**

- Once per run, same value applied throughout system
- Once every time step, same value applied throughout system
- Once every time step, varies by location throughout system
- Other (please specify): \_\_\_\_\_

**4. What air property basis is applied for HVAC system air density calculations for each system?**

- Single fixed air condition (please specify): \_\_\_\_\_  
Dry air at 20°C drybulb at local barometric pressure to account for altitude using equation for "standard atmospheric" pressure on p 6.1 of the ASHRAE 1997 HOF (SI edition) to initialize the air systems being simulated.  $p=101325*(1-2.25577E-05*Z)**5.2559$   
where p=pressure in Pa and Z=altitude in m.
- Multiple fixed air conditions depending upon operating mode or system type  
(please specify): \_\_\_\_\_
- Air condition at each time step at a single specific system location \_\_\_\_\_  
(please specify location): \_\_\_\_\_
- Local air conditions at each time step at each system location
- Other (please specify): \_\_\_\_\_

**5. What air pressure is used in HVAC system air density calculation?**

- Altitude adjusted standard pressure (see above)
- Weather file barometric pressure
- Other (please specify): \_\_\_\_\_

**6. Are fan/duct/system pressure differences accounted for in the HVAC air density calculations?**

- Yes
- No

-

**7. Are system models used for test cases the same as those commonly used by modelers to model similar systems?**

- Fan Coil:         Yes    No  
Single Zone:     Yes    No  
CV Reheat:      Yes    No  
VAV:             Yes    No

**8. How are zone loads defined?**

- Zone loads specified directly from input tables (e.g., Part I, Table 1-2)  
 Using alternative zone definition (e.g., Part I, Section 1.5.5.1.1.5)

**9. Thermostat Control**

- Perfect control  
 On/Off thermostatic control  
 On/Off thermostatic control with deadband  
 Proportional control  
 Other (please specify): \_\_\_\_\_

**10. Were adjustments to zone temperature set points required to attain test case specified zone temperature?**

- Yes, consistent adjustment for all cases  
 Yes, different adjustment for each case  
 No  
 Other, or additional discussion (please specify): \_\_\_\_\_

**11. Were adjustments to CV and VAV system supply air temperature set points required to attain test case specified system supply air temperature?**

- Yes, consistent adjustment for all cases  
 Yes, different adjustment for each case  
 No  
 Other, or additional discussion (please specify): \_\_\_\_\_

**12. How was the zero bypass factor idealization modeled in FC/SZ test cases?**

- Directly, using the program's standard input scheme  
 Using custom supply air delivery temperature inputs for each test case  
 Other, or additional discussion (*describe*): For fancoil cases, in Sizing:Zone object the Zone Cooling Design Supply Air Temperature was set to 14.00C and Zone Cooling Design Supply Air Humidity Ratio was set to 0.006 kg/kg. For the wet coil case AE104 this resulted in a cooling coil leaving relative humidity of 99.6%. For single zone cases, in Sizing:System object the Zone Cooling Design Supply Air Temperature was set to 14.00C and Zone Cooling Design Supply Air Humidity Ratio was set to 0.006 kg/kg. For the wet coil cases AE204, AE206 and AE226 this resulted in cooling coil leaving relative humidities from 99.0% to 99.8%. \_\_\_\_\_  
 Was not successful

**13. How was the zero bypass factor idealization modeled in CV/VAV test cases?**

- Directly, using the program’s standard input scheme
- Using custom supply air delivery temperature inputs for each test case
- Other, or additional discussion (*describe*): In Sizing: System object the Central Cooling Design Supply Air Temperature was set to 12.78C and Central Cooling Design Supply Air Humidity Ratio was set to 0.008 kg/kg (Default). For wet coil cases AE304, AE 306, AE326, AE404, AE406 and AE426, this resulted in cooling coil leaving air relative humidities from 98.7% to 99.6%.
- Was not successful

**14. Which Section 1.6.5.n.3 specified detailed output variables were reported, and at which specified locations?**

Temperatures:  All locations reported  Some locations reported  Not reported  
*Describe specified output locations not reported:* \_\_\_\_\_

Humidity Ratios:  All locations reported  Some locations reported  Not reported  
*Describe specified output locations not reported:* \_\_\_\_\_

Specific Volumes:  All locations reported  Some locations reported  Not reported  
*Describe specified output locations not reported:* \_\_\_\_\_

Enthalpies:  All locations reported  Some locations reported  Not reported  
*Describe specified output locations not reported:* \_\_\_\_\_

Mass Flows:  All locations reported  Some locations reported  Not reported  
*Describe specified output locations not reported:* \_\_\_\_\_

**15. For reported detailed variables, how were outputs generated?**

- Temperatures:  Direct from model  Required unit conversion  Other post processing\*
- Humidity Ratios:  Direct from model  Required unit conversion  Other post processing\*
- Specific Volumes:  Direct from model  Required unit conversion  Other post processing\*
- Enthalpies:  Direct from model  Required unit conversion  Other post processing\*
- Mass Flows:  Direct from model  Required unit conversion  Other post processing\*

\* “Other post processing” includes calculations outside of the model, and/or output that is inferred as equivalent or approximately equivalent from another location, e.g., for the SZ system return air fan inlet conditions and zone air conditions might be assumed to be equal or approximately so.

Describe any post-processing (outside-the-model) calculations and/or assumptions used to produce detailed outputs: \_\_\_\_\_

## Appendix B: Modeling Notes for ASHRAE Standard 140

*[Editor's note: Content here is summarized from S140outNotes.txt form submitted during the simulation trials.]*

### A. Software Information

1. Vendor: U.S. Department of Energy
2. Software Name: EnergyPlus
3. Version (unique software version identifier): 8.2.0.e53a815bb2
4. Operating System Requirements:
  - Windows 7 & 8: 64-bit version (32-bit version also available)
  - Linux (Ubuntu 14.04 and compatible) 64 bit versions  
(32-bit and RHEL-compatible versions may be available via Helpdesk)
  - Mac OSX 10.9 64 bit versions

### B. Alternative Modeling Methods

#### NOTE 1 - Inside convection algorithm

##### 1.1 Simulated Effect:

Convective heat transfer exchange related to interior surfaces

##### 1.2 Optional Settings or Modeling Capabilities

- 1.2.1 SurfaceConvectionAlgorithm:Inside = Simple  
Applies constant heat transfer coefficients depending on surface orientation
- 1.2.2 SurfaceConvectionAlgorithm:Inside = TARP  
Uses variable natural convection based on temperature difference
- 1.2.3 SurfaceConvectionAlgorithm:Inside = Ceiling Diffuser  
A mixed and forced convection model for ceiling diffuser configurations
- 1.2.4 SurfaceConvectionAlgorithm:Inside = AdaptiveConvectionAlgorithm  
A dynamic algorithm that organizes a large number of different convection models and automatically selects the one that best applies

##### 1.3 Setting or Capability Used:

SurfaceConvectionAlgorithm:Inside = TARP

#### NOTE 2 - Outside convection algorithm

##### 2.1 Simulated Effect:

Convective heat transfer exchange related to exterior surfaces

##### 2.2 Optional Settings or Modeling Capabilities

- 2.2.1 SurfaceConvectionAlgorithm:Outside = SimpleCombined  
Applies constant heat transfer coefficients depending on surface orientation
  - 2.2.2 SurfaceConvectionAlgorithm:Outside = TARP  
Uses variable natural convection based on temperature difference
  - 2.2.3 SurfaceConvectionAlgorithm:Outside = MoWitt  
Derived from field measurements
  - 2.2.4 SurfaceConvectionAlgorithm:Outside = DOE-2  
A correlation from measurements by Klems and Yazdanian for rough surfaces
  - 2.2.5 SurfaceConvectionAlgorithm:Outside = AdaptiveConvectionAlgorithm  
A dynamic algorithm that organizes a large number of different convection models and automatically selects the one that best applies
- 2.3 Setting or Capability Used:  
SurfaceConvectionAlgorithm:Inside = DOE-2

NOTE 3 - Solar distribution effects for shaded surfaces

- 3.1 Simulated Effect:  
Determines how EnergyPlus treats beam solar radiation and reflectances from exterior surfaces that strike the building and, ultimately, enter the zone
- 3.2 Optional Settings or Modeling Capabilities
  - 3.2.1 Solar Distribution = MinimalShadowing  
No exterior shadowing except from window and door reveals. All beam solar radiation entering the zone is assumed to fall on the floor
  - 3.2.2 Solar Distribution = FullExterior  
Shadow patterns on exterior surfaces caused by detached shading, wings, overhangs, and exterior surfaces of all zones are computed
  - 3.2.3 Solar Distribution = FullInteriorAndExterior  
Same as FullExterior except that instead of assuming all transmitted beam solar falls on the floor, the program calculates the amount of beam radiation falling on each surface in the zone, including floor, walls and windows
  - 3.2.4 Solar Distribution = FullExteriorWithReflections  
Same as FullExterior except that instead of assuming all transmitted beam solar Shadow patterns on exterior surfaces caused by detached shading, wings, overhangs, and exterior surfaces of all zones are computed
  - 3.2.5 Solar Distribution = FullInteriorAndExteriorWithReflections  
Same as FullExterior except that instead of assuming all transmitted beam solar falls on the floor the program calculates the amount of beam radiation falling on each surface in the zone, including floor, walls and windows

3.3 Setting or Capability Used:

Solar Distribution = FullInteriorAndExterior

NOTE 4 - Simulation time increment

4.1 Simulated Effect:

The Timestep object specifies the "basic" timestep for the simulation

4.2 Optional Settings or Modeling Capabilities

4.2.1 Timestep = whole number between 1 and 60 divisible into 60

4.3 Setting or Capability Used:

Timestep = 6

NOTE 5 - Frequency of solar and shadow calculation

5.1 Simulated Effect:

Used to control details of the solar, shading, and daylighting models

5.2 Optional Settings or Modeling Capabilities

5.2.1 ShadowCalculation >= 1, default = 20

5.3 Setting or Capability Used:

ShadowCalculation = 20

### **C. Equivalent Modeling Methods**

NONE

### **D. Non-Specified Inputs**

NONE

### **E. Omitted Test Cases and Results**

NONE

### **F. Changes to Source Code for the Purpose of Running the Tests, Where Such Changes are Not Available in Publicly Released Versions of the Software**

NONE

### **G. Anomalous Results**

NONE

## **Appendix III-E: IES-VE, Integrated Environmental Solutions, United Kingdom**

### **Appendix A: Pro-Forma Questionnaire for Airside HVAC Tests, August 2015**

Instructions:

- *Replace empty boxes with symbols just below, as appropriate.*
  - Possible to use
  - Used to develop simulation trial results
- *Provide additional information and comments as appropriate; add lines as needed.*
- *If specific responses require disclosure of proprietary data not normally available to users in the program's engineering manual, leave the response blank, and include a note to the effect that including a response would require disclosure of proprietary information.*
- *If modeling varies among systems (and/or test cases), specify which systems (and/or test cases) are associated with each response; recopy a query as needed for this purpose (e.g., create 14a, 14b, etc.).*

**Program name, including version number**

ApacheSim Version: <VE>2014 Feature Pack 2

---

**Your name and organization**

Integrated Environmental Solutions Ltd

---

**1. Program status**

- Public domain
- Commercial
- Other (please specify): \_\_\_\_\_

**2. What weather inputs are used by the program for these tests?**

- Provided TMY2 weather data
- Directly entered DB and DP
- Directly entered DB and RH
- Directly entered DB and W
- Directly entered DB and WB
- Other (please specify): FWT weather files (proprietary format) have been created to provide the required weather inputs to ApacheSim. Each FWT files store DB and WB values on an hourly basis, linear interpolation is used to provide values at intermediate timesteps. The steady state conditions mean that the same DB and WB values are set for each hour. Five FWT files have been created.

Note: DB = outdoor dry-bulb temperature; DP = outdoor dew-point temperature; RH = outdoor relative humidity; W = outdoor humidity ratio; WB = outdoor wet-bulb temperature.

**2a. If weather inputs provided from TMY2 data, which data are used for calculating outdoor air humidity ratio?**

- DB and DP
- DB and RH
- Other (please specify): Outdoor humidity ratio is calculated from DB and WB temperatures defined in the FWT file.

**3. How frequently is the HVAC system air density calculated for each system?**

- Once per run, same value applied throughout system
- Once every time step, same value applied throughout system
- Once every time step, varies by location throughout system
- Other (please specify): Volume flow rates are defined at key locations on the HVAC network; these are converted to mass flow rates using a standard air density as a pre-simulation step. ApacheSim balances mass flow rates through the system. The mass flow rates are updated each timestep.

**4. What air property basis is applied for HVAC system air density calculations for each system?**

- Single fixed air condition (please specify): Density of air fixed at 1.2 kg/m<sup>3</sup>.
- Multiple fixed air conditions depending upon operating mode or system type (please specify): \_\_\_\_\_
- Air condition at each time step at a single specific system location \_\_\_\_\_ (please specify location): \_\_\_\_\_
- Local air conditions at each time step at each system location
- Other (please specify): ApacheSim tracks air mass flow rates in the HVAC system.

**5. What air pressure is used in HVAC system air density calculation?**

- Altitude adjusted standard pressure
- Weather file barometric pressure
- Other (please specify): ApacheSim tracks air mass flow rates in the HVAC system.

**6. Are fan/duct/system pressure differences accounted for in the HVAC air density calculations?**

- Yes
- No



**7. Are system models used for test cases the same as those commonly used by modelers to model similar systems?**

- Fan Coil:  Yes  No  
Single Zone:  Yes  No  
CV Reheat:  Yes  No  
VAV:  Yes  No

■ Other (please specify): ApacheHVAC provides a configurable network model allowing HVAC components to be placed and controlled to define the required system characteristics.

**8. How are zone loads defined?**

- Zone loads specified directly from input tables (e.g., Part I, Table 1-2)  
 Using alternative zone definition (e.g., Part I, Section 1.5.5.1.1.5)

**9. Thermostat Control**

- Perfect control  
 On/Off thermostatic control  
 On/Off thermostatic control with deadband  
■ Proportional control  
 Other (please specify): \_\_\_\_\_

**10. Were adjustments to zone temperature set points required to attain test case specified zone temperature?**

- Yes, consistent adjustment for all cases  
■ Yes, different adjustment for each case  
 No  
 Other, or additional discussion (please specify): \_\_\_\_\_

**11. Were adjustments to CV and VAV system supply air temperature set points required to attain test case specified system supply air temperature?**

- Yes, consistent adjustment for all cases  
■ Yes, different adjustment for each case  
 No  
 Other, or additional discussion (please specify): \_\_\_\_\_

**12. How was the zero bypass factor idealization modeled in FC/SZ test cases?**

- Directly, using the program's standard input scheme  
 Using custom supply air delivery temperature inputs for each test case  
 Other, or additional discussion (*describe*): \_\_\_\_\_  
 Was not successful

**13. How was the zero bypass factor idealization modeled in CV/VAV test cases?**

- Directly, using the program's standard input scheme  
 Using custom supply air delivery temperature inputs for each test case  
 Other, or additional discussion (*describe*): \_\_\_\_\_  
 Was not successful

**14. Which Section 1.6.5.n.3 specified detailed output variables were reported, and at which specified locations?**

Temperatures:       All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:*\_\_\_\_\_

Humidity Ratios:    All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:*\_\_\_\_\_

Specific Volumes:    All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:*\_\_\_\_\_

Enthalpies:             All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:*\_\_\_\_\_

Mass Flows:             All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:*\_\_\_\_\_

Note: Values not available directly from the results file have not be reported.

**15. For reported detailed variables, how were outputs generated?**

Temperatures:       Direct from model    Required unit conversion    Other post processing\*  
Humidity Ratios:    Direct from model    Required unit conversion    Other post processing\*  
Specific Volumes:    Direct from model    Required unit conversion    Other post processing\*  
Enthalpies:          Direct from model    Required unit conversion    Other post processing\*  
Mass Flows:          Direct from model    Required unit conversion    Other post processing\*

\* “Other post processing” includes calculations outside of the model, and/or output that is inferred as equivalent or approximately equivalent from another location, e.g., for the SZ system return air fan inlet conditions and zone air conditions might be assumed to be equal or approximately so.

Describe any post-processing (outside-the-model) calculations and/or assumptions used to produce detailed outputs: All values are obtained directly from VistaPro, a post processing tool packaged with the <VE> simulation suite.\_\_\_\_\_

## Appendix III-F

### Modeler Report for HVAC Air Distribution System Tests Life Cycle Energy Management (LCEM) Version 3.1

by

Kentaro Kimura  
Takasago Thermal Engineering  
Japan

March 2015

#### 1. Introduction

Test Program Name: LCEM Ver. 3.10 (LCEM: Life Cycle Energy Management software tool developed by Ministry of Land, Infrastructure, Transport and Tourism of Japan)

- how it models mechanical air distribution systems:  
In the calculation of LCEM tool Excel is used for iterative calculation function.
- timestep: 1 hour

#### 2. Modeling Assumptions

Test suites

Case	Outside air		Room No.	Design condition		Zone Thermal Load(W)			Coil Type
	DB (deg.c)	RH (%)		DB (°C)	RH (%)	Heating	Cooling	Latent	
AE101	-29.0	100	1	21.1	Uncontrolled	2931	0	586.1	High Heating
AE103	15.5	27	1	23.3	Uncontrolled	0	1465	586.1	Low Cooling, dry coil
AE104	26.9	75	1	23.9	Uncontrolled	0	2931	586.1	High Cooling, wet coil

#### *Assumed Conditions for simulation:*

- Temperature rise by the supply fan: 0.589 deg. C
- Temperature rise by the return fan: 0.295 deg. C
- Chilled water temperature: 7 deg. C
- Hot water temperature: 45 deg. C  
The inputs for chilled and hot water temperatures are the common values in Japan.
- Supply air relative humidity: 100%
- The specific volume is fixed as 833 L/kg in LCEM.

Additional information is included in the pro-forma questionnaire responses; see Appendix A of this supplementary modeler report.

#### 3. Modeling Difficulties

The suites of the 300- and 400-series test cases cannot be run because LCEM cannot make the model having a pre-heating coil as it is not common in Japan. Reheating is used quite often in the real system in Japan, but it cannot also be modeled unfortunately by the present version of LCEM.

At the first try there was failure to make an appropriate model because of a mistake in unit conversion. This was detected because of very different values compared to the results shown in the report. However, if there is no reference results it is very difficult to detect this fault.

#### 4. Software Errors Discovered and/or Comparison Between Different Versions of the Same Software

LCEM includes many coil models which were developed by different manufacturers and Mr. Kimura found that one of them failed due to a software fault, which must be fixed.

Initially supply and return fan temperature rises were input as 0.5°C and 0.3°C, respectively. For later-round simulation trial results the test spec authors observed that the temperature across the supply and return fans differed from the other programs. The fan air temperature rise is not calculated by LCEM and it must be input. The input was recalculated and adjusted to:  
Supply fan air temperature rise = 0.589°C (before = 0.5°C)  
Return fan air temperature rise = 0.295°C (before = 0.3°C)

The calculations are based on:

$$\Delta T = \Delta P / (\eta \cdot \rho \cdot C_p), \text{ where}$$

$\Delta P$  = fan total pressure rise, 498 Pa for the supply fan, and 249 Pa for the return fan, see test spec [Part I] Sections 1.5.5.2.1.1.1 and 1.5.5.2.1.1.2  
 $\eta$  = 0.7, fan mechanical efficiency from test spec sections 1.5.5.2.1.1.1 and 1.5.5.2.1.1.2  
 $\rho$  = 1.2 kg/m<sup>3</sup>, air density, constant value in LCEM  
 $C_p$  = 1006 J/(kg·K), air specific heat, constant value in LCEM.

#### 5. Results

Case AE101: Succeeded

Case AE103: Succeeded

Case AE104: Succeeded

- The cooling coil outlet humidity became 95%. This is because LCEM limits the relative humidity less than 95%. Although if a user has knowledge to eliminate the restriction he can change it; so this restriction was eliminated, allowing cooling coil outlet relative humidity of 100%.

Following is a summary of remaining minor disagreements identified by the NREL team test spec authors for Simulation Trial Round 3 (spec distributed 10/12/14), with logical reasons for differences described. Figures 3F-1 through 3F-10 are provided to support the discussion, and indicate LCEM results for October 2014 and January 2015, along with others' results. *[Editor's Note: These figures are included after Section 8 of this modeler report.]* The primary difference in the LCEM results is the fan heat input, described above, as shown in Figures 3F-5 and 3F-10. The Jan 2015 results also include the following corrections in precision:

##### 1. Cancellation of significant digits

An example: Supply Airflow Rate 1019.41 CMH (before = 1019 CMH)

## 2. Input errors of ambient conditions

An example: Relative humidity 100% (before = 99.983%).

### ***Sensible and Latent Cooling Loads***

LCEM's total coil loads, shown in Figure 3F-1, indicate generally better agreement with the quasi-analytical solution (QAS) than its sensible and latent cooling coil loads, shown in Figures 3F-2 and 3F-3, respectively. The sensible coil load results trend lower as much as 4% when compared to the QAS results and the latent coil loads trend higher by as much as 9%. The cooling coil leaving air temperatures in Figure 3F-4 are very close to or slightly higher than the QAS results due to LCEM's reduced supply fan heat gain shown in Figure 3F-5. This should result in slightly lower latent loads rather than the higher latent loads shown for LCEM. In the wet coil cases the cooling coil leaving RH shown in Figure 3F-6 is at 100%, indicating the model achieved zero by-pass factor, and the zone latent gain as shown in Figure 3F-7 is very close to the QAS results. The absence of other explanations leads to the possibility that there may be a difference in how LCEM attributes total cooling load to the sensible and latent components, which may explain some of the sensible and latent coil load difference.

### ***Mass Flow Differences***

Outdoor air mass flow, presented in Figure 3F-8, is 1.5%-3% higher than the QAS results in all non-economizer cooling cases. Supply air mass flow is shown in Figure 3F-9 and is 3% high in the FC and SZ system heating cases, and ranges from 0.4% low to 1.3% high in the cooling cases. These differences result from LCEM using a fixed specific volume of 833 L/kg in calculating supply, exhaust, and outdoor air mass flow rates. This causes the predicted result supply mass flow rate to be one value for all cooling cases and a slightly different value for both heating cases, rather than a different value for each test case as in the QAS. This approach is common among the participant programs.

### ***FC and SZ Supply Fan and SZ Return Fan Heat Gain***

Relative to the QAS for the final (Jan 2015) results, the supply fan heat gain, shown in Figure 3F-5, is low in Cases AE101 and AE201 and high in all cooling cases. Relative to the QAS, the return fan heat gain, shown in Figure 3F-10, is high in Case AE201 and low in all other SZ cases. These differences appear large in relative terms but are only a 0.01° C to 0.04° C difference in air temperature rise. As discussed previously, fan temperature rise is an input to the LCEM and has been entered as a constant value; it therefore differs from the QAS where variation of the fan inlet conditions results in different fan mass flows. Given the small impact on coil loads this seems like a reasonable modeling simplification.

With respect to the change in fan heat inputs between the Oct 2014 and Jan 2015 results, the effect on changes to other outputs is logical. For example the rise in supply fan heat results in reduced cooling coil outlet, or fan inlet, air temperature (see Figure 3F-4), so that the resulting zone supply air temperature is the same as before; this causes increased sensible cooling coil loads (see Figure 3F-2). The reduced cooling coil outlet air temperature also drives greater latent loads (see Figure 3F-3) because the coil is colder.

## **6. Modeling Notes for ASHRAE Standard 140 (S140outNotes.txt)**

See Appendix B of this supplementary modeler report.

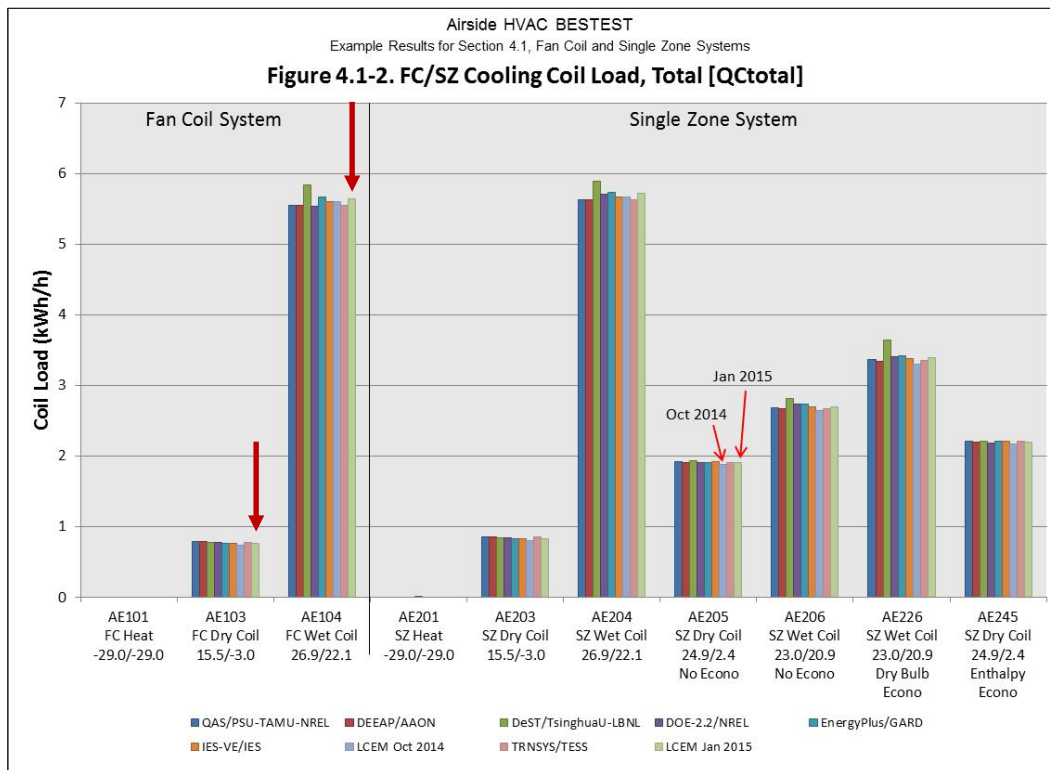
## **7. Conclusions and Recommendations**

The test suite was very useful. Comparing to the other programs results, we noticed the input mistakes.

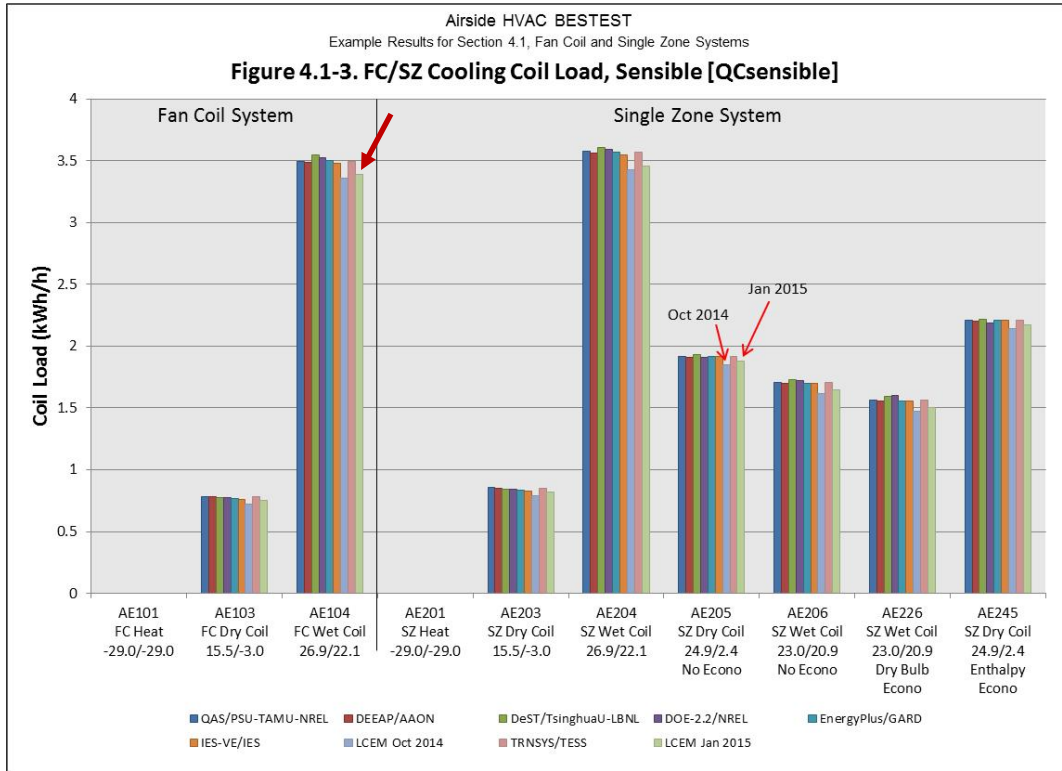
## 8. References

M.Ito, S.Murakami, et al., 2007. Development of HVAC System Simulation Tool for Life Cycle Energy Management, Part-1: Outline of the Development Simulation Tool for Life Cycle Energy Management, Proceedings of BS07, pp.1610-1621.

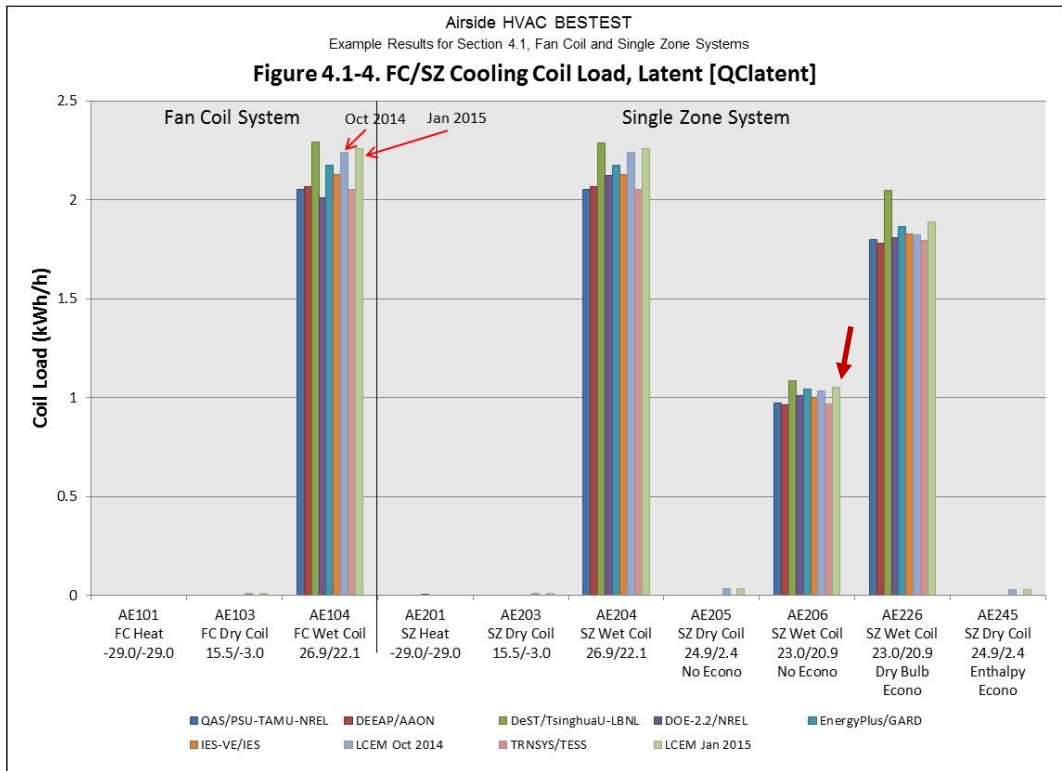
Y.Sugihara, H.Niwa, et al., 2007. Development of HVAC System Simulation Tool for Life Cycle Energy Management, Part-2: Development of Component Models for HVAC Equipment, Proceedings of BS07, pp.1616-1622.



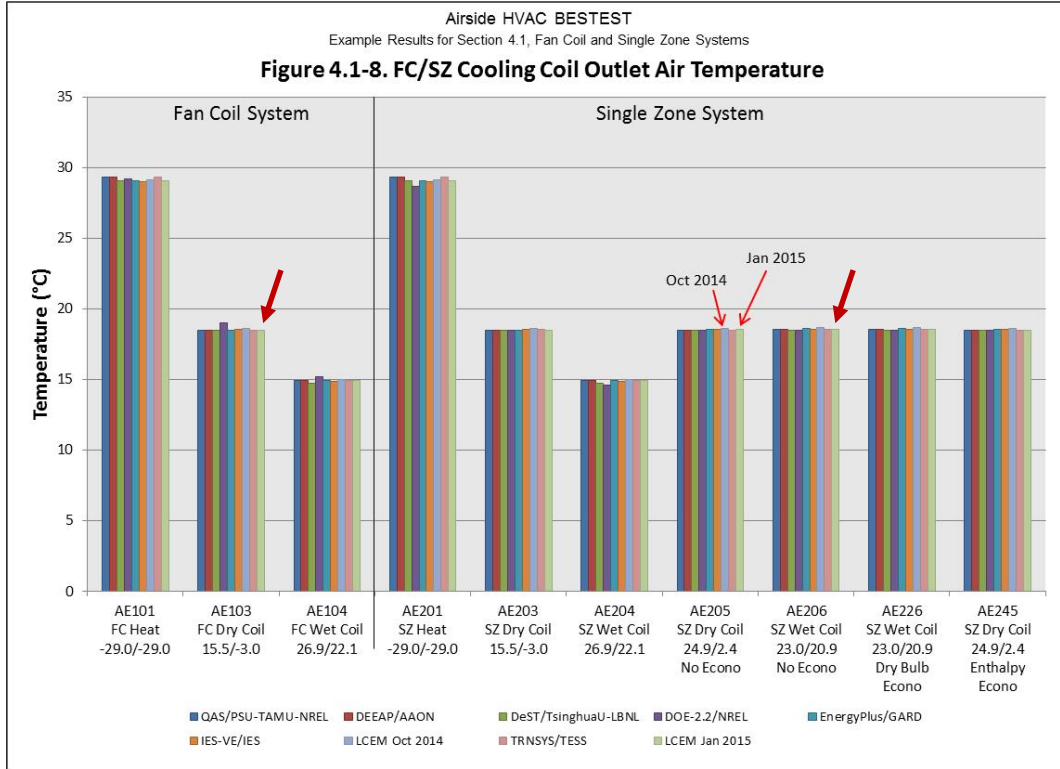
**Figure 3F-1. Cooling Coil Total Load**



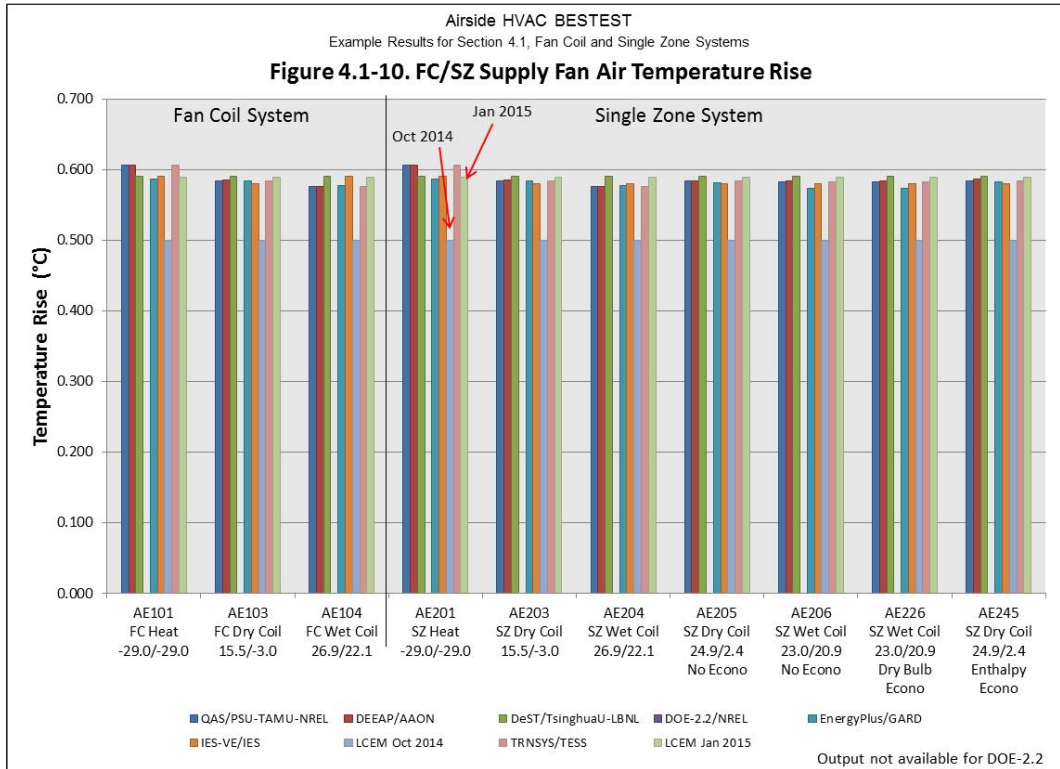
**Figure 3F-2. FC/SZ Cooling Coil Sensible Load**



**Figure 3F-3. FC/SZ Cooling Coil Latent Load**

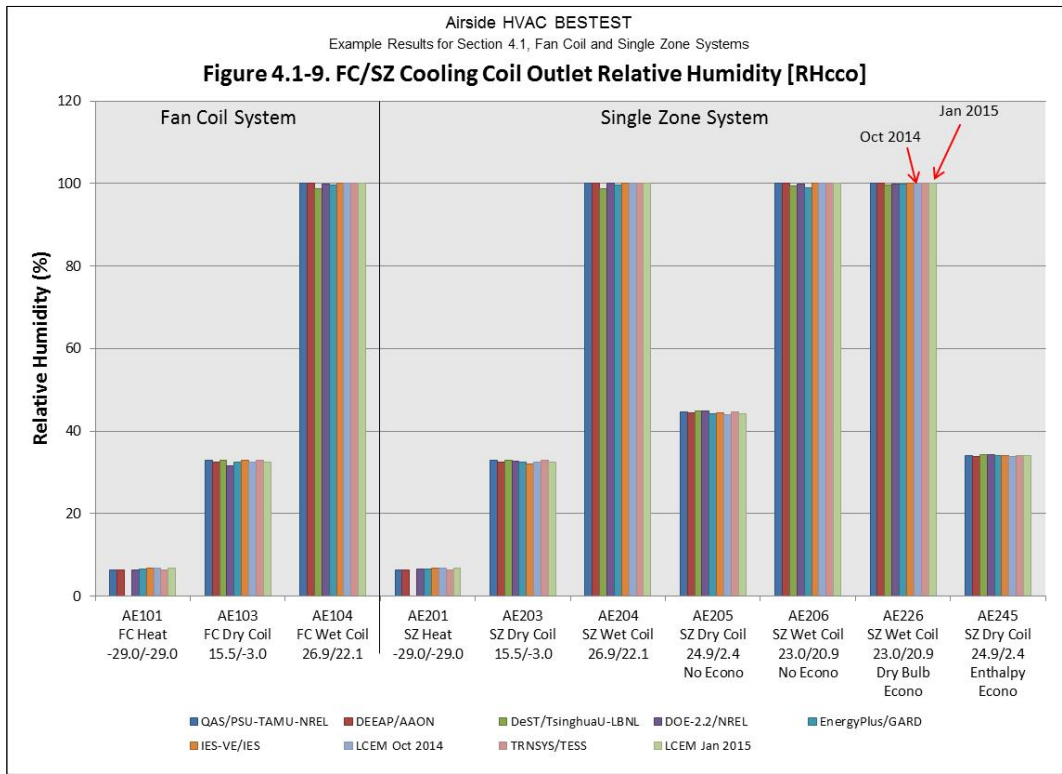


**Figure 3F-4. FC/SZ Cooling Coil Outlet Air Temperature**

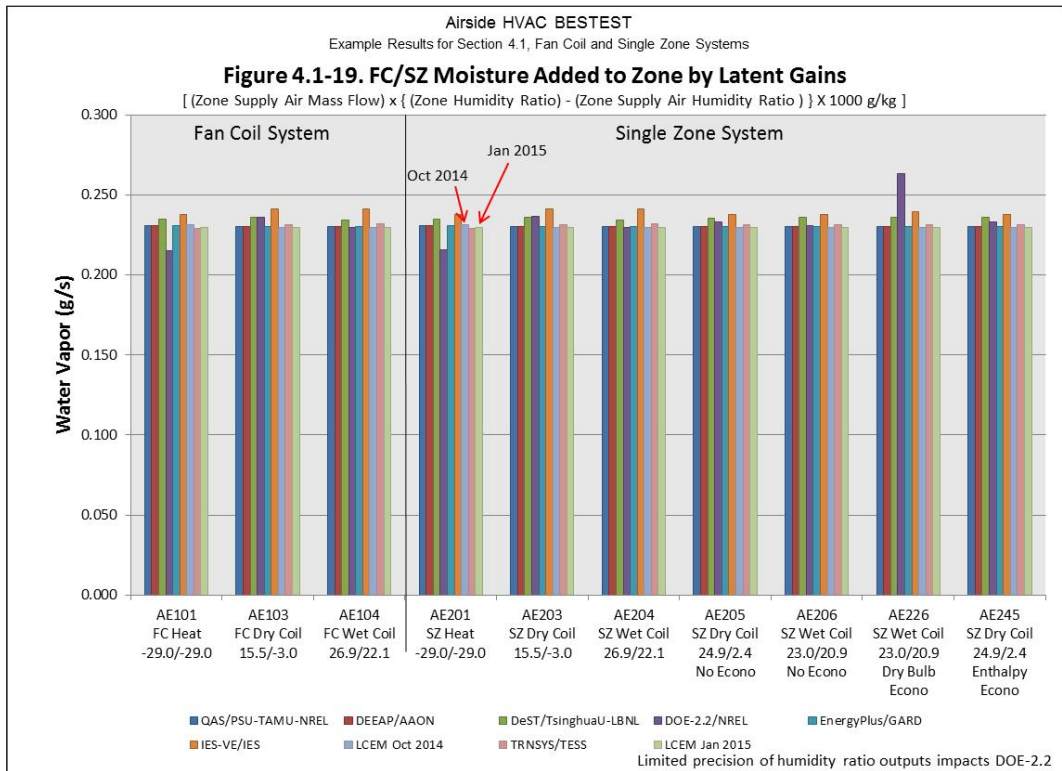


**Figure 3F-5. FC / SZ Supply Fan Air Temperature Rise**





**Figure 3F-6. FC/SZ Cooling Coil Leaving Air Relative Humidity**



**Figure 3F-7. Moisture Added to Zone**

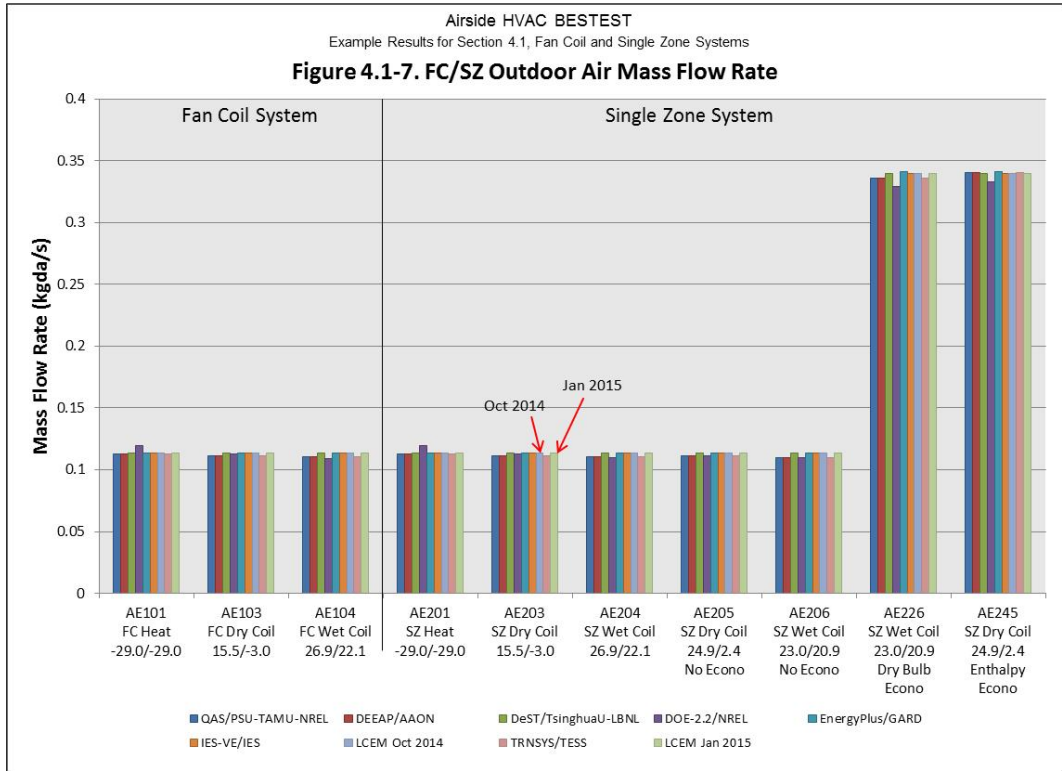


Figure 3F-8. FC/SZ Outdoor Air Mass Flow Rate

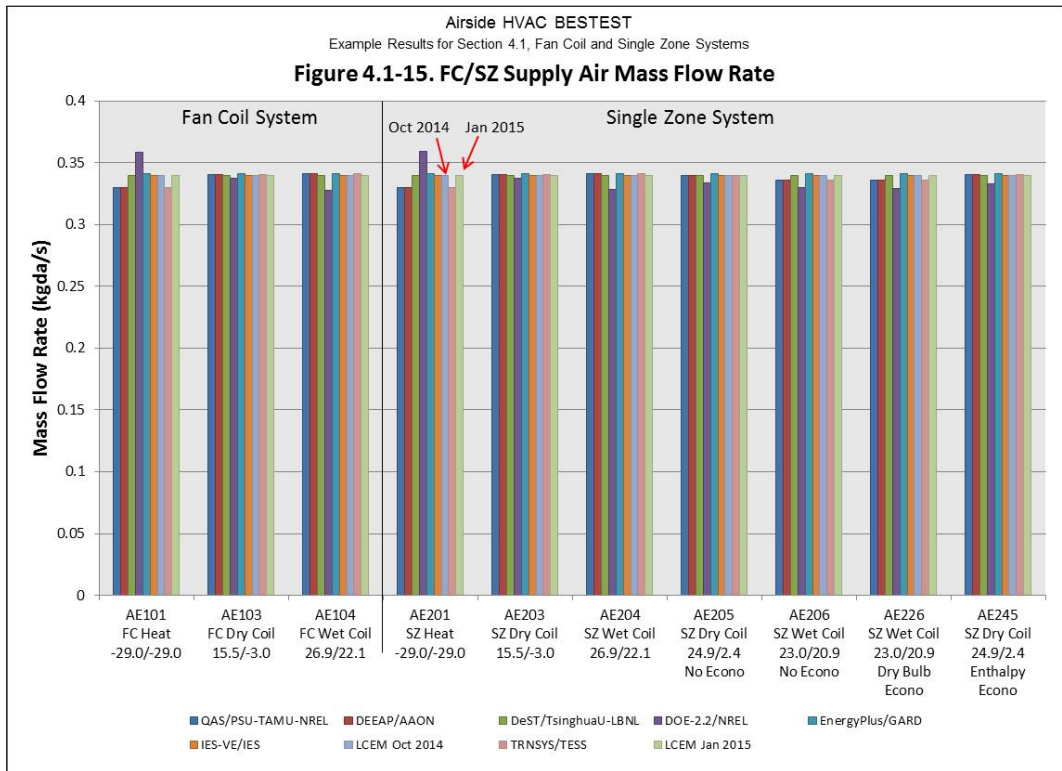
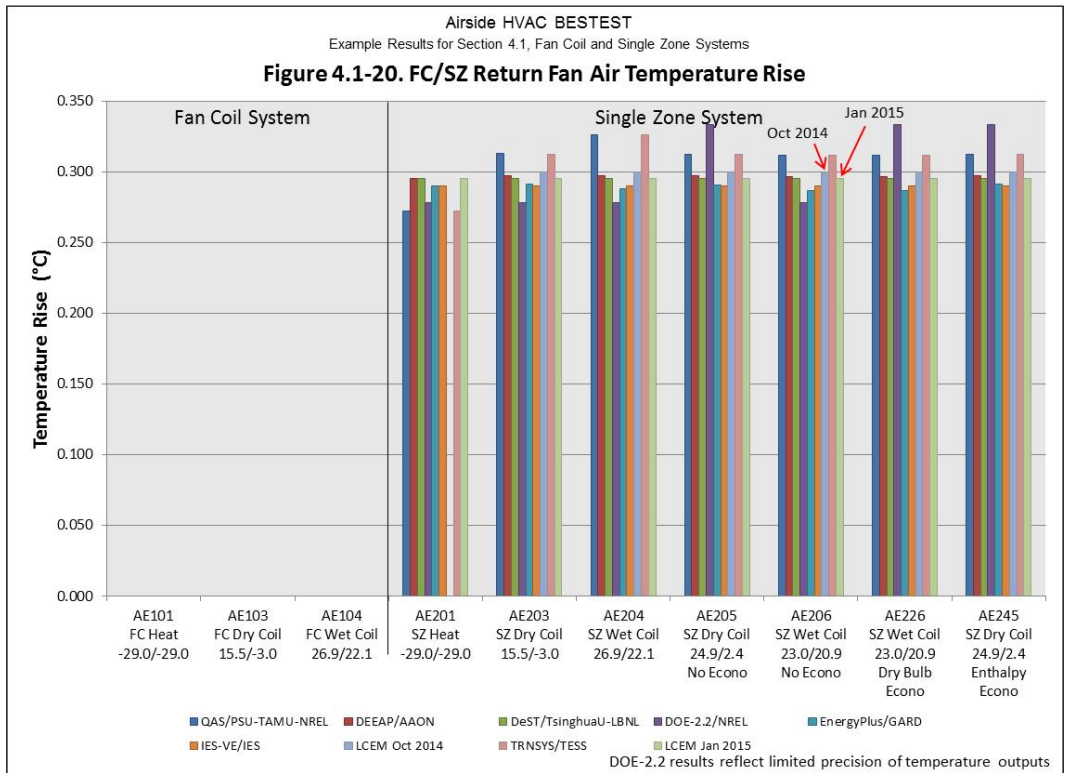


Figure 3F-9. FC/SZ Supply Air Mass Flow Rate



**Figure 3F-10. SZ Return Fan Air Temperature Rise**

## Appendix A: Pro-Forma Questionnaire for Airside HVAC Tests, October 2014

### *Instructions:*

- *Replace empty boxes with symbols just below, as appropriate.*
  - Possible to use
  - Used to develop simulation trial results
- *Provide additional information and comments as appropriate; add lines as needed.*
- *If specific responses require disclosure of proprietary data not normally available to users in the program's engineering manual, leave the response blank, and include a note to the effect that including a response would require disclosure of proprietary information.*
- *If modeling varies among systems (and/or test cases), specify which systems (and/or test cases) are associated with each response; recopy a query as needed for this purpose (e.g., create 14a, 14b, etc.).*

### **Program name, including version number**

LCEM Tool Ver3.10

---

### **Your name and organization**

Kentaro Kimura / Takasago Thermal Engineering Co., Ltd.

---

**1. Program status**

- Public domain
- Commercial
- Other (please specify): \_\_\_\_\_

**2. What weather inputs are used by the program for these tests?**

- Provided TMY2 weather data
- Directly entered DB and DP
- Directly entered DB and RH
- Directly entered DB and W
- Directly entered DB and WB

Note: DB = outdoor dry-bulb temperature; DP = outdoor dew-point temperature; RH = outdoor relative humidity; W = outdoor humidity ratio; WB = outdoor wet-bulb temperature.

**3. How frequently is the HVAC system air density calculated for each system?**

- Once per run, same value applied throughout system
- Once every time step, same value applied throughout system
- Once every time step, varies by location throughout system
- Other (please specify): Air density fixed to 1.2 (kg/m<sup>3</sup>).

**4. What air property basis is applied for HVAC system air density calculations for each system?**

- Single fixed air condition (please specify): \_\_\_\_\_
- Multiple fixed air conditions depending upon operating mode or system type (please specify): \_\_\_\_\_
- Air condition at each time step at a single specific system location \_\_\_\_\_ (please specify location): \_\_\_\_\_
- Local air conditions at each time step at each system location
- Other (please specify): Air density fixed to 1.2 (kg/m<sup>3</sup>).

**5. What air pressure is used in HVAC system air density calculation?**

- Altitude adjusted standard pressure
- Weather file barometric pressure
- Other (please specify): Air density fixed to 1.2 (kg/m<sup>3</sup>).

**6. Are fan/duct/system pressure differences accounted for in the HVAC air density calculations?**

- Yes
- No

**7. Are system models used for test cases the same as those commonly used by modelers to model similar systems?**

- Fan Coil:       Yes    No
- Single Zone:    Yes    No
- CV Reheat:      Yes    No   CV test cases not run
- VAV:             Yes    No   VAV test cases not run

**8. How are zone loads defined?**

- Zone loads specified directly from input tables (e.g., Part I, Table 1-2)
- Using alternative zone definition (e.g., Part I, Section 1.5.5.1.1.5)

**9. Thermostat Control**

- Perfect control
- On/Off thermostatic control
- On/Off thermostatic control with deadband
- Proportional control
- Other (please specify): \_\_\_\_\_

**10. Were adjustments to zone temperature set points required to attain test case specified zone temperature?**

- Yes, consistent adjustment for all cases
- Yes, different adjustment for each case
- No
- Other, or additional discussion (please specify): \_\_\_\_\_

**11. Were adjustments to CV and VAV system supply air temperature set points required to attain test case specified system supply air temperature?**

- Yes, consistent adjustment for all cases
- Yes, different adjustment for each case
- No
- Other, or additional discussion (please specify): It is not possible to simulate CV/VAV test cases because this program can not model reheat system.

**12. How was the zero bypass factor idealization modeled in FC/SZ test cases?**

- Directly, using the program's standard input scheme
- Using custom supply air delivery temperature inputs for each test case
- Other, or additional discussion (*describe*): \_\_\_\_\_
- Was not successful

**13. How was the zero bypass factor idealization modeled in CV/VAV test cases?**

- Directly, using the program's standard input scheme
- Using custom supply air delivery temperature inputs for each test case
- Other, or additional discussion (*describe*): It is not possible to simulate CV/VAV test cases because this program can not model reheat system.
- Was not successful

**14. Which Section 1.6.5.n.3 specified detailed output variables were reported, and at which specified locations?**

Temperatures:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:* \_\_\_\_\_

Humidity Ratios:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:* \_\_\_\_\_

Specific Volumes:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:* \_\_\_\_\_

Enthalpies:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:* \_\_\_\_\_

Mass Flows:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:* \_\_\_\_\_

**15. For reported detailed variables, how were outputs generated?**

Temperatures:  Direct from model  Required unit conversion  Other post processing\*

Humidity Ratios:  Direct from model  Required unit conversion  Other post processing\*

Specific Volumes:  Direct from model  Required unit conversion  Other post processing\*

Enthalpies:  Direct from model  Required unit conversion  Other post processing\*

Mass Flows:  Direct from model  Required unit conversion  Other post processing\*

\* “Other post processing” includes calculations outside of the model, and/or output that is inferred as equivalent or approximately equivalent from another location, e.g., for the SZ system return air fan inlet conditions and zone air conditions might be assumed to be equal or approximately so.

Describe any post-processing (outside-the-model) calculations and/or assumptions used to produce detailed outputs: Specific volumes fixed to 833 (L/kgDA).

## Appendix B. Modeling Notes for ASHRAE Standard 140

*[Editor's note: Content here summarized from S140outNotes.txt form submitted during the simulation trials.]*

### A. Software Information

1. Vendor:  
Ministry of Land, Infrastructure, Transportation and Tourism, Japan
2. Software Name:  
LCEM (Life Cycle Energy Management tool)
3. Version (unique software version identifier):  
3.10
4. Operating System Requirements:  
Windows8  
Windows7  
Windows Vista  
Windows XP  
Windows 2000
5. Approximate Hard Disk Space Required for Installation:  
94.2 MB
6. Minimum RAM Required for Software Operation:  
NONE
7. Minimum Display Monitor Requirements:  
NONE
8. Other Hardware or Software-Related Requirements:  
Microsoft Excel 2013/2010/2003/2000

### B. Alternative Modeling Methods

NONE

### C. Equivalent Modeling Methods

NOTE 1 - Fan air temperature rise  
(Section 1.5.5.2.1.1) [see Part I of this document]

- 1.1 Simulated Effect:  
Fan air temperature rise
- 1.2 Sections of the Test Specification where Relevant Inputs are Specified:  
1.5.5.2.1.1 and 1.5.5.2.1.2 [see Part I of this document]
- 1.3 Equivalent Inputs Used:



Temperature rise by the supply fan: 0.589 deg. C  
Temperature rise by the return fan: 0.295 deg. C

1.4 Physical, Mathematical or Logical Justification of the Equivalent Inputs:

The calculations are based on:

$\Delta T = \Delta P / (\eta \cdot \rho \cdot C_p)$ , where

$\Delta P$  = fan total pressure rise, 498 Pa for the supply fan,

and 249 Pa for the return fan, see test spec [Part I] Sections 1.5.5.2.1.1.1 and 1.5.5.2.1.1.2

$\eta = 0.7$ , fan mechanical efficiency from test spec [Part I] sections 1.5.5.2.1.1.1 and 1.5.5.2.1.1.2

$\rho = 1.2 \text{ kg/m}^3$ , air density, constant value in LCEM

$C_p = 1006 \text{ J/(kg}\cdot\text{K)}$ , air specific heat, constant value in LCEM.

NOTE 2 - Cooling coil bypass factor (BF)=0

2.1 Simulated Effect:

Cooling coil bypass factor (BF)=0

2.2 Sections of the Test Specification where Relevant Inputs are Specified:

1.5.5.1.1.3 [see Part I]

2.3 Equivalent Input Used:

Cooling coil outlet relative humidity changed from 95% to 100%.

2.4 Physical, Mathematical or Logical Justification of Equivalent Input:

LCEM cannot set the BF.

**D. Non-Specified Inputs**

NONE

**E. Omitted Test Cases and Results**

NOTE 1 - Operation of Pre-heating, Cooling and Reheat Coils.

1.1 Cases where Results Were Omitted, and which Results Were Omitted for the Cases:

AE301 AE303 AE304 AE305 AE306 AE326 AE345

AE401 AE403 AE404 AE405 AE406 AE426 AE445

1.2 Explanation for Omitting Test Case Results:

The suites of the 300- and 400-series test cases cannot be run because LCEM cannot make the model having a pre-heating coil as it is not common in Japan. Reheating is used quite often in the real system in Japan, but it cannot also be modeled unfortunately by the present version of LCEM.

**F. Changes to Source Code for the Purpose of Running the Tests, Where Such Changes are Not Available in Publicly Released Versions of the Software**

NONE

**G. Anomalous Results**

NONE

## **Appendix III-G**

### **Modeler Report for HVAC Air Distribution System Tests**

#### **TRNSYS with TESS Libraries Version 17.01.0028**

by

**Timothy P. McDowell  
Thermal Energy System Specialists  
United States**

**March 2015**

#### **1. Introduction**

**SOFTWARE NAME: TRNSYS with TESS Libraries**

**SOFTWARE VERSION: 17.01.0028**

**SOFTWARE VENDOR: TESS, Madison, Wisconsin, United States**

TRNSYS is an extremely flexible graphically based software environment used to simulate the behavior of transient systems. While the vast majority of simulations are focused on assessing the performance of thermal and electrical energy systems, TRNSYS can equally well be used to model other dynamic systems such as traffic flow, or biological processes.

TRNSYS is made up of two parts. The first is an engine (called the kernel) that reads and processes the input file, iteratively solves the system, determines convergence, and plots system variables. The kernel also provides utilities that (among other things) determine thermophysical properties, invert matrices, perform linear regressions, and interpolate external data files. The second part of TRNSYS is an extensive library of components, each of which models the performance of one part of the system. The standard library includes approximately 150 models ranging from pumps to multizone buildings, wind turbines to electrolyzers, weather data processors to economics routines, and basic HVAC equipment to cutting edge, emerging technologies. Models are constructed in such a way that users can modify existing components or write their own, extending the capabilities of the environment.

TIMESTEP: Unlimited.

#### **2. Modeling Assumptions**

Additional information is included in the pro-forma questionnaire responses; see Appendix A of this supplementary modeler report.

Information regarding “Modeling Options” is included with Report Block B of S140outNotes.txt (see Appendix B of this modeler report).

#### **3. Modeling Difficulties**

*[Editor’s Note: None reported.]*

#### **4. Software Errors Discovered, Program Improvements, Input Corrections, and/or Other Comparisons Between Different Versions of the Same Software**

No software errors were found, but some new and modified component models were created.

Several input changes were made over the course of the simulation trials.

##### ***Early Round Input Revisions Related to Test Spec Clarifications for the CV and VAV Systems***

Clarifications of the test specification for simulation trial #2a (December 2012) included comprehensive revisions related to: system airflow, fan configuration, coil operation, schematic diagrams, and economizer control. Input revisions for the CV and VAV systems for consistency with the test specification revisions resulted in up to 7% difference in total cooling coil loads for the CV and VAV systems, based on comparison of results submitted January 10, 2013 versus those submitted June 7, 2012.

##### ***Delta AE201-AE101 Heating Coil Load Sensitivity to System Pressure***

For later-round simulation trial results the test spec authors observed that the delta AE201-AE101 heating coil load sensitivity was less negative than the merged solution [defined as the quasi-analytical solution (QAS) for the final report] or other program results. This output sensitivity should represent the impact of return fan heat, but it is sensitive to any other differences between the FC and SZ models. The delta heating coil load is smaller because the AE101 heating load estimate is lower (0.5%) than the merged solution and the AE201 heating load estimate agrees.

This was caused by a difference in airflow rates. The difference in airflow rates was caused by different assumptions concerning the system pressure at different points in the system model. Based on a clarifying conversation where we determined that the standard assumes constant pressure throughout the system (except for fan power and temperature changes), all of the models have been converted to use a constant pressure of 1 atm. The heating coil load difference is now in better agreement with other programs and the quasi-analytical solution (QAS).

##### ***OA Humidity Ratio***

The test specification authors observed the OA humidity ratio was slightly higher in all cooling cases for the round 2 through round 3b results. The round 2 through 3b test case specifications specify the ambient humidity ratio as exact and equivalent moisture parameters for wet bulb, dew point, and relative humidity are calculated using a real gas correlation.

TRNSYS is using the humidity ratio input directly. In some cases this may be adjusted to the maximum possible humidity ratio for the input dry bulb temperature and atmospheric pressure. This check is done using perfect gas laws.

In rounds 2 through 3b, the QAS calculated the outdoor air humidity ratio using perfect gas laws from the dry bulb and wet bulb temperatures. This calculated value is not the same as the value specified in the test cases, which is based on a real gas correlation. In the final round, the specification switched to using dew point as the exact variable, and the QAS was revised to input the dew point temperature. The dew point temperature was also used as input in the final TRNSYS models, which now have results in better agreement with the QAS values.

### ***SZ and CV System Return Fan Heat Gain***

For later-round simulation trial results the test spec authors observed that relative to the QAS, the return fan heat gain is 8% high in Case AE201 and low in all other SZ and CV cases. These differences appear large in relative terms but are only a 0.03°C to 0.06°C difference in return air temperature rise.

This was because the return fan pressure change was not being changed with the change in flow rate for the non-VAV cases. This has now been added to all of the cases. It does raise once again the issue of defining something based on the volumetric flow rate without defining at what state the density of the air is being used when the software will be using the mass flow rate of air for the rest of the system modeling.

### ***CV Reheat Loads***

For later simulation trial rounds the spec authors observed that the CV system reheat coil loads were 0.3% - 2.5% lower than the merged solution in the cooling cases. When it comes to applying the zone loads there is a fundamental difference in how TRNSYS makes the calculation from how the QAS would make the calculation. TRNSYS does not assume that there is a zone at the setpoint. It calculates the effect of the “zone” loads on an air stream representing each zone and calculates the resultant air conditions exiting the zone air streams. The reheat coil leaving air temperatures are varied until the resultant air temperatures leaving each zone air stream are at the specified “zone” setpoints. What this means is that the air properties (and thus the specific heat) are calculated at the “zone supply” conditions and not the “zone” conditions. So there should be some difference between the results. With the change of the outdoor air properties calculated from dew point, the humidity ratios are more similar between TRNSYS and the QAS, and this has resulted in better agreement between the “zone” and “zone supply” air specific heats.

### ***Latent Cooling Loads***

In the round 3a runs latent coil loads were 1.7% to 4% higher than the QAS. Several factors contributed to this. The previously mentioned ambient moisture input difference resulted in TRNSYS results, which used the specified humidity ratio directly, having an ambient moisture level 0.5% higher than that calculated by the QAS. Three additional factors were also involved: condensate leaving condition, zone latent loads, and constant system pressure assumption (the system pressure and the zone latent loads may be inter-related). Discussion follows.

#### ***Condensate Leaving Condition***

The condensate leaving condition is not specified in the test case specification. TRNSYS assumed that the condensate left the system with zero enthalpy and that the energy to cool the condensate from the condensing temperature to zero was part of the latent load. The round 2 through round 3b specifications did not state the condensate leaving condition but the QAS assumes that condensate leaves the system at the cooling coil leaving air temperature. Since the condensate left the system at a lower enthalpy in TRNSYS than in the QAS, and because the energy to cool the condensate from the condensation point to the leaving condition was attributed to latent load, TRNSYS’ latent load was higher. With the final test case specification these terms are more clearly defined and the TRNSYS models were revised, and now produce results in better agreement with the QAS.

#### ***Zone Latent Loads***

The moisture added to the zones was calculated and found to be higher than the QAS. The difference is caused by different formulas for calculating the latent effect on the humidity ratio. The TRNSYS model

uses:

$$Q_{\text{latent}} = \dot{m}_{\text{supply}} * h_{\text{fg}} * (W_{\text{zone}} - W_{\text{supply}}),$$

where

$h_{\text{fg}}$  is the enthalpy difference of water at liquid and vapor states at the zone temperature,  
 $\dot{m}_{\text{supply}}$  is zone supply air mass flow rate,  
 $W_{\text{zone}}$  is humidity ratio of air exiting the “zone” air stream  
 $W_{\text{supply}}$  is humidity ratio of the zone supply air.

The test spec and QAS use:

$$Q_{\text{latent}} = \dot{m}_{\text{supply}} * ((W_{\text{zone}} - W_{\text{supply}}) * (i_o + C_p * T_{\text{zone}})),$$

where:

$i_o$  is the enthalpy of water vapor at 0 degrees,  
 $C_p$  is the specific heat of water vapor,  
 $T_{\text{zone}}$  is the zone air temperature.

For  $i_o$ , it was not clear from the test spec whether the 0 degrees is C, F or K, and there is no reference temperature for the specific heat.

The TRNSYS model was changed to use the  $(i_o + C_p * T_{\text{zone}})$ . However, TRNSYS does not have a zone temperature. As discussed above (see “CVReheat Loads”), it calculates the resultant temperature from a load imposed on an air stream rather than calculating the load required to maintain a setpoint. So in the TRNSYS formulation of  $Q_{\text{latent}}$ , then  $W_{\text{zone}}$ ,  $T_{\text{zone}}$ , and  $C_p$  are evaluated at the zone air inlet conditions.

### *Constant System Pressure Assumption*

An additional issue that is responsible for some moisture variance arose as a direct consequence of changing the TRNSYS model to use constant pressure (see above section “Delta AE201-AE101 Heating Coil Load Sensitivity to System Pressure”). The air conditions leaving the supply fan are calculated using the air pressure after the pressure rise from the fan. The fan model cannot be changed to a constant pressure model, because the pressure rise is used to determine the fan power and thus the heat added to the airstream. When the temperature and humidity ratio from that calculation are used in the next component with the constant system pressure you get a different humidity ratio.

### ***Other***

Other issues between TRNSYS and the QAS evident in earlier results sets that have been resolved by the above changes or through incidental changes include:

- Sensible cooling coil load within about 1 or 2% of the QAS, but relatively higher in wet coil cases than in dry coil cases
- Delta sensible cooling (AE2xx-AE1xx and AE3xx-AE4xx) is slightly low in all non-economizer cases.

## 5. Results

### *Remaining Minor Disagreements with Quasi-Analytical Solution*

Following is summary of remaining minor disagreements identified by the NREL team test spec authors for Simulation Trial Final Round (12/2/14):

- Return fan mass flow rate is 1.2% lower than the QAS in cases AE405, AE406, AE426, and AE445.
- Temperature rise across the supply fan in the VAV cooling cases is generally 0.5% lower than the QAS prediction and CASE AE404 is 2.4% lower.
- Supply air specific volume is lower than the QAS in all cases (~0.5%) but mass flows are in near perfect agreement. This is caused by calculating the supply fan leaving air conditions at the leaving air pressure which is greater than 1 atmosphere.

## 6. Other (optional)

- None

## 7. Conclusions and Recommendations

*[Editor's Note: None reported]*

## 8. References

TRNSYS 17. (2015). <http://www.tess-inc.com/trnsys>.

## Appendix A. Pro-Forma Questionnaire for Airside HVAC Tests, October 2014

### *Instructions:*

- *Replace empty boxes with symbols just below, as appropriate.*
  - Possible to use
  - Used to develop simulation trial results
- *Provide additional information and comments as appropriate; add lines as needed.*
- *If specific responses require disclosure of proprietary data not normally available to users in the program's engineering manual, leave the response blank, and include a note to the effect that including a response would require disclosure of proprietary information.*
- *If modeling varies among systems (and/or test cases), specify which systems (and/or test cases) are associated with each response; recopy a query as needed for this purpose (e.g., create 14a, 14b, etc.).*

### **Program name, including version number**

TRNSYS 17.02.0028

---

### **Your name and organization**

Timothy P McDowell – Thermal Energy System Specialists, LLC

---

**1. Program status**

- Public domain
- Commercial
- Other (please specify): \_\_\_\_\_

**2. What weather inputs are used by the program for these tests?**

- Provided TMY2 weather data
- Directly entered DB and DP
- Directly entered DB and RH
- Directly entered DB and W
- Directly entered DB and WB

Note: DB = outdoor dry-bulb temperature; DP = outdoor dew-point temperature; RH = outdoor relative humidity; W = outdoor humidity ratio; WB = outdoor wet-bulb temperature.

**3. How frequently is the HVAC system air density calculated for each system?**

- Once per run, same value applied throughout system
- Once every time step, same value applied throughout system
- Once every time step, varies by location throughout system
- Other (please specify): \_\_\_\_\_

**4. What air property basis is applied for HVAC system air density calculations for each system?**

- Single fixed air condition (please specify): \_\_\_\_\_
- Multiple fixed air conditions depending upon operating mode or system type (please specify): \_\_\_\_\_
- Air condition at each time step at a single specific system location \_\_\_\_\_ (please specify location): \_\_\_\_\_
- Local air conditions at each time step at each system location
- Other (please specify): \_\_\_\_\_

**5. What air pressure is used in HVAC system air density calculation?**

- Altitude adjusted standard pressure
- Weather file barometric pressure
- Other (please specify): Test specification pressure directly

**6. Are fan/duct/system pressure differences accounted for in the HVAC air density calculations?**

- Yes
- No

**7. Are system models used for test cases the same as those commonly used by modelers to model similar systems?**

- Fan Coil:  Yes  No
- Single Zone:  Yes  No
- CV Reheat:  Yes  No
- VAV:  Yes  No



**8. How are zone loads defined?**

- Zone loads specified directly from input tables (e.g., Part I, Table 1-2)
- Using alternative zone definition (e.g., Part I, Section 1.5.5.1.1.5)

**9. Thermostat Control**

- Perfect control
- On/Off thermostatic control
- On/Off thermostatic control with deadband
- Proportional control
- Other (please specify): \_\_\_\_\_

**10. Were adjustments to zone temperature set points required to attain test case specified zone temperature?**

- Yes, consistent adjustment for all cases
- Yes, different adjustment for each case
- No
- Other, or additional discussion (please specify): Zone setpoints only used in VAV. In the CV cases the temperatures leaving the reheat coils are adjusted, but not the zone setpoints.

**11. Were adjustments to CV and VAV system supply air temperature set points required to attain test case specified system supply air temperature?**

- Yes, consistent adjustment for all cases
- Yes, different adjustment for each case
- No
- Other, or additional discussion (please specify): \_\_\_\_\_

**12. How was the zero bypass factor idealization modeled in FC/SZ test cases?**

- Directly, using the program's standard input scheme
- Using custom supply air delivery temperature inputs for each test case
- Other, or additional discussion (*describe*): \_\_\_\_\_
- Was not successful

**13. How was the zero bypass factor idealization modeled in CV/VAV test cases?**

- Directly, using the program's standard input scheme
- Using custom supply air delivery temperature inputs for each test case
- Other, or additional discussion (*describe*): \_\_\_\_\_
- Was not successful

**14. Which Section 1.6.5.n.3 specified detailed output variables were reported, and at which specified locations?**

Temperatures:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:*\_\_\_\_\_

Humidity Ratios:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:*\_\_\_\_\_

Specific Volumes:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:*\_\_\_\_\_

Enthalpies:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:*\_\_\_\_\_

Mass Flows:  All locations reported  Some locations reported  Not reported

*Describe specified output locations not reported:*\_\_\_\_\_

**15. For reported detailed variables, how were outputs generated?**

Temperatures:  Direct from model  Required unit conversion  Other post processing\*

Humidity Ratios:  Direct from model  Required unit conversion  Other post processing\*

Specific Volumes:  Direct from model  Required unit conversion  Other post processing\*

Enthalpies:  Direct from model  Required unit conversion  Other post processing\*

Mass Flows:  Direct from model  Required unit conversion  Other post processing\*

\* “Other post processing” includes calculations outside of the model, and/or output that is inferred as equivalent or approximately equivalent from another location, e.g., for the SZ system return air fan inlet conditions and zone air conditions might be assumed to be equal or approximately so.

Describe any post-processing (outside-the-model) calculations and/or assumptions used to produce detailed outputs:\_\_\_\_\_

## **Appendix B. Modeling Notes for ASHRAE Standard 140**

*[Editor's note: Content here summarized from S140outNotes.txt form submitted during the simulation trials.]*

### **A. Software Information**

1. Vendor: [TESS, Madison, Wisconsin, United States](#)
2. Software Name: [TRNSYS with TESS Libraries](#)
3. Version (unique software version identifier): [17.01.0028](#)
4. Operating System Requirements: [Windows XP, Windows 7, Windows 8](#)
5. Approximate Hard Disk Space Required for Installation: [500 MB](#)
6. Minimum RAM Required for Software Operation: [8 GB](#)
7. Minimum Display Monitor Requirements: [SVGA with 1024x768 resolution and 256 colors](#)

### **B. Alternative Modeling Methods**

[NOTE 1 - Humidity Mode for Psychrometric Calculations \(Section 1.5.5 \[see Part I of this document\]\)](#)

#### 1.1 Simulated Effect:

[The properties used to calculate the air properties input to components.](#)

#### 1.2 Optional Settings or Modeling Capabilities

##### 1.2.1 Humidity Mode = 1

[Physical Meaning: The properties of the air input to a component model will be calculated using the Drybulb Temperature and the Absolute Humidity Ratio.](#)

##### 1.2.2 Humidity Mode = 2

[Physical Meaning: The properties of the air input to a component model will be calculated using the Drybulb Temperature and the Relative Humidity.](#)

#### 1.3 Setting or Capability Used: [Humidity Mode = 1](#)

[NOTE 2 - Cooling Coil Control Mode \(Section 1.5.5 \[see Part I of this document\]\)](#)

#### 2.1 Simulated Effect:

[The cooling coil model \(Type 752\) can control the leaving air conditions based on different air properties.](#)

#### 2.2 Optional Settings or Modeling Capabilities

##### 2.2.1 Control Mode = 1

[Physical Meaning: The coil is controlled based on the leaving air temperature.](#)

##### 2.2.2 Control Mode = 2

Physical Meaning: The coil is controlled based on the leaving air temperature and absolute humidity ratio.

#### 2.2.3 Control Mode = 3

Physical Meaning: The coil is controlled based on the leaving air temperature and relative humidity.

#### 2.2.4 Control Mode = 4

Physical Meaning: The coil is controlled based on the leaving air absolute humidity ratio.

2.3 Setting or Capability Used: [Control Mode = 1](#)

### **C. Equivalent Modeling Methods**

*[Editor's Note: No response submitted.]*

### **D. Non-Specified Inputs**

*[Editor's Note: No response submitted.]*

### **E. Omitted Test Cases and Results**

NONE

### **F. Changes to Source Code for the Purpose of Running the Tests, Where Such Changes are Not Available in Publicly Released Versions of the Software**

#### [NOTE 1 - Cooling Coil Latent and Condensate Loads](#)

1.1 Changes to the source code:

The TRNSYS cooling coil model (Type 752) was modified to calculate the latent load of the coil with the condensate temperature equal to the leaving air temperature rather than with zero enthalpy.

1.2 Test Cases Relevant to Changes in the Source Code:

[All cases in Section 1.5.5 \[see Part I of this document\].](#)

1.3 Explanation of Why the Change Is Not Included in the Publicly Released Version of the Software:

These are both valid assumptions for calculating the latent performance of a cooling coil.

#### [NOTE 2 - Fan Power Calculations](#)

2.1 Changes to the source code:

New TRNSYS fan models (Type 294 & 293) were written that calculate the fan power consumption based on the input pressure change across the fan. The usual method is to define the relationship between the control signal (or mass flowrate) and the power consumption.

## 2.2 Test Cases Relevant to the Changes in the Source Code:

All cases in Section 1.5.5 [see Part I of this document].

## 2.3 Explanation of Why the Change Is Not Included in the Publicly Released Version of the Software:

These are both valid assumptions for calculating the fan power.

### NOTE 3 - Zone Latent Loads

#### 3.1 Changes to the source code:

The TRNSYS load on an airstream component (Type 693) was changed to calculate the leaving air humidity ratio based on the enthalpy of the water vapor rather than the heat of vaporization of the water vapor.

#### 3.2 Test Cases Relevant to the Changes in the Source Code:

All cases in Section 1.5.5 [see Part I].

#### 3.3 Explanation of Why the Change Is Not Included in the Publicly Released Version of the Software:

These are both valid assumptions for calculating the latent loads in air.

### NOTE 4 - VAV Box Model

#### 4.1 Changes to the source code:

A new TRNSYS component model for VAV boxes (Type 4503) was written to model a VAV box that modulates the airflow and/or reheat to maintain a zone setpoint.

#### 4.2 Test Cases Relevant to the Changes in the Source Code:

All cases in Section 1.5.5.4 [see Part I]

#### 4.3 Explanation of Why the Change Is Not Included in the Publicly Released Version of the Software:

The model is still in the development stage.

## **G. Anomalous Results**

NONE

### 3.10 Quasi-Analytical Solution Pro-Forma Modeler Report

#### Pro-Forma Questionnaire for Airside HVAC Tests, July 2015

***Instructions:***

- *Replace empty boxes with symbols just below, as appropriate.*
  - Possible to use
  - Used to develop simulation trial results
- *Provide additional information and comments as appropriate; add lines as needed.*
- *If specific responses require disclosure of proprietary data not normally available to users in the program's engineering manual, leave the response blank, and include a note to the effect that including a response would require disclosure of proprietary information.*
- *If modeling varies among systems (and/or test cases), specify which systems (and/or test cases) are associated with each response; recopy a query as needed for this purpose (e.g., create 14a, 14b, etc.).*

**Program name, including version number**

Quasi-Analytical Solution (QAS), v2 10Oct2014

---

**Your name and organization**

Mike Kennedy, Mike D Kennedy Inc.

---

**1. Program status**

- Public domain
- Commercial
- Other (please specify): \_\_\_\_\_

**2. What weather inputs are used by the program for these tests?**

- Provided TMY2 weather data
- Directly entered DB and DP
- Directly entered DB and RH
- Directly entered DB and W
- Directly entered DB and WB

Note: DB = outdoor dry-bulb temperature; DP = outdoor dew-point temperature; RH = outdoor relative humidity; W = outdoor humidity ratio; WB = outdoor wet-bulb temperature.

**2a. If weather inputs provided from TMY2 data, which data are used for calculating outdoor air humidity ratio?**

- DB and DP
- DB and RH
- Other (please specify): \_\_\_\_\_

**3. How frequently is the HVAC system air density calculated for each system?**

- Once per run, same value applied throughout system
- Once every time step, same value applied throughout system
- Once every time step, varies by location throughout system
- Other (please specify): \_\_\_\_\_

**4. What air property basis is applied for HVAC system air density calculations for each system?**

- Single fixed air condition (please specify): \_\_\_\_\_
- Multiple fixed air conditions depending upon operating mode or system type (please specify): \_\_\_\_\_
- Air condition at each time step at a single specific system location \_\_\_\_\_ (please specify location): \_\_\_\_\_
- Local air conditions at each time step at each system location
- Other (please specify): \_\_\_\_\_

**5. What air pressure is used in HVAC system air density calculation?**

- Altitude adjusted standard pressure
- Weather file barometric pressure
- Other (please specify): pressure is input to program – standard pressure (101.325 kPa) is used for these runs

**6. Are fan/duct/system pressure differences accounted for in the HVAC air density calculations?**

- Yes
- No

-

**7. Are system models used for test cases the same as those commonly used by modelers to model similar systems?**

- Fan Coil:  Yes  No  
Single Zone:  Yes  No  
CV Reheat:  Yes  No  
VAV:  Yes  No

Other (please specify): This question is less appropriate to the context of the QAS.

**8. How are zone loads defined?**

- Zone loads specified directly from input tables (e.g., Part I, Table 1-2)  
 Using alternative zone definition (e.g., Part I, Section 1.5.5.1.1.5)

**9. Thermostat Control**

- Perfect control  
 On/Off thermostatic control  
 On/Off thermostatic control with deadband  
 Proportional control  
 Other (please specify): \_\_\_\_\_

**10. Were adjustments to zone temperature set points required to attain test case specified zone temperature?**

- Yes, consistent adjustment for all cases  
 Yes, different adjustment for each case  
 No  
 Other, or additional discussion (please specify): \_\_\_\_\_

**11. Were adjustments to CV and VAV system supply air temperature set points required to attain test case specified system supply air temperature?**

- Yes, consistent adjustment for all cases  
 Yes, different adjustment for each case  
 No  
 Other, or additional discussion (please specify): \_\_\_\_\_

**12. How was the zero bypass factor idealization modeled in FC/SZ test cases?**

- Directly, using the program's standard input scheme  
 Using custom supply air delivery temperature inputs for each test case  
 Other, or additional discussion (*describe*): QAS assumes zero bypass. No other value possible.  
 Was not successful

**13. How was the zero bypass factor idealization modeled in CV/VAV test cases?**

- Directly, using the program's standard input scheme  
 Using custom supply air delivery temperature inputs for each test case  
 Other, or additional discussion (*describe*): QAS assumes zero bypass. No other value possible.  
 Was not successful



**14. Which Section 1.6.5.n.3 specified detailed output variables were reported, and at which specified locations?**

Temperatures:      All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:\_\_\_\_\_*

Humidity Ratios:    All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:\_\_\_\_\_*

Specific Volumes:    All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:\_\_\_\_\_*

Enthalpies:          All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:\_\_\_\_\_*

Mass Flows:          All locations reported    Some locations reported    Not reported  
*Describe specified output locations not reported:\_\_\_\_\_*

**15. For reported detailed variables, how were outputs generated?**

Temperatures:      Direct from model    Required unit conversion    Other post processing\*  
Humidity Ratios:    Direct from model    Required unit conversion    Other post processing\*  
Specific Volumes:    Direct from model    Required unit conversion    Other post processing\*  
Enthalpies:          Direct from model    Required unit conversion    Other post processing\*  
Mass Flows:          Direct from model    Required unit conversion    Other post processing\*

\* “Other post processing” includes calculations outside of the model, and/or output that is inferred as equivalent or approximately equivalent from another location, e.g., for the SZ system return air fan inlet conditions and zone air conditions might be assumed to be equal or approximately so.

Describe any post-processing (outside-the-model) calculations and/or assumptions used to produce detailed outputs:\_\_\_\_\_

## 4.0 Part IV: Simulation Field Trial Results

### 4.1 Introduction

Here we present the results for the simulation trials of cases AE101 – AE445. These are results after numerous iterations to incorporate clarifications to the test specification, simulation input corrections, and simulation software improvements. Where improvements to simulation programs or simulation inputs were made as a result of running the tests, such improvements must have mathematical and physical bases and must be applied consistently across tests. Also, all improvements were required to be documented in modeler reports (Part III, Section 3.9). Arbitrary modification of a simulation program’s input or internal code just for the purpose of more closely matching a given set of results is not allowed.

Table 4-1 summarizes the following information for the eight models, including the quasi-analytical solution, which were implemented by the seven organizations that participated in this project: model-authoring organization, model testing organization (“Implemented by”), and abbreviation labels used in the results graphs and tables.

Fan coil and single zone systems results are presented in Section 4.3, and constant volume and variable air volume systems results are presented in Section 4.4. These sections each present graphs of the results first, followed by tables of the results. Quasi-analytical solution results are the leftmost bar in the bar charts. The x-axis labels of the charts include abbreviated descriptions of the test cases.

Definitions of the abbreviations and acronyms used in charts and tables are provided in Section 4.2. Case descriptions are summarized in Part I, Appendix B, Tables B-1 through B-8.

An electronic version of these results is included with accompanying files Results-FCSZ.xlsm, and Results-CVVV.xlsm (see subfolder “PartIV-Files”), with navigation instructions included in the “Read Me” tab of each workbook file.

**Table 4-1 Airside HVAC Cases  
Participating Organizations and Models**

<b>Model</b>	<b>Authoring Organization</b>	<b>Implemented by</b>	<b>Abbreviation</b>
Quasi-Analytical Solution (QAS)	PSU <sup>a</sup> /UNO <sup>b</sup> /TAMU <sup>c</sup> /NREL <sup>d</sup> / JNA <sup>e</sup> /MDK <sup>f</sup> , United States	NREL <sup>d</sup> /JNA <sup>e</sup> /MDK <sup>f</sup> , United States	QAS/PSU-TAMU-NREL
DEEAP <sup>g</sup> 1.1.2	AAON, Inc., United States	AAON, Inc., United States	DEEAP/AAON
DeST <sup>h</sup> 2	Tsinghua University, China	Tsinghua University, China / LBNL <sup>i</sup> , United States	DeST/TsinghuaU-LBNL
DOE-2.2 V48L	JJH <sup>j</sup> /LBNL <sup>i</sup> /UC <sup>k</sup> , United States	NREL <sup>d</sup> /JNA <sup>e</sup> /MDK <sup>f</sup> , United States	DOE-2.2/NREL
EnergyPlus 8.2.0	DOE-BT <sup>l</sup> , United States	GARD Analytics, Inc., United States	EnergyPlus/GARD
IES-VE <sup>m</sup> 2014.2	IES <sup>n</sup> , United Kingdom	IES <sup>n</sup> , United Kingdom	IES-VE/IES
LCEM <sup>o</sup> 3.10	MLIT <sup>p</sup> , Japan	TTE <sup>q</sup> , Japan	LCEM/MLIT-TTE
TRNSYS 17.01.0028	TESS <sup>r</sup> /UWM <sup>s</sup> , United States	TESS <sup>r</sup> , United States	TRNSYS/TESS

<sup>a</sup> PSU: The Pennsylvania State University, United States

<sup>b</sup> UNO: University of Nebraska - Omaha, United States

<sup>c</sup> TAMU: Texas A&M University, United States

<sup>d</sup> NREL: National Renewable Energy Laboratory, United States

<sup>e</sup> J. Neymark & Associates, United States

<sup>f</sup> Mike D. Kennedy, Inc., United States

<sup>g</sup> DEEAP: Detailed Energy and Economic Analysis Program

<sup>h</sup> DeST: Designer's Simulation Toolkit

<sup>i</sup> LBNL: Lawrence Berkeley National Laboratory, United States

<sup>j</sup> JJH: James J. Hirsch & Associates, United States

<sup>k</sup> UC: University of California, United States

<sup>l</sup> DOE-BT: U.S. Department of Energy, Office of Building Technologies, Energy Efficiency and Renewable Energy, United States

<sup>m</sup> IES-VE: Integrated Environmental Solutions - Virtual Environment

<sup>n</sup> IES: Integrated Environmental Solutions, United Kingdom

<sup>o</sup> LCEM: Life Cycle Energy Management tool

<sup>p</sup> MLIT: Ministry of Land, Infrastructure, Transportation and Tourism, Japan

<sup>q</sup> TTE: Takasago Thermal Engineering, Japan

<sup>r</sup> TESS: Thermal Energy System Specialists, United States

<sup>s</sup> UWM: University of Wisconsin – Madison, United States

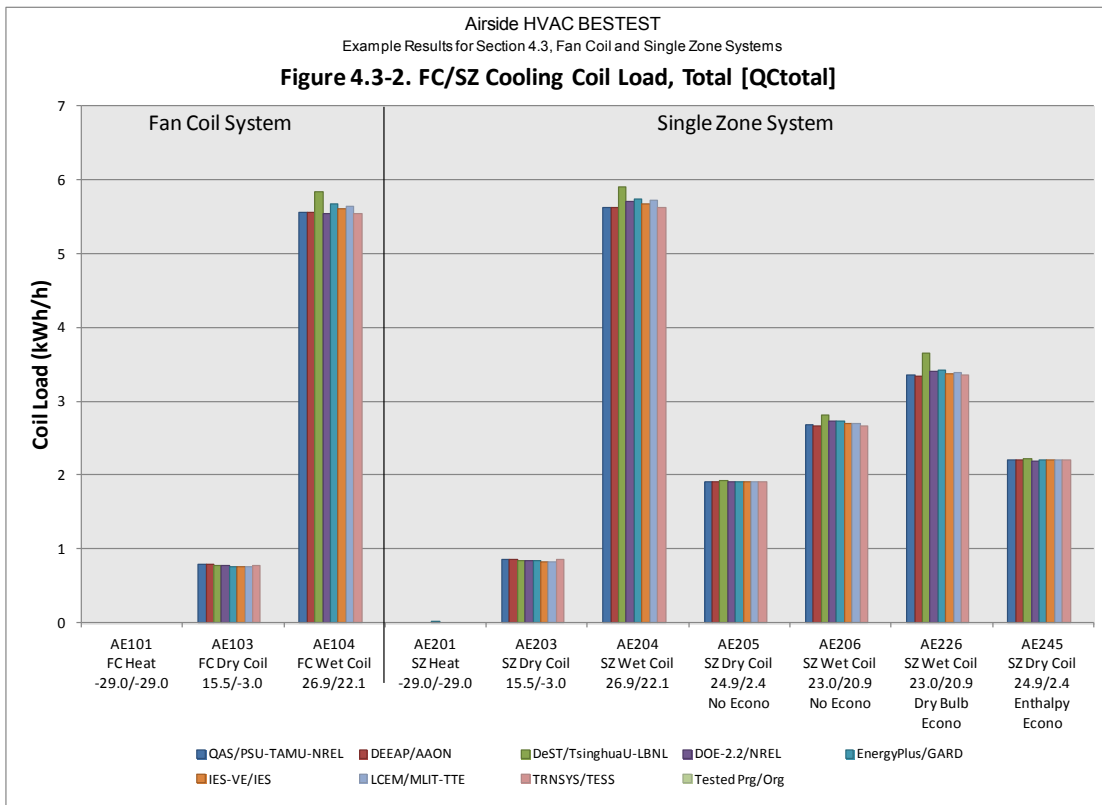
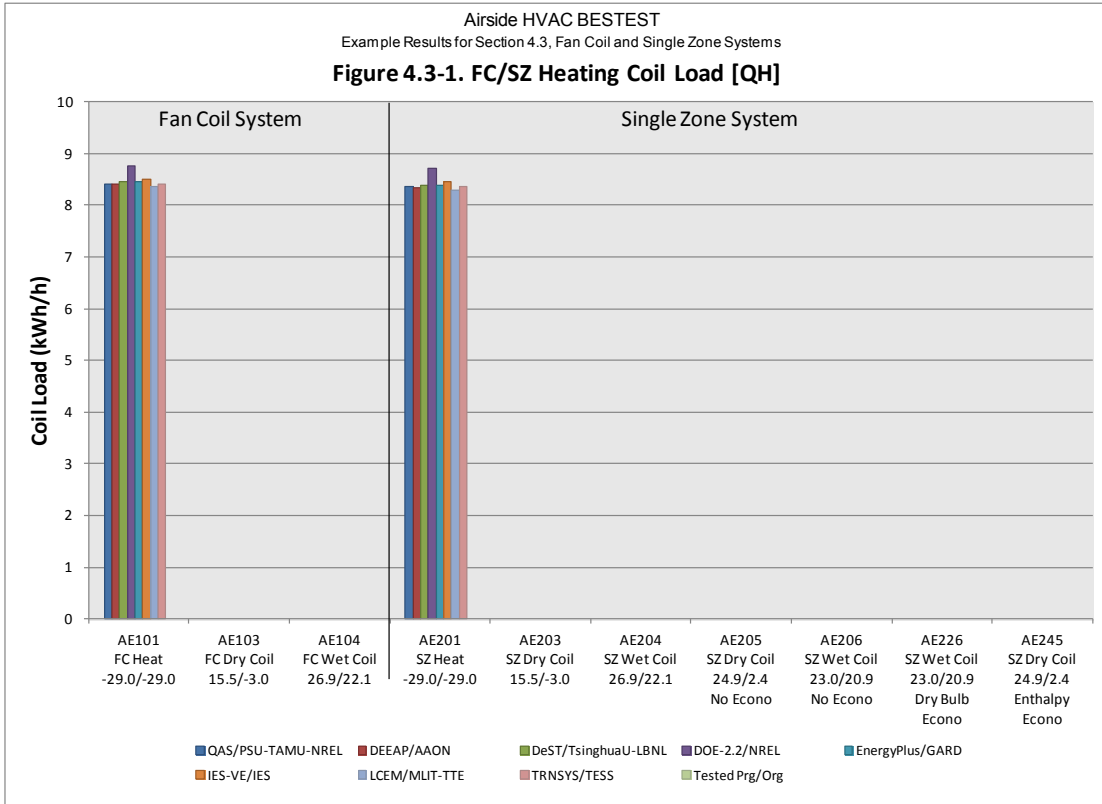
## 4.2 Abbreviations and Acronyms in Results Charts and Tables

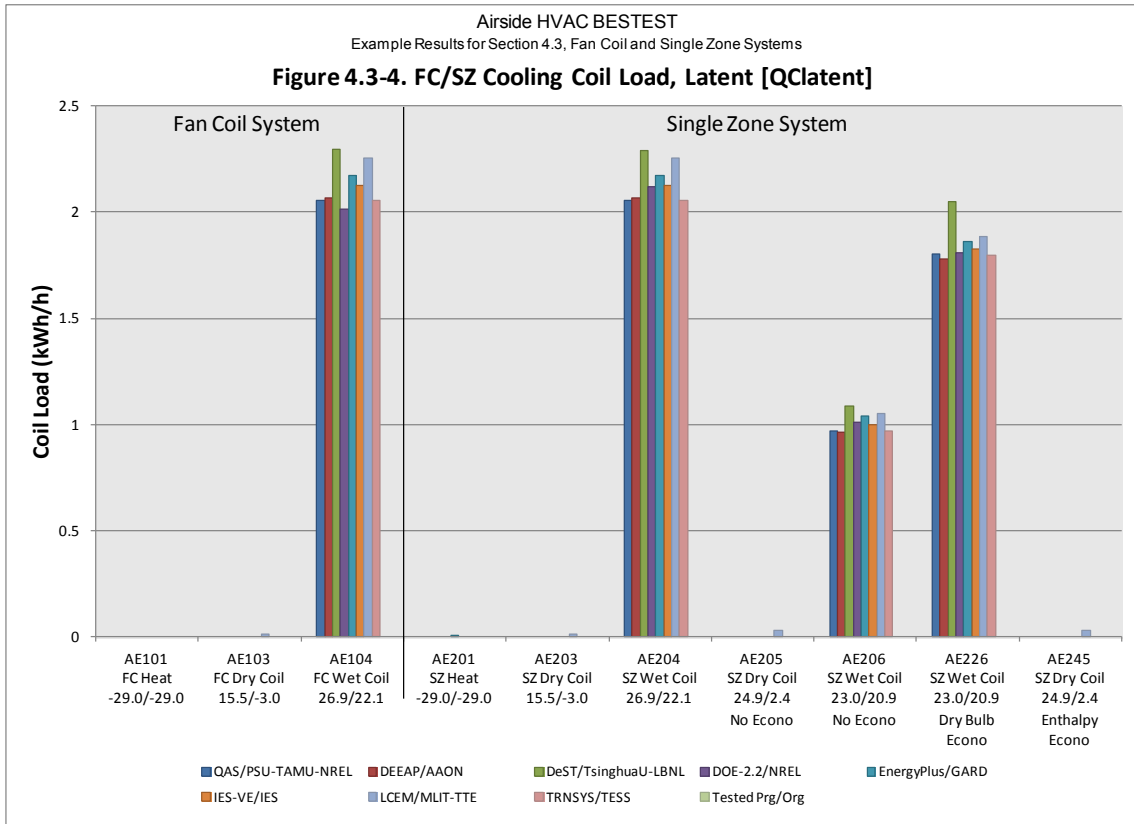
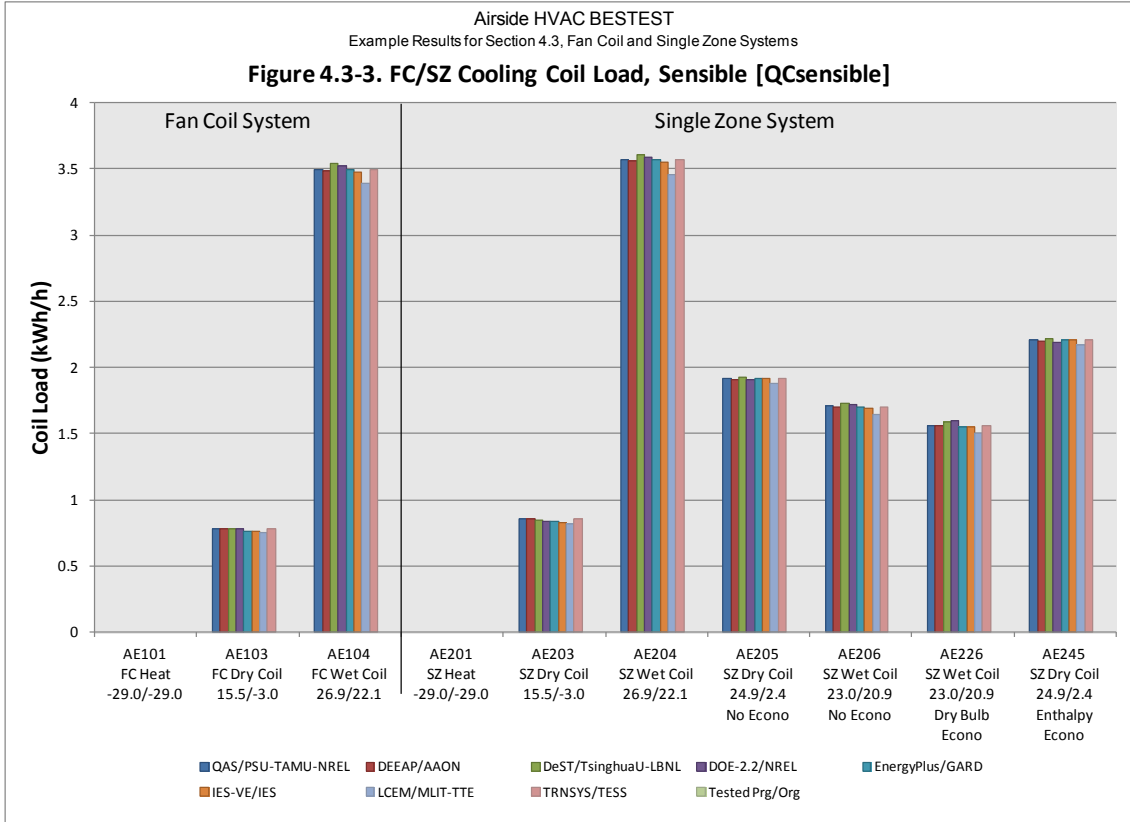
Abbreviations for example results identifiers used in the chart legends and tables are in listed in Table 4-1; additional abbreviations used in the charts and tables are listed below. Also see Part I, Section 1.5.3.2.

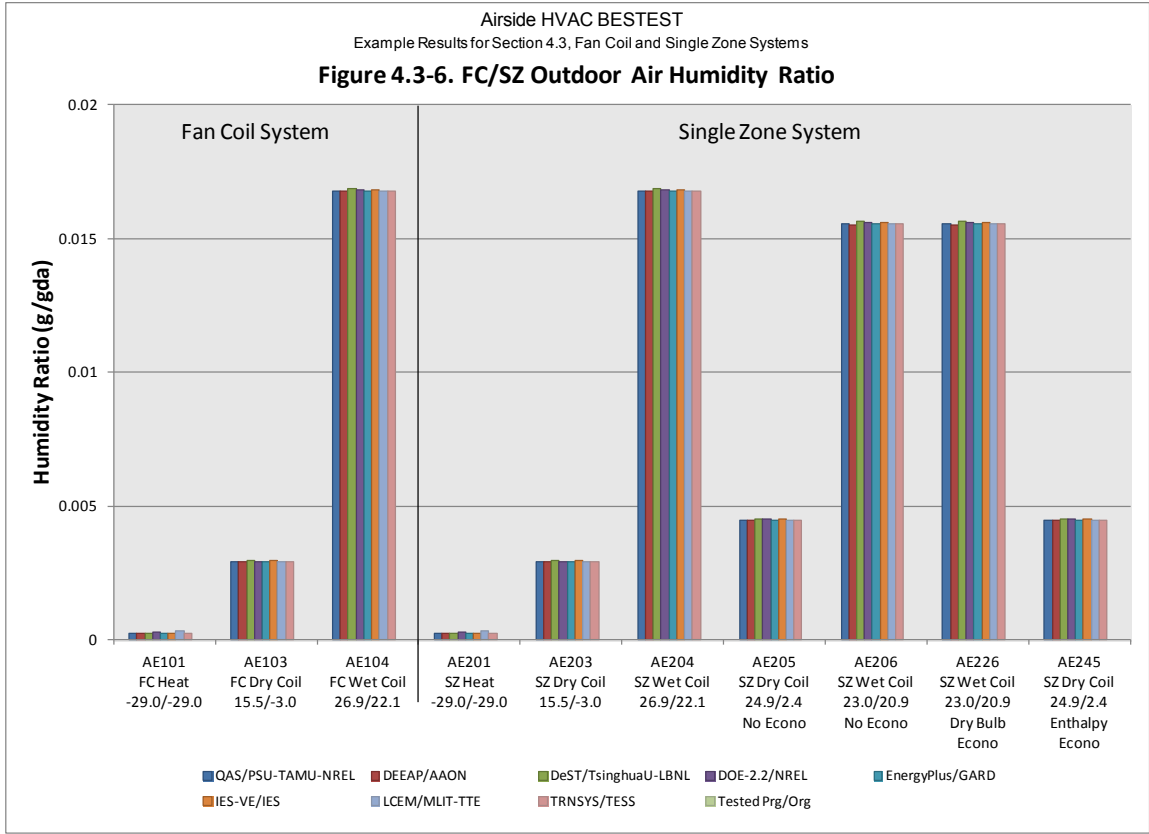
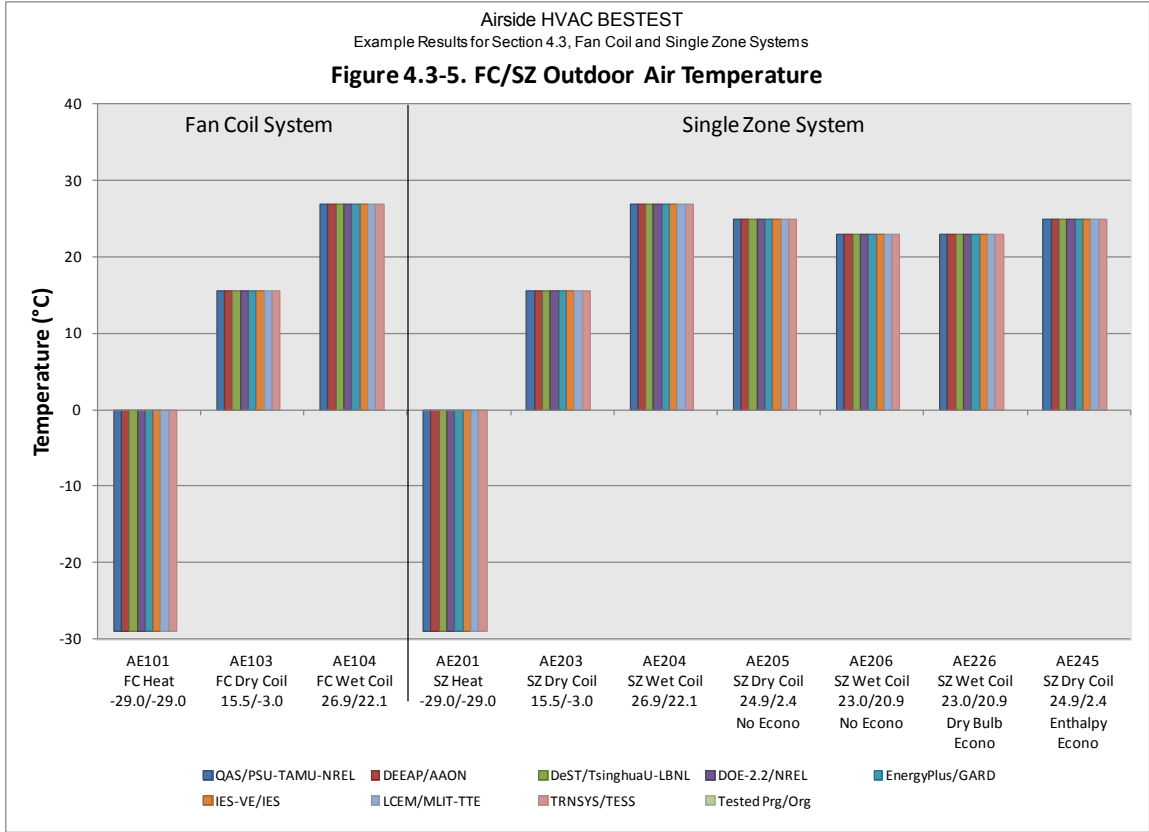
CV:	constant volume terminal reheat system, see Part I, Section 1.5.5.3
DB econo:	economizer with return air comparative dry-bulb temperature control
Econo:	economizer
Enth. Eco:	economizer with return air comparative enthalpy control
FC:	fan coil system, see Part I, Section 1.5.5.1
gda:	grams dry air
kgda:	kilograms dry air
SZ:	single zone system, see Part I, Section 1.5.5.2
VAV:	variable air volume system, see Part I, Section 1.5.5.4

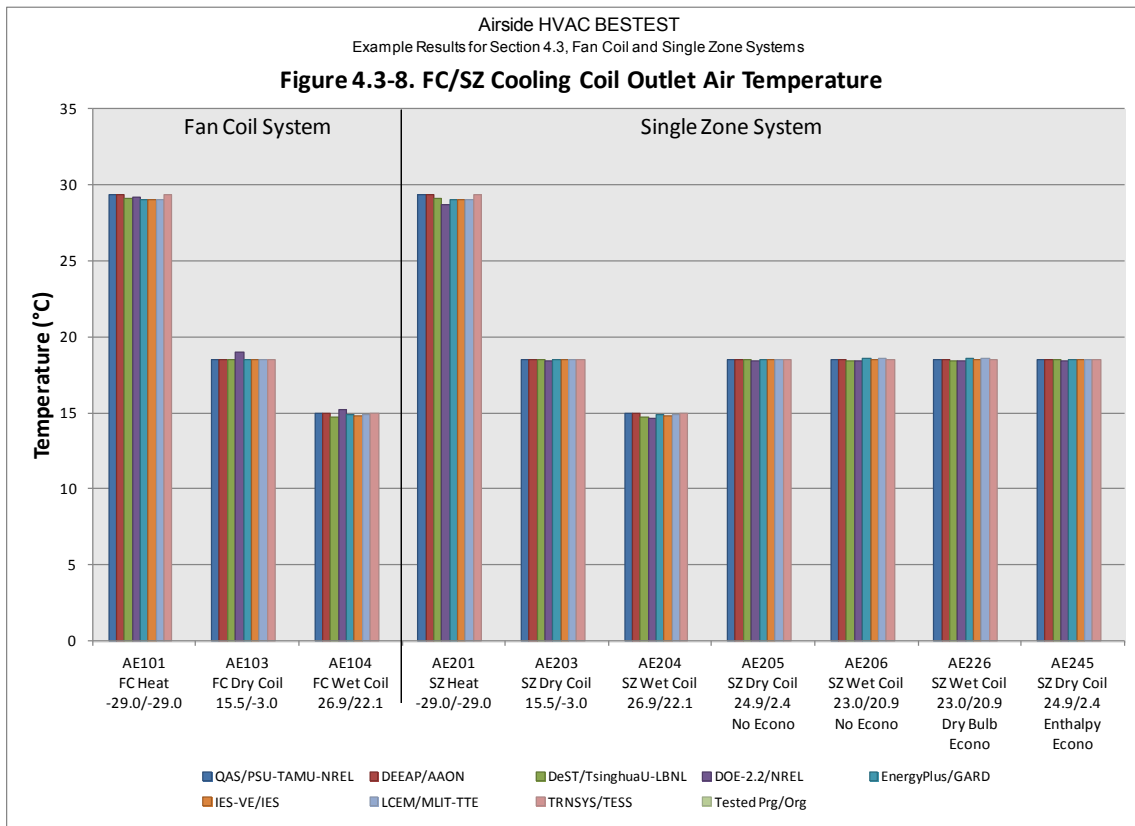
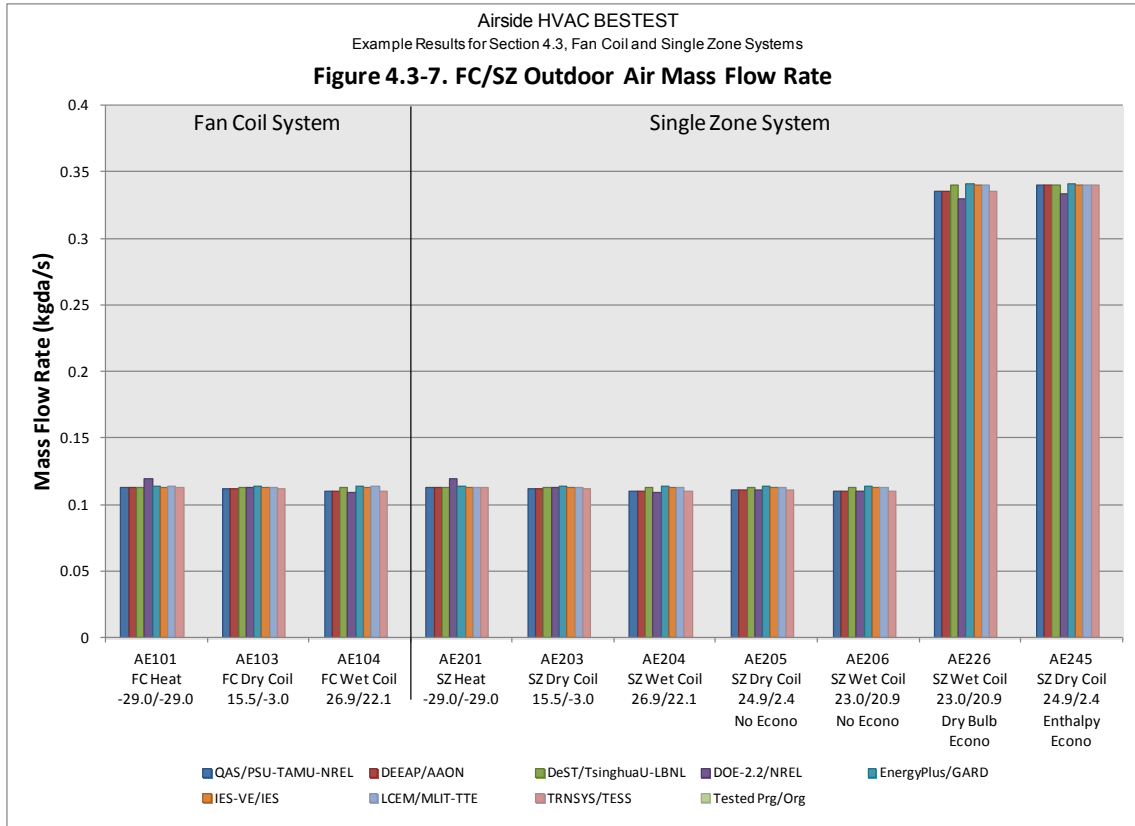
Numeric values listed in the chart x-axis labels as “value/value” (e.g., “29.4/2.4”) are outdoor dry-bulb and outdoor dew-point temperatures, respectively.

### 4.3 Results Graphs and Tables for Fan Coil and Single Zone Systems

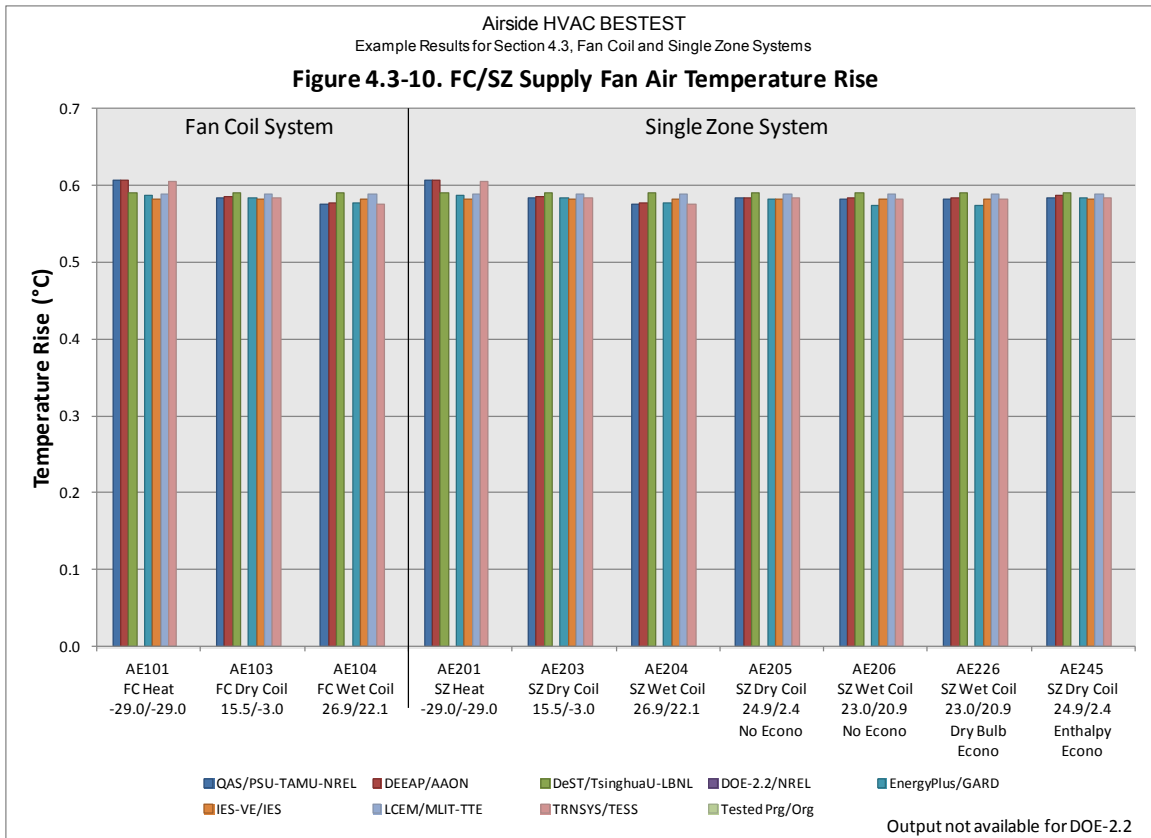
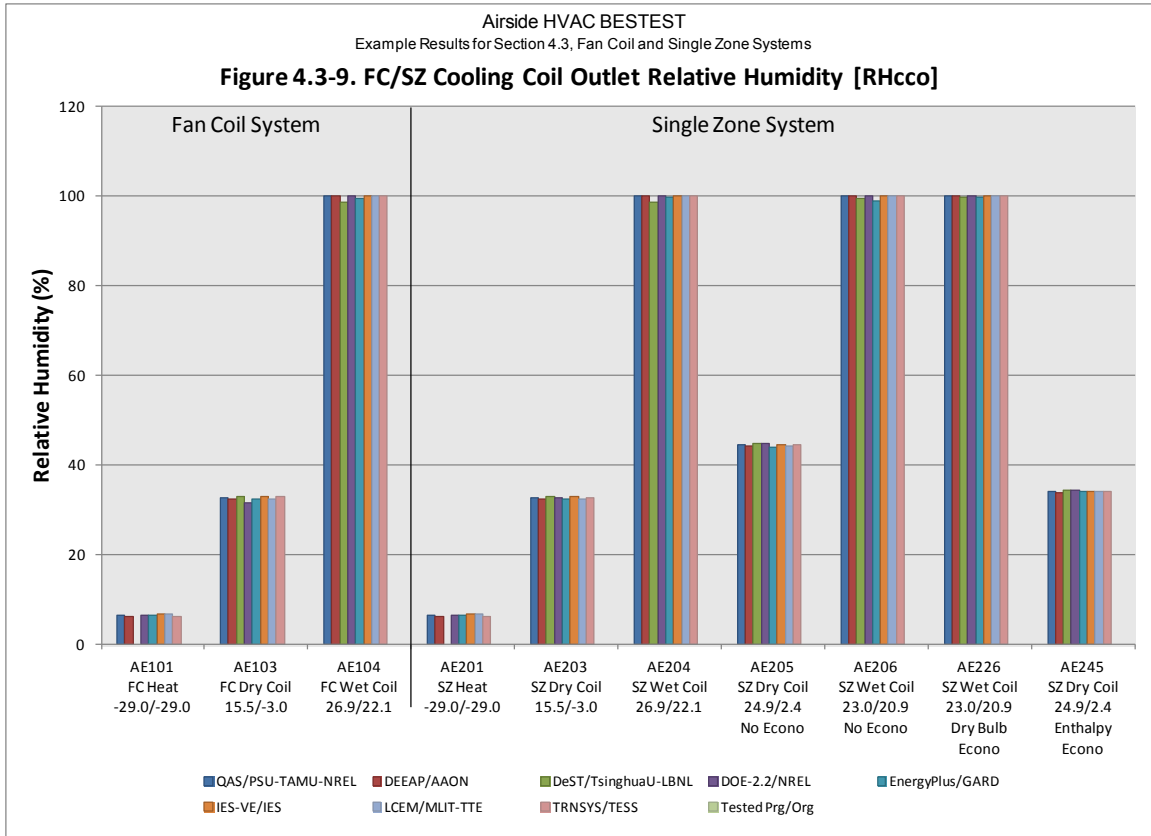


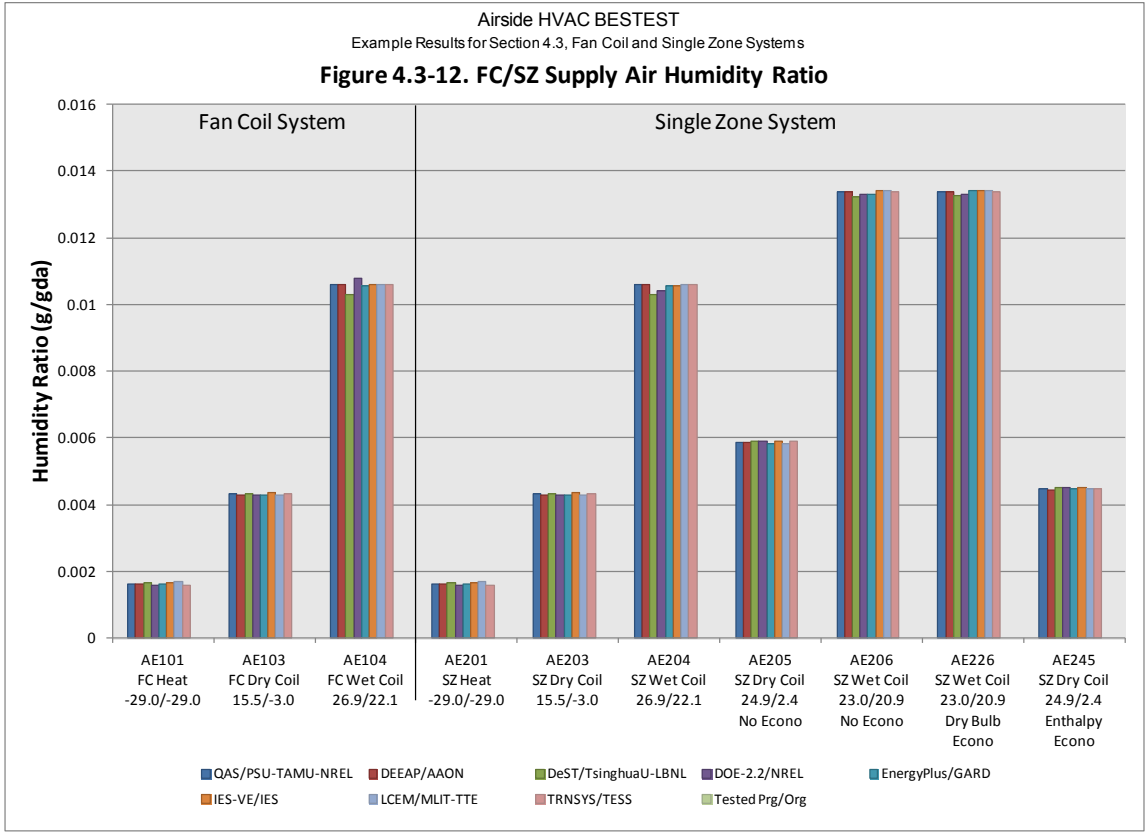
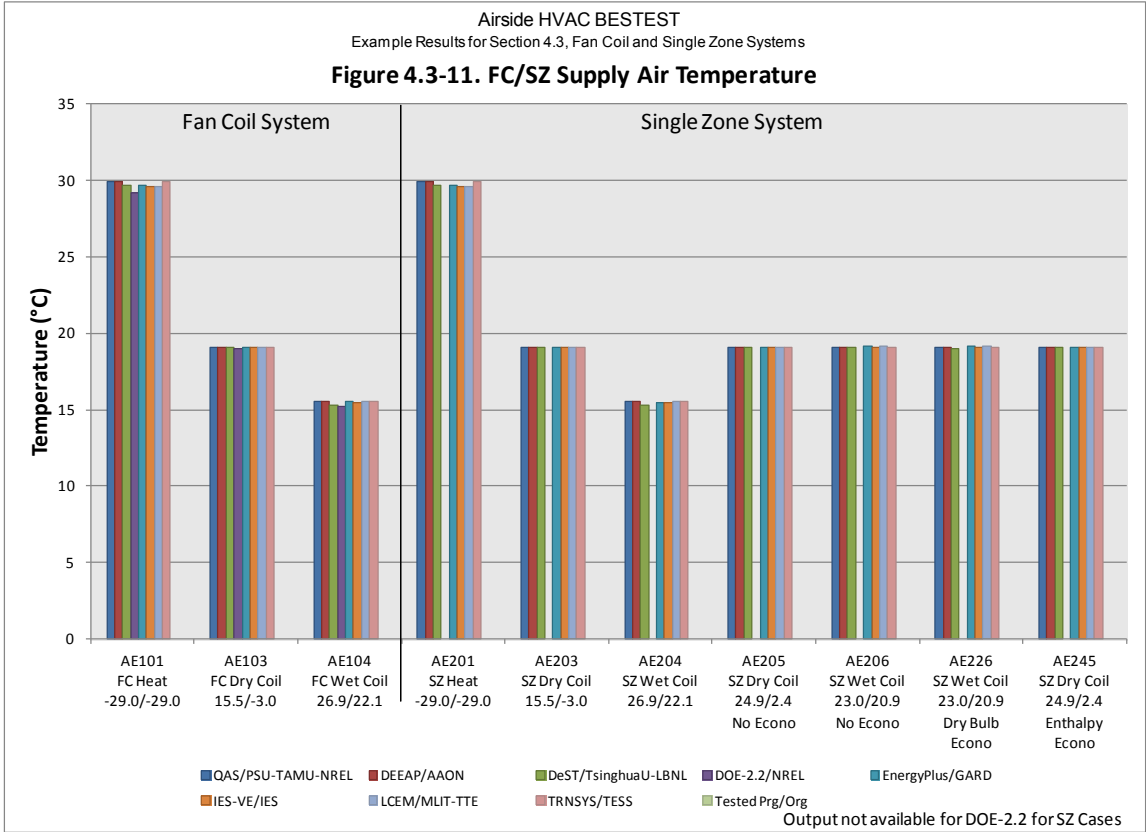


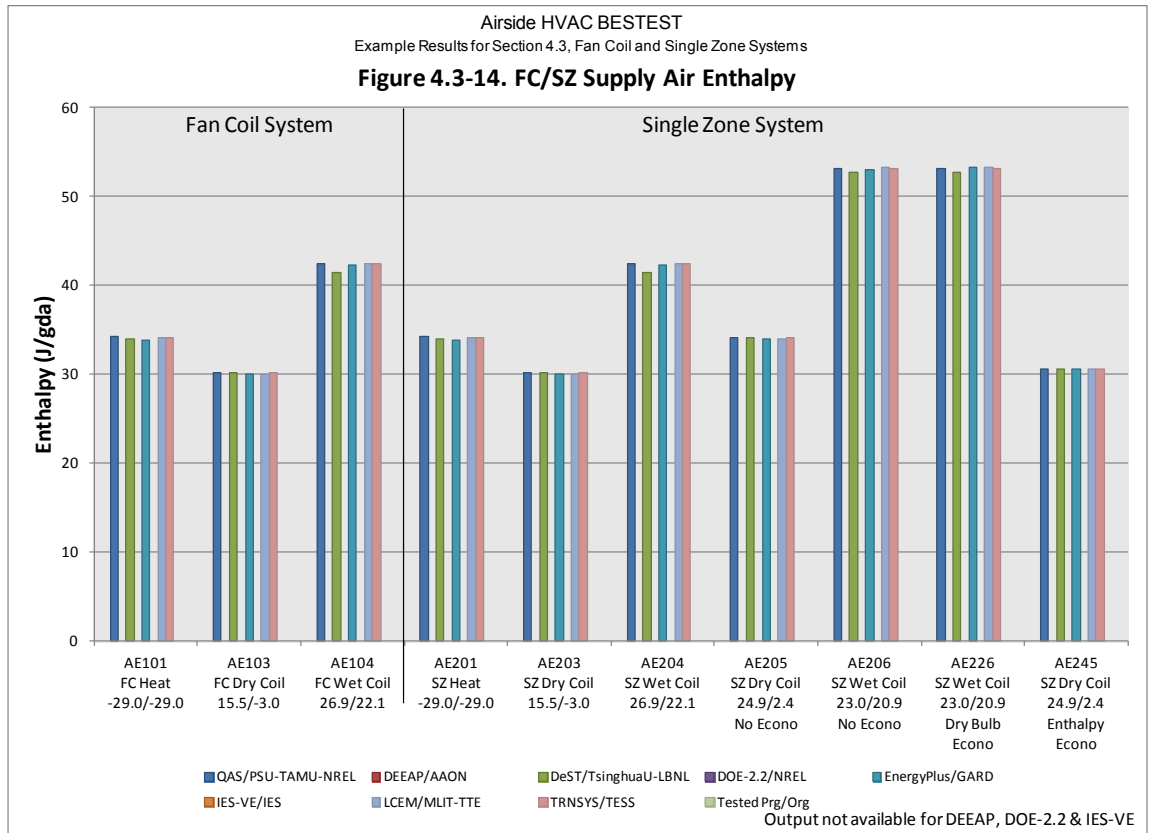
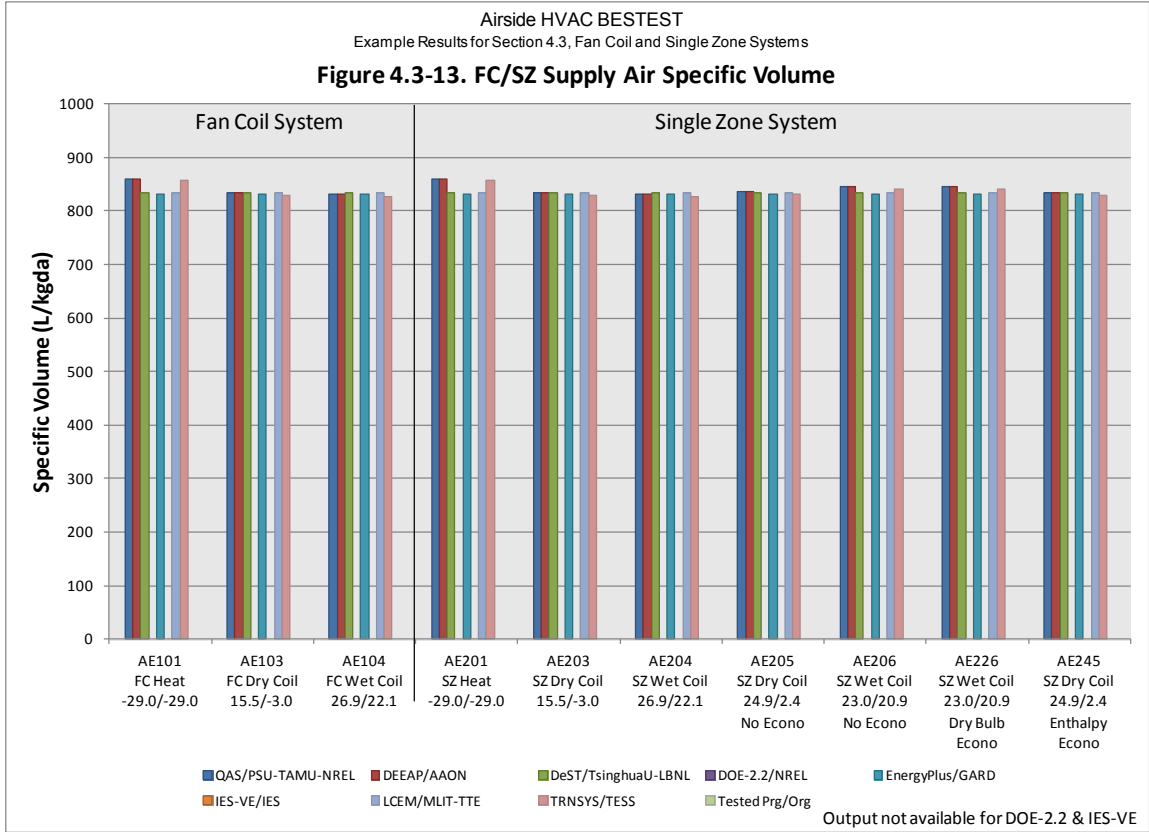


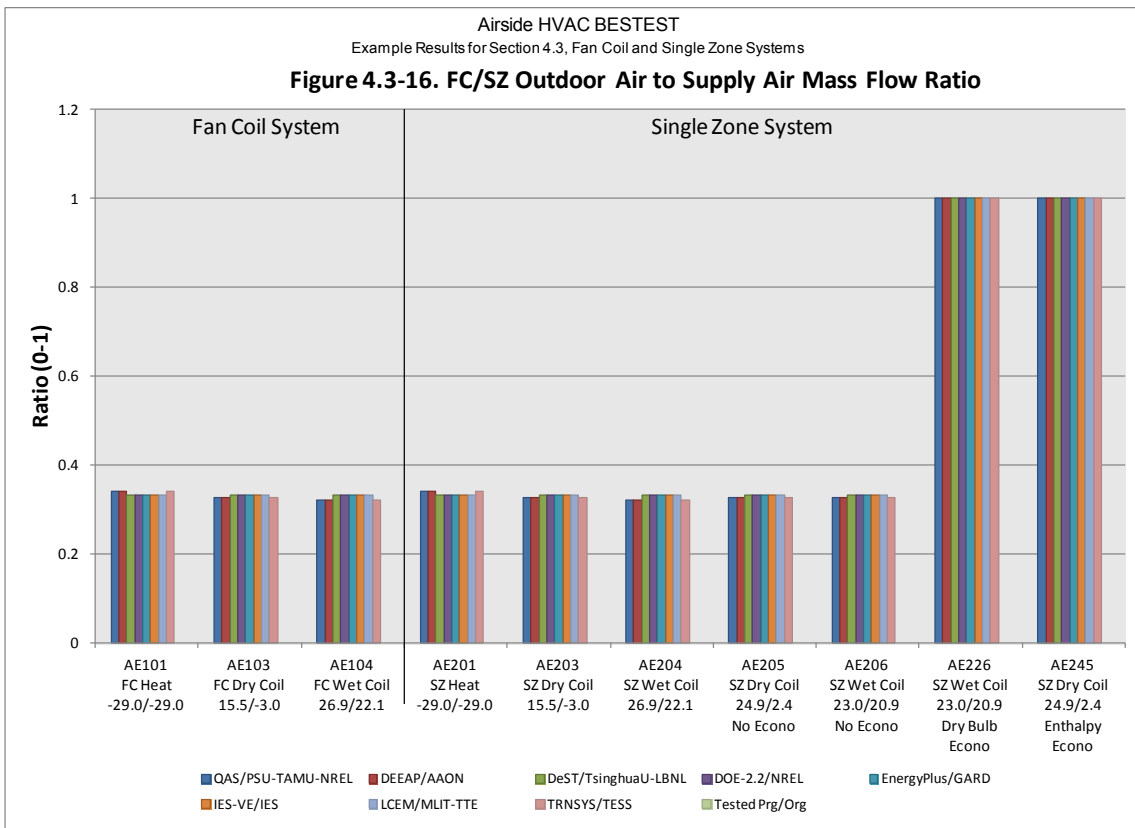
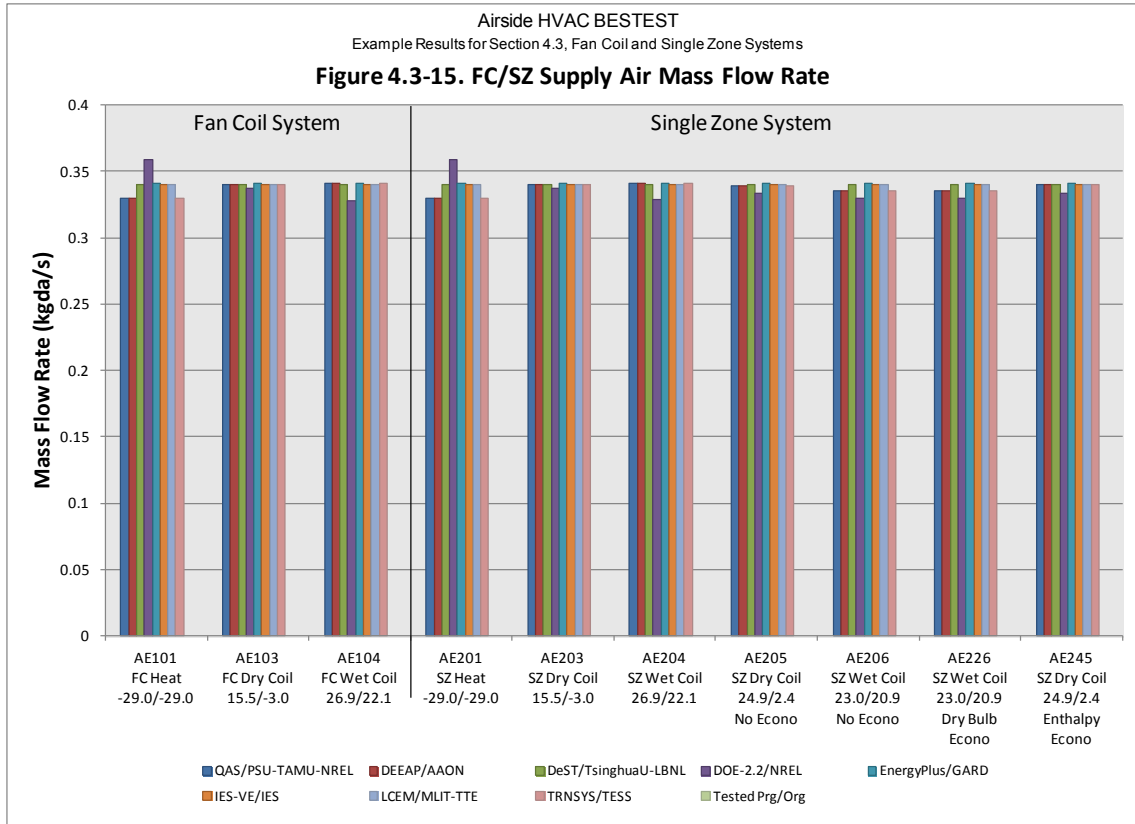


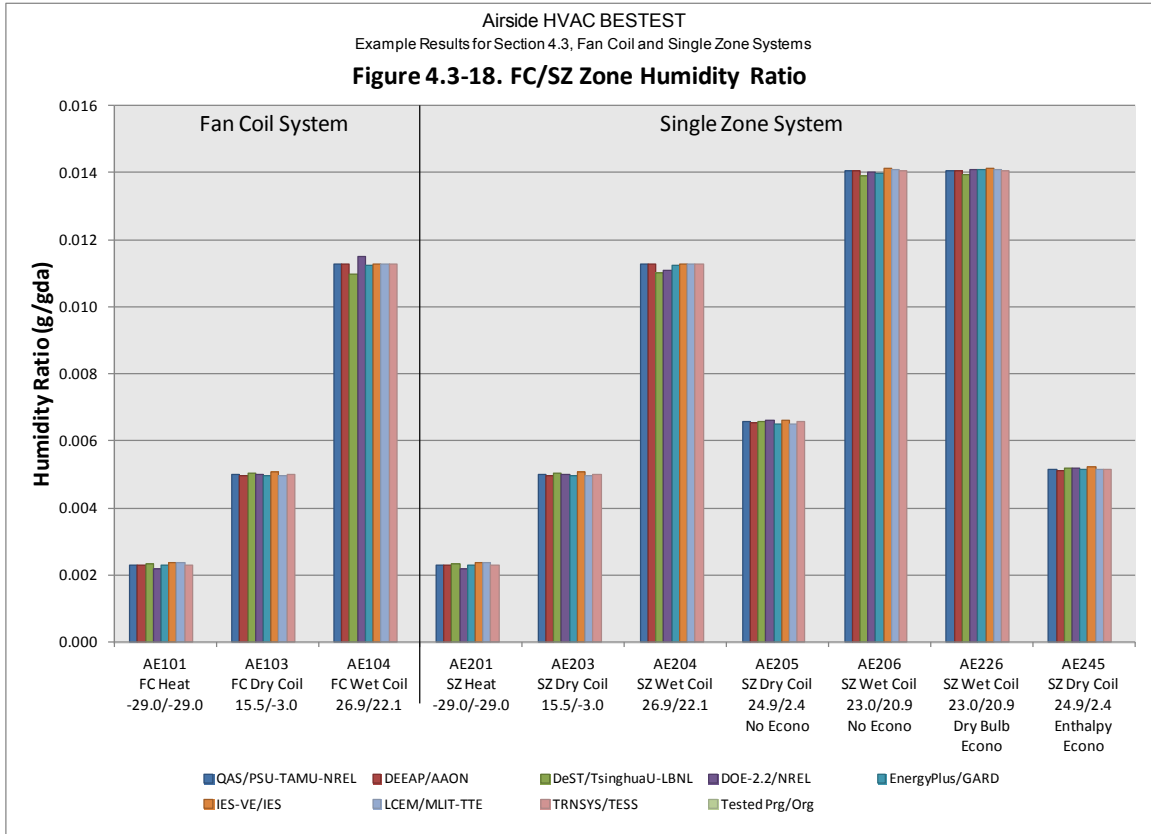
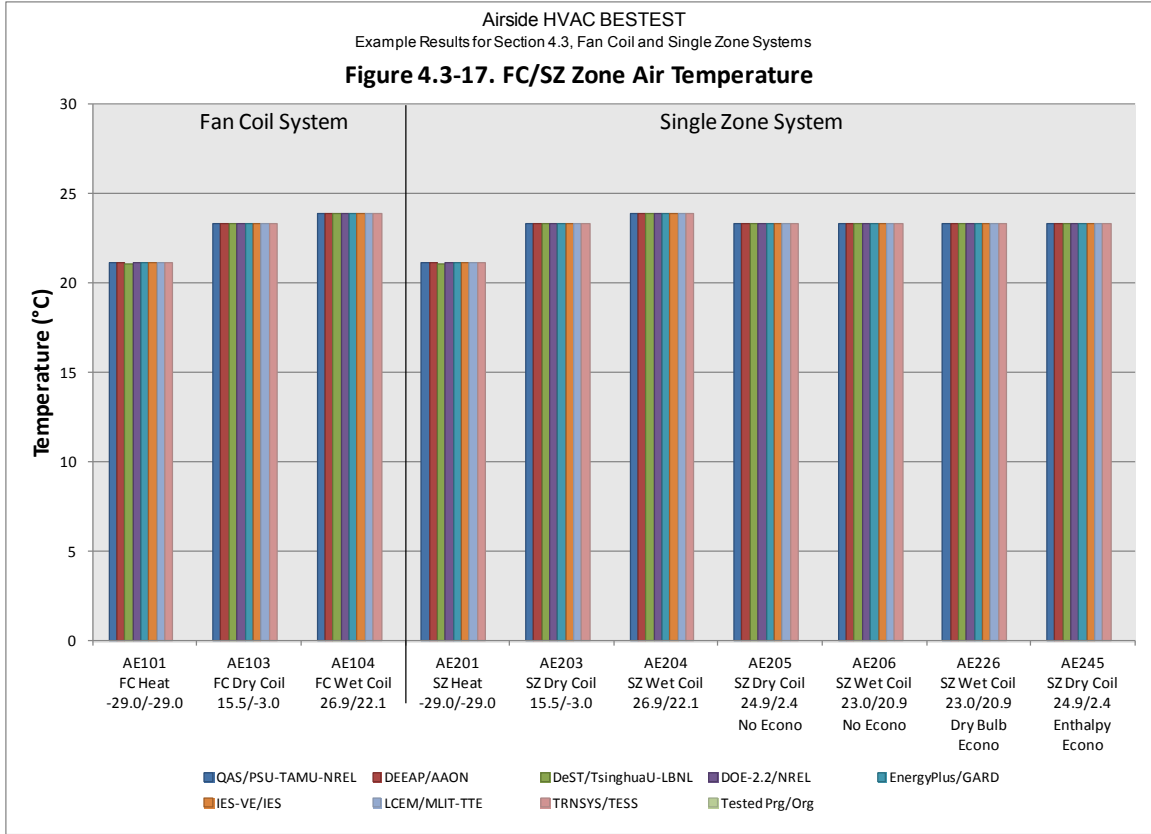


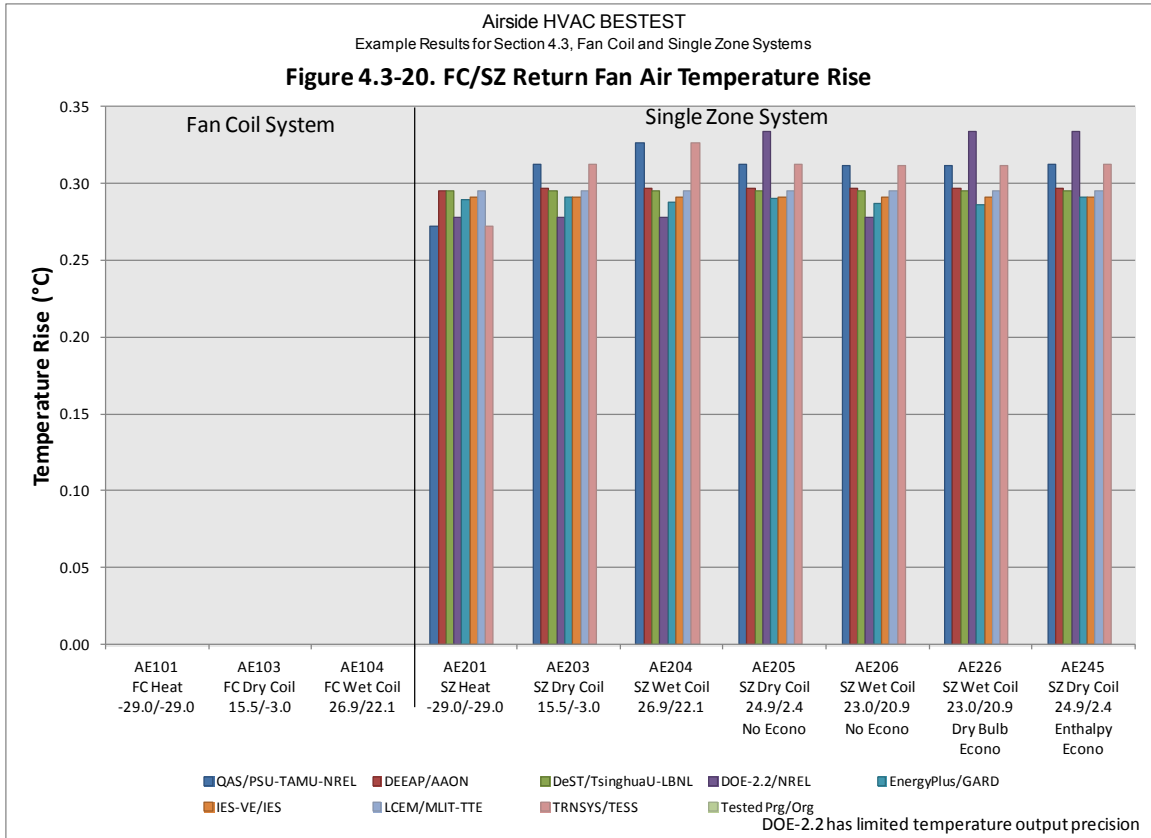
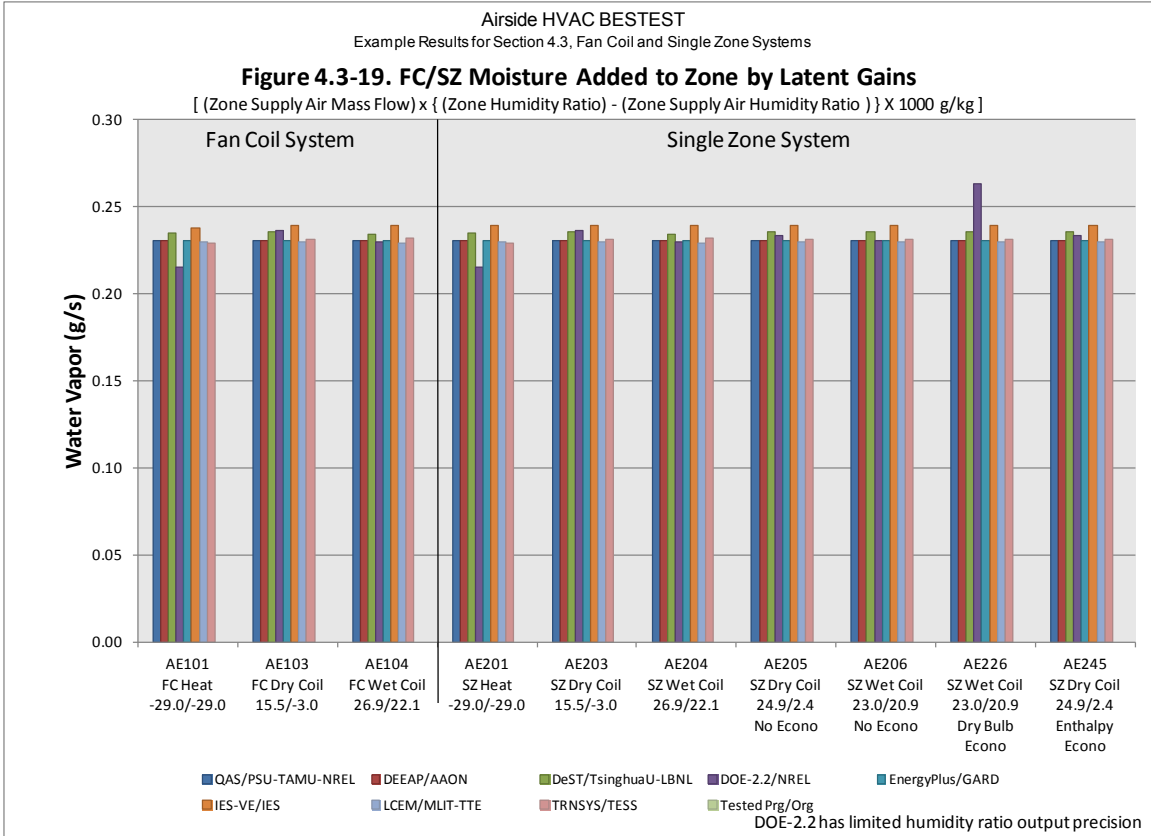


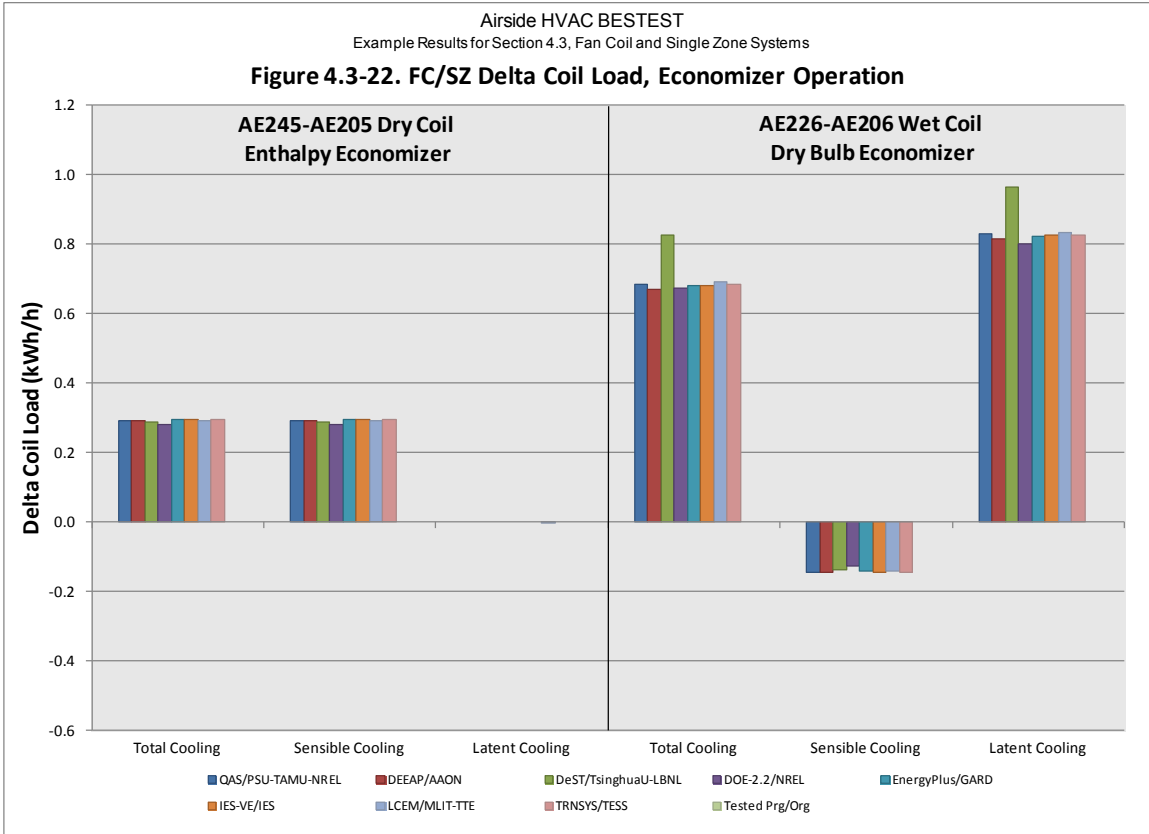
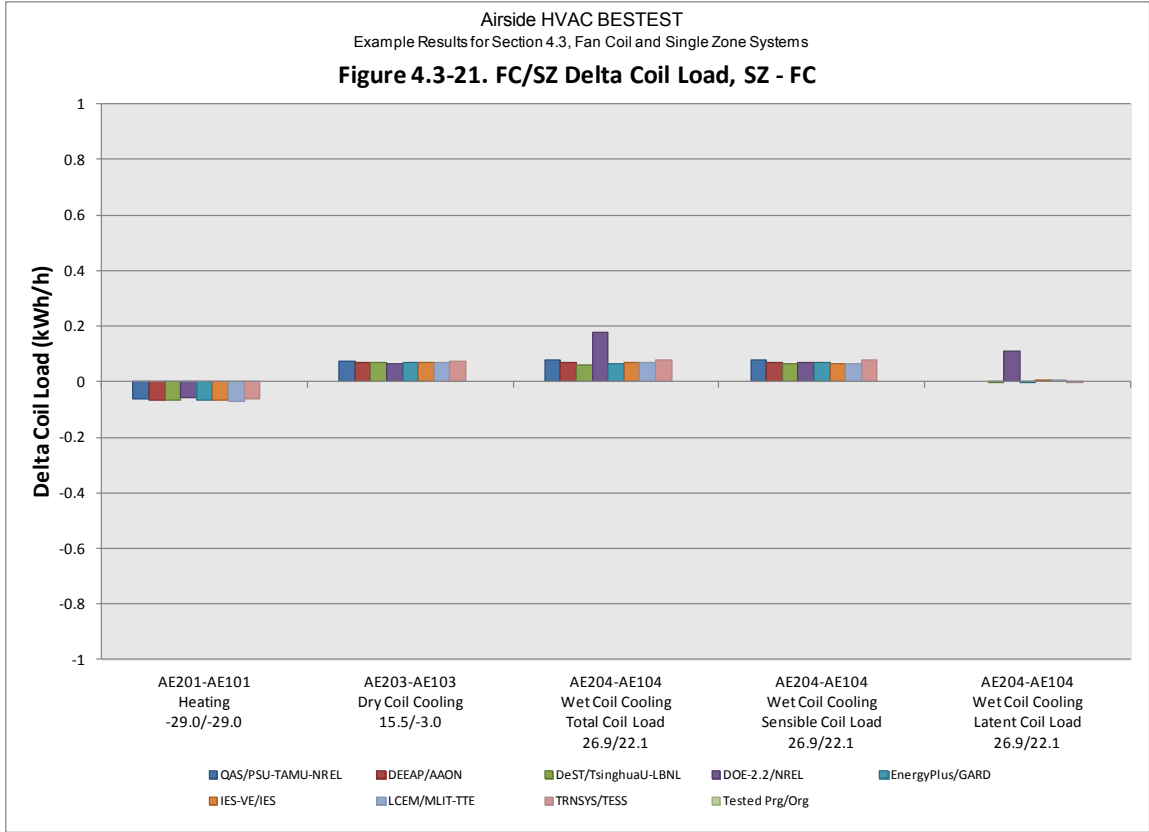












**Table 4.3-1. FC/SZ Heating Coil Load [QH] (kWh/h)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	8.420	8.413	8.452	8.768	8.453	8.513	8.372	8.416	
AE103	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE104	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE201	8.360	8.349	8.385	8.709	8.387	8.446	8.302	8.356	
AE203	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE204	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE205	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE206	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE226	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE245	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	

**Table 4.3-2. FC/SZ Total Cooling Coil Load [Qctotal] (kW)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE103	0.783	0.784	0.778	0.777	0.766	0.762	0.763	0.780	
AE104	5.551	5.553	5.836	5.533	5.673	5.607	5.646	5.548	
AE201	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE203	0.856	0.853	0.845	0.840	0.833	0.830	0.830	0.852	
AE204	5.628	5.623	5.897	5.711	5.739	5.675	5.713	5.625	
AE205	1.916	1.910	1.929	1.908	1.913	1.915	1.911	1.913	
AE206	2.677	2.664	2.818	2.731	2.738	2.696	2.699	2.671	
AE226	3.362	3.335	3.643	3.405	3.418	3.377	3.391	3.355	
AE245	2.208	2.202	2.216	2.188	2.207	2.210	2.202	2.209	

**Table 4.3-3. FC/SZ Sensible Cooling Coil Load [QCsensible] (kWh/h)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE103	0.783	0.784	0.778	0.777	0.766	0.762	0.752	0.780	
AE104	3.497	3.488	3.543	3.521	3.499	3.480	3.389	3.493	
AE201	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE203	0.856	0.853	0.845	0.840	0.833	0.830	0.819	0.852	
AE204	3.574	3.559	3.608	3.590	3.566	3.547	3.456	3.571	
AE205	1.916	1.910	1.929	1.908	1.913	1.915	1.880	1.913	
AE206	1.706	1.701	1.731	1.723	1.697	1.695	1.646	1.703	
AE226	1.562	1.557	1.593	1.597	1.555	1.552	1.503	1.560	
AE245	2.208	2.202	2.216	2.188	2.207	2.210	2.171	2.209	

**Table 4.3-4. FC/SZ Latent Cooling Coil Load [QClatent] (kWh/h)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE103	0.000	0.000	0.000	0.000	0.000	0.000	0.010	0.000	
AE104	2.054	2.065	2.293	2.012	2.174	2.127	2.256	2.054	
AE201	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
AE203	0.000	0.000	0.000	0.000	0.000	0.000	0.011	0.000	
AE204	2.054	2.065	2.289	2.121	2.173	2.128	2.257	2.054	
AE205	0.000	0.000	0.000	0.000	0.000	0.000	0.031	0.000	
AE206	0.970	0.963	1.087	1.008	1.042	1.000	1.053	0.968	
AE226	1.800	1.778	2.050	1.808	1.863	1.825	1.888	1.795	
AE245	0.000	0.000	0.000	0.000	0.000	0.000	0.031	0.000	



**Table 4.3-5. FC/SZ Outdoor Air Temp (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	-29.000	-29.000	-29.000	-29.000	-29.000	-29.000	-29.000	-29.000	
AE103	15.500	15.500	15.500	15.500	15.500	15.500	15.500	15.500	
AE104	26.900	26.889	26.900	26.889	26.900	26.900	26.900	26.900	
AE201	-29.000	-29.000	-29.000	-29.000	-29.000	-29.000	-29.000	-29.000	
AE203	15.500	15.500	15.500	15.500	15.500	15.500	15.500	15.500	
AE204	26.900	26.889	26.900	26.889	26.900	26.900	26.900	26.900	
AE205	24.900	24.889	24.900	24.889	24.900	24.900	24.900	24.900	
AE206	23.000	23.000	23.000	23.000	23.000	23.000	23.000	23.000	
AE226	23.000	23.000	23.000	23.000	23.000	23.000	23.000	23.000	
AE245	24.900	24.889	24.900	24.889	24.900	24.900	24.900	24.900	

**Table 4.3-6. FC/SZ Outdoor Air Humidity Ratio (%)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	0.000259	0.000259	0.000260	0.000300	0.000259	0.000262	0.000344	0.000259	
AE103	0.002936	0.002895	0.002948	0.002900	0.002933	0.002947	0.002934	0.002936	
AE104	0.016774	0.016783	0.016850	0.016800	0.016772	0.016839	0.016761	0.016774	
AE201	0.000259	0.000259	0.000260	0.000300	0.000259	0.000262	0.000344	0.000259	
AE203	0.002936	0.002895	0.002948	0.002900	0.002933	0.002947	0.002934	0.002936	
AE204	0.016774	0.016783	0.016850	0.016800	0.016772	0.016839	0.016761	0.016774	
AE205	0.004491	0.004457	0.004510	0.004500	0.004481	0.004508	0.004487	0.004491	
AE206	0.015556	0.015523	0.015630	0.015600	0.015565	0.015616	0.015546	0.015556	
AE226	0.015556	0.015523	0.015630	0.015600	0.015565	0.015616	0.015546	0.015556	
AE245	0.004491	0.004457	0.004510	0.004500	0.004481	0.004508	0.004487	0.004491	

**Table 4.3-7. FC/SZ Outdoor Air Mass Flow Rate (kgda/s)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	0.113	0.113	0.113	0.120	0.114	0.113	0.113	0.113	
AE103	0.111	0.112	0.113	0.112	0.114	0.113	0.113	0.111	
AE104	0.110	0.110	0.113	0.109	0.114	0.113	0.113	0.110	
AE201	0.113	0.113	0.113	0.120	0.114	0.113	0.113	0.113	
AE203	0.111	0.112	0.113	0.113	0.114	0.113	0.113	0.111	
AE204	0.110	0.110	0.113	0.109	0.114	0.113	0.113	0.110	
AE205	0.111	0.111	0.113	0.111	0.114	0.113	0.113	0.111	
AE206	0.110	0.110	0.113	0.110	0.114	0.113	0.113	0.110	
AE226	0.336	0.336	0.340	0.329	0.341	0.340	0.340	0.335	
AE245	0.340	0.340	0.340	0.333	0.341	0.340	0.340	0.340	

**Table 4.3-8. FC/SZ Cooling Coil Outlet Air Temperature (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	29.32	29.33	29.09	29.22	29.05	28.99	29.04	29.31	
AE103	18.50	18.50	18.46	19.00	18.51	18.52	18.51	18.51	
AE104	14.94	14.93	14.70	15.22	14.93	14.84	14.92	14.95	
AE201	29.32	29.33	29.09	28.67	29.05	28.99	29.04	29.31	
AE203	18.50	18.50	18.46	18.44	18.51	18.52	18.51	18.51	
AE204	14.94	14.93	14.72	14.61	14.93	14.84	14.92	14.95	
AE205	18.51	18.50	18.46	18.44	18.52	18.52	18.52	18.51	
AE206	18.51	18.51	18.46	18.44	18.59	18.52	18.58	18.52	
AE226	18.51	18.51	18.44	18.44	18.59	18.52	18.58	18.52	
AE245	18.50	18.50	18.46	18.44	18.51	18.52	18.51	18.50	

**Table 4.3-9. FC/SZ Relative Humidity at Cooling Coil Outlet [RHcco] (%)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	6.39	6.38		6.40	6.52	6.75	6.87	6.35	
AE103	32.84	32.54	33.03	31.65	32.51	32.91	32.55	32.87	
AE104	100.00	100.00	98.66	99.96	99.62	100.00	100.00	100.00	
AE201	6.39	6.38		6.61	6.51	6.75	6.87	6.35	
AE203	32.84	32.54	33.03	32.77	32.51	32.91	32.55	32.86	
AE204	100.00	100.00	98.62	100.00	99.64	100.00	100.00	100.00	
AE205	44.56	44.31	44.81	44.85	44.11	44.59	44.20	44.59	
AE206	100.00	100.00	99.36	99.93	99.03	100.00	100.00	100.00	
AE226	100.00	100.00	99.65	99.93	99.81	100.00	100.00	100.00	
AE245	34.09	33.84	34.33	34.29	33.99	34.06	34.07	34.10	

**Table 4.3-10. FC/SZ Supply Fan Air Temperature Rise (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	0.6058	0.6066	0.5899	0.0000	0.5861	0.5818	0.5890	0.6057	
AE103	0.5838	0.5845	0.5899	0.0000	0.5833	0.5818	0.5890	0.5837	
AE104	0.5758	0.5765	0.5899	0.0000	0.5766	0.5818	0.5890	0.5757	
AE201	0.6058	0.6066	0.5899		0.5861	0.5818	0.5890	0.6057	
AE203	0.5838	0.5845	0.5899		0.5833	0.5818	0.5890	0.5837	
AE204	0.5758	0.5765	0.5899		0.5766	0.5818	0.5890	0.5757	
AE205	0.5836	0.5843	0.5899		0.5816	0.5818	0.5890	0.5835	
AE206	0.5826	0.5833	0.5899		0.5738	0.5818	0.5890	0.5825	
AE226	0.5826	0.5833	0.5899		0.5737	0.5818	0.5890	0.5825	
AE245	0.5838	0.5863	0.5899		0.5831	0.5818	0.5890	0.5837	

**Table 4.3-11. FC/SZ Supply Air Temperature (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	29.92	29.93	29.68	29.22	29.64	29.58	29.62	29.91	
AE103	19.09	19.08	19.05	19.00	19.09	19.10	19.10	19.09	
AE104	15.51	15.51	15.29	15.22	15.50	15.43	15.51	15.52	
AE201	29.92	29.93	29.68		29.64	29.58	29.62	29.91	
AE203	19.09	19.08	19.05		19.09	19.10	19.10	19.10	
AE204	15.51	15.51	15.31		15.50	15.42	15.51	15.52	
AE205	19.09	19.09	19.05		19.11	19.10	19.11	19.09	
AE206	19.10	19.10	19.05		19.16	19.10	19.16	19.11	
AE226	19.10	19.10	19.03		19.16	19.10	19.16	19.10	
AE245	19.09	19.09	19.05		19.09	19.10	19.10	19.08	

**Table 4.3-12. FC/SZ Supply Air Humidity Ratio (g/gda)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	0.001604	0.001604	0.001642	0.001600	0.001612	0.001670	0.001696	0.001595	
AE103	0.004325	0.004284	0.004337	0.004300	0.004284	0.004355	0.004285	0.004330	
AE104	0.010605	0.010599	0.010296	0.010800	0.010558	0.010581	0.010585	0.010612	
AE201	0.001604	0.001604	0.001642	0.001600	0.001612	0.001670	0.001698	0.001595	
AE203	0.004325	0.004285	0.004337	0.004300	0.004284	0.004355	0.004285	0.004330	
AE204	0.010605	0.010600	0.010306	0.010400	0.010561	0.010580	0.010585	0.010612	
AE205	0.005884	0.005850	0.005900	0.005900	0.005832	0.005916	0.005838	0.005888	
AE206	0.013368	0.013367	0.013227	0.013300	0.013302	0.013424	0.013413	0.013376	
AE226	0.013368	0.013367	0.013257	0.013300	0.013410	0.013424	0.013413	0.013373	
AE245	0.004491	0.004457	0.004510	0.004500	0.004481	0.004508	0.004487	0.004491	

**Table 4.3-13. FC/SZ Supply Air Specific Volume (L/kgda)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	860.78	860.69	833.33		830.35		833.00	856.54	
AE103	833.63	833.45	833.33		830.35		833.00	829.58	
AE104	831.69	831.55	833.33		830.35		833.00	827.67	
AE201	860.78	860.69	833.33		830.35		833.00	856.54	
AE203	833.63	833.45	833.33		830.35		833.00	829.59	
AE204	831.69	831.55	833.33		830.35		833.00	827.67	
AE205	835.71	835.55	833.33		830.35		833.00	831.64	
AE206	845.69	845.57	833.33		830.35		833.00	841.60	
AE226	845.69	845.57	833.33		830.35		833.00	841.58	
AE245	833.85	833.70	833.33		830.35		833.00	829.76	

**Table 4.3-14. FC/SZ Supply Air Enthalpy (J/gda)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	34.20		34.02		33.90		34.14	34.17	
AE103	30.17		30.14		30.05		30.08	30.19	
AE104	42.44		41.40		42.29		42.37	42.46	
AE201	34.21		34.02		33.90		34.14	34.17	
AE203	30.18		30.14		30.05		30.08	30.19	
AE204	42.44		41.44		42.29		42.37	42.46	
AE205	34.13		34.10		33.99		34.03	34.14	
AE206	53.12		52.67		53.00		53.29	53.15	
AE226	53.12		52.74		53.27		53.29	53.14	
AE245	30.60		30.58		30.55		30.59	30.59	

**Table 4.3-15. FC/SZ Supply Air Mass Flow Rate (kgda/s)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	0.3296	0.3297	0.3398	0.3586	0.3410	0.3398	0.3398	0.3296	
AE103	0.3404	0.3404	0.3398	0.3372	0.3410	0.3398	0.3398	0.3404	
AE104	0.3412	0.3412	0.3398	0.3279	0.3410	0.3398	0.3398	0.3411	
AE201	0.3296	0.3297	0.3398	0.3591	0.3410	0.3398	0.3398	0.3296	
AE203	0.3404	0.3404	0.3398	0.3377	0.3410	0.3398	0.3398	0.3403	
AE204	0.3412	0.3412	0.3398	0.3284	0.3410	0.3398	0.3398	0.3411	
AE205	0.3395	0.3396	0.3398	0.3333	0.3410	0.3398	0.3398	0.3395	
AE206	0.3355	0.3356	0.3398	0.3295	0.3410	0.3398	0.3398	0.3355	
AE226	0.3355	0.3356	0.3398	0.3294	0.3410	0.3398	0.3398	0.3355	
AE245	0.3403	0.3403	0.3398	0.3331	0.3410	0.3398	0.3398	0.3403	

**Table 4.3-16. FC/SZ Outdoor Air Mass Flow to Supply Air Mass Flow Ratio (fraction)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	0.342	0.342	0.333	0.333	0.333	0.333	0.334	0.342	
AE103	0.328	0.328	0.333	0.333	0.333	0.333	0.333	0.328	
AE104	0.323	0.323	0.333	0.333	0.333	0.333	0.334	0.323	
AE201	0.342	0.342	0.333	0.333	0.333	0.333	0.333	0.342	
AE203	0.328	0.328	0.333	0.333	0.333	0.333	0.333	0.328	
AE204	0.323	0.323	0.333	0.333	0.333	0.333	0.333	0.323	
AE205	0.328	0.328	0.333	0.333	0.333	0.333	0.333	0.328	
AE206	0.328	0.328	0.333	0.333	0.333	0.333	0.333	0.328	
AE226	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	
AE245	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	

**Table 4.3-17. FC/SZ Zone Air Temperature (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	21.11	21.11	21.10	21.11	21.12	21.11	21.11	21.11	
AE103	23.33	23.33	23.35	23.33	23.33	23.33	23.33	23.33	
AE104	23.89	23.89	23.88	23.89	23.88	23.89	23.89	23.89	
AE201	21.11	21.11	21.10	21.11	21.12	21.11	21.11	21.11	
AE203	23.33	23.33	23.35	23.33	23.33	23.33	23.33	23.34	
AE204	23.89	23.89	23.90	23.89	23.88	23.89	23.89	23.89	
AE205	23.33	23.33	23.35	23.33	23.33	23.33	23.33	23.33	
AE206	23.33	23.33	23.34	23.33	23.33	23.33	23.33	23.34	
AE226	23.33	23.33	23.33	23.33	23.33	23.33	23.33	23.33	
AE245	23.33	23.33	23.35	23.33	23.33	23.33	23.33	23.32	

**Table 4.3-18. FC/SZ Zone Humidity Ratio (g/gda)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	0.002304	0.002304	0.002332	0.002200	0.002289	0.002370	0.002373	0.002291	
AE103	0.005002	0.004961	0.005031	0.005000	0.004960	0.005059	0.004960	0.005009	
AE104	0.011280	0.011274	0.010986	0.011500	0.011233	0.011285	0.011261	0.011291	
AE201	0.002304	0.002304	0.002332	0.002200	0.002289	0.002374	0.002374	0.002291	
AE203	0.005002	0.004961	0.005031	0.005000	0.004960	0.005059	0.004960	0.005009	
AE204	0.011280	0.011274	0.010995	0.011100	0.011236	0.011284	0.011261	0.011291	
AE205	0.006562	0.006528	0.006593	0.006600	0.006507	0.006620	0.006513	0.006569	
AE206	0.014054	0.014053	0.013921	0.014000	0.013978	0.014128	0.014089	0.014065	
AE226	0.014054	0.014053	0.013951	0.014100	0.014086	0.014128	0.014089	0.014062	
AE245	0.005168	0.005134	0.005204	0.005200	0.005156	0.005212	0.005162	0.005170	

**Table 4.3-19. FC/SZ Moisture Added to Zone by Latent Gains (g/s)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE101	0.231	0.231	0.235	0.215	0.231	0.238	0.230	0.229	
AE103	0.230	0.230	0.236	0.236	0.230	0.239	0.230	0.231	
AE104	0.230	0.230	0.234	0.229	0.230	0.239	0.229	0.232	
AE201	0.231	0.231	0.235	0.215	0.231	0.239	0.230	0.229	
AE203	0.230	0.230	0.236	0.236	0.230	0.239	0.230	0.231	
AE204	0.230	0.230	0.234	0.230	0.230	0.239	0.229	0.232	
AE205	0.230	0.230	0.236	0.233	0.230	0.239	0.230	0.231	
AE206	0.230	0.230	0.236	0.231	0.230	0.239	0.230	0.231	
AE226	0.230	0.230	0.236	0.264	0.230	0.239	0.230	0.231	
AE245	0.230	0.230	0.236	0.233	0.230	0.239	0.230	0.231	

[ (Zone Supply Air Mass Flow) x { (Zone Humidity Ratio) - (Zone Supply Air Humidity Ratio) } X 1000 g/kg ]

**Table 4.3-20. FC/SZ Return Fan Air Temperature Rise (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE201	0.2721	0.2950	0.2950	0.2778	0.2897	0.2909	0.2950	0.2720	
AE203	0.3126	0.2970	0.2950	0.2778	0.2913	0.2909	0.2950	0.3125	
AE204	0.3261	0.2972	0.2950	0.2778	0.2880	0.2909	0.2950	0.3260	
AE205	0.3125	0.2969	0.2950	0.3333	0.2904	0.2909	0.2950	0.3123	
AE206	0.3119	0.2964	0.2950	0.2778	0.2865	0.2909	0.2950	0.3118	
AE226	0.3119	0.2964	0.2950	0.3333	0.2865	0.2909	0.2950	0.3118	
AE245	0.3126	0.2970	0.2950	0.3333	0.2912	0.2909	0.2950	0.3125	

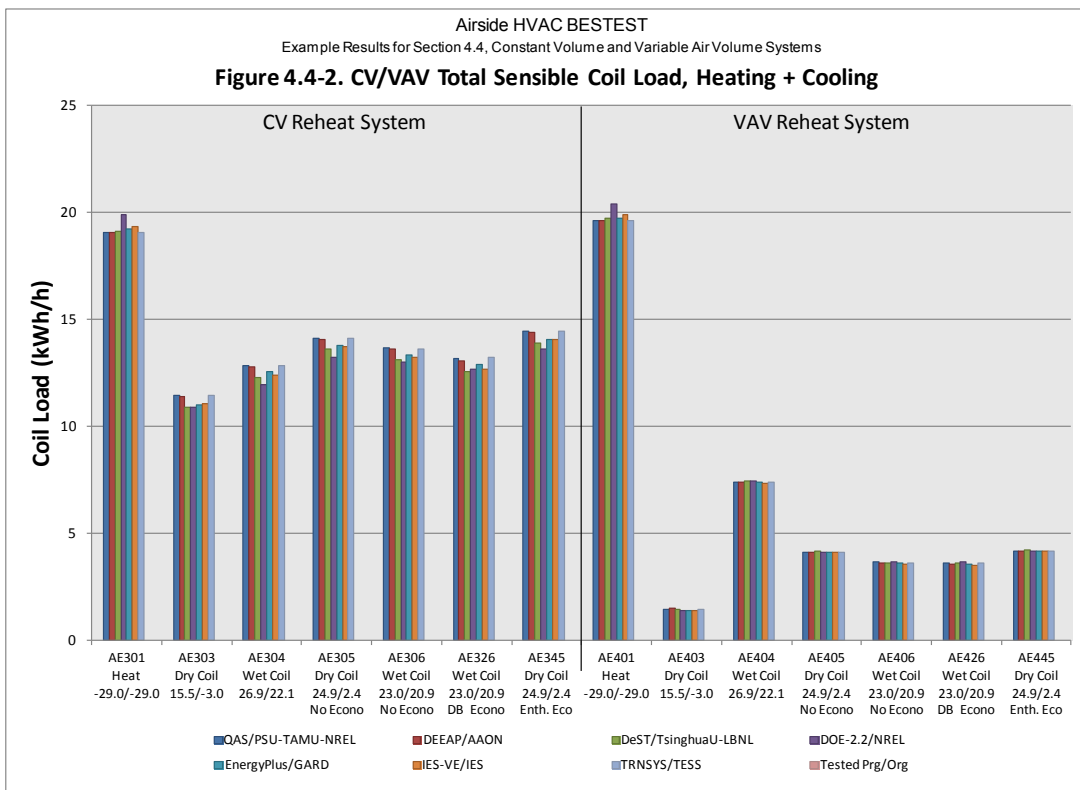
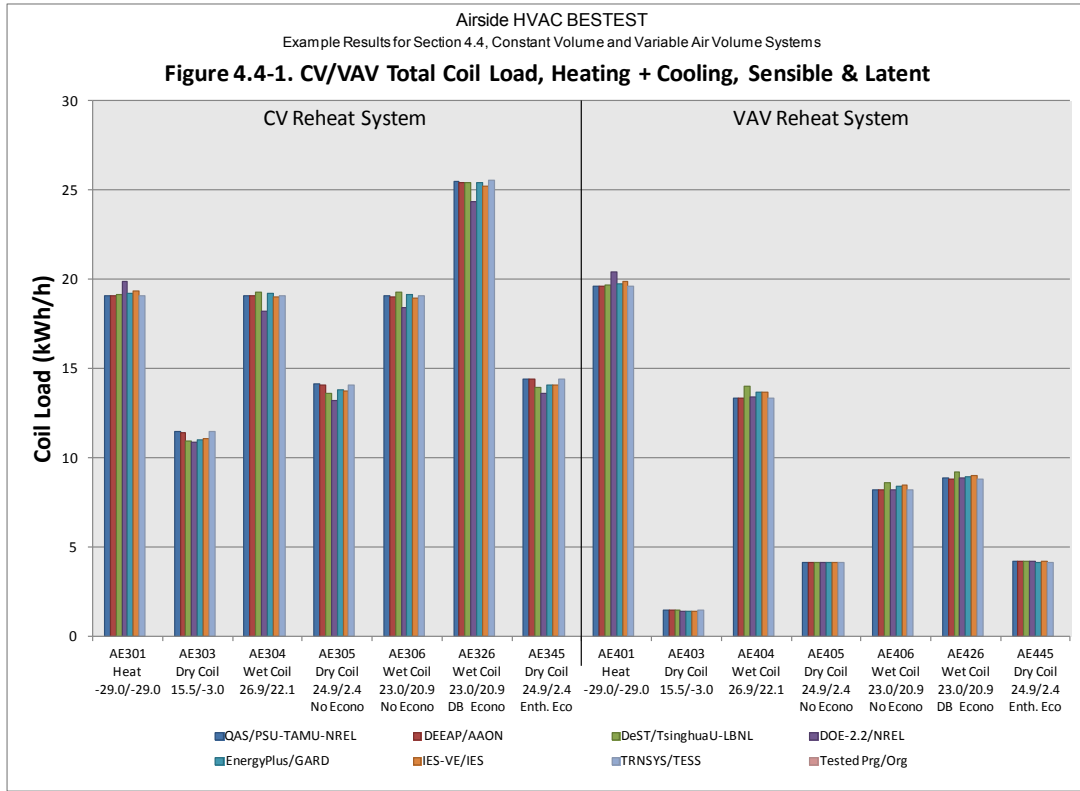
**Table 4.3-21. FC/SZ Delta Coil Load: SZ - FC (kWh/h)**

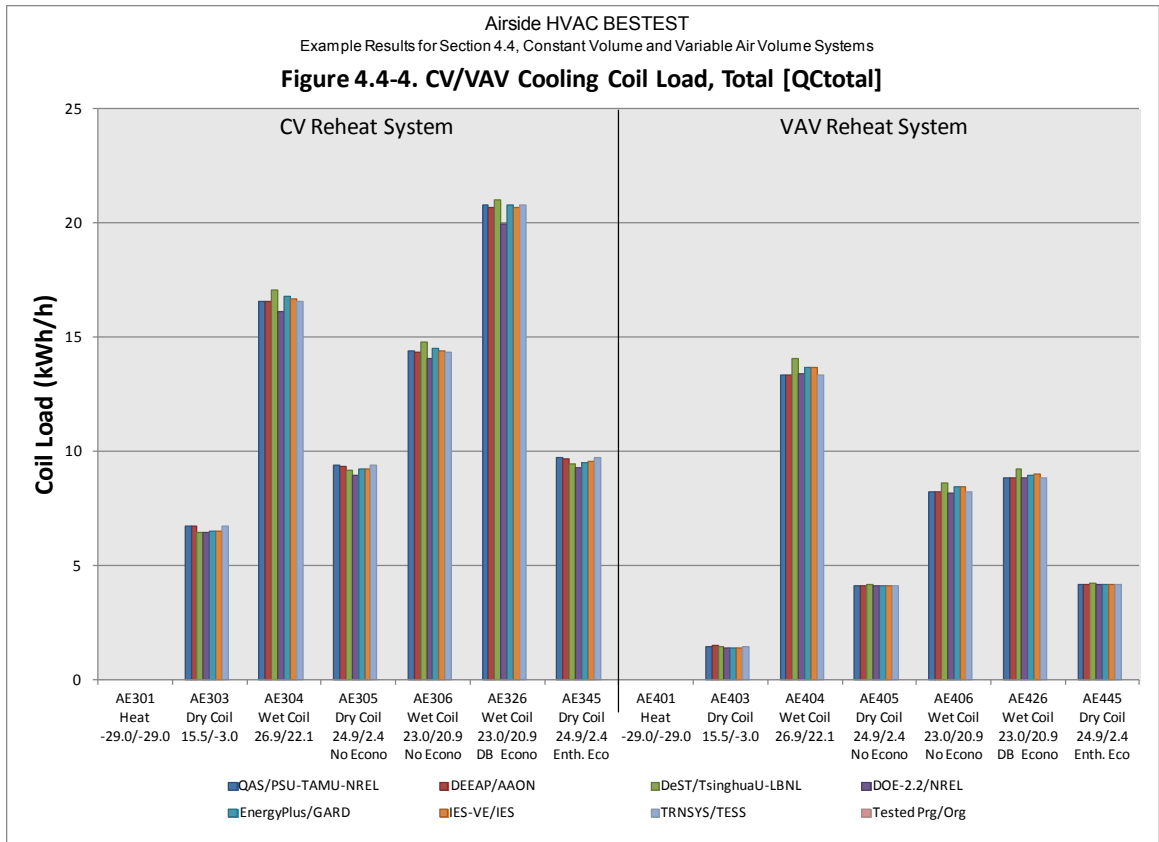
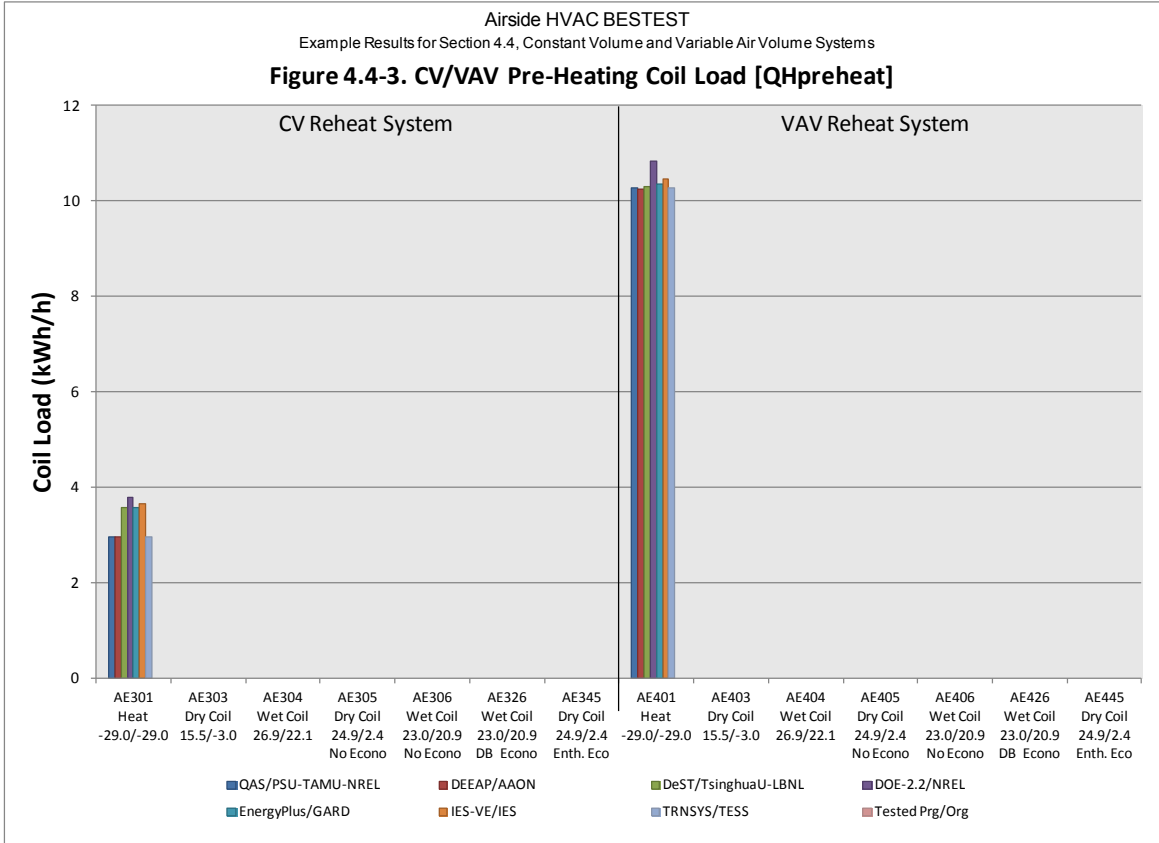
Test Case	Load	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE201-AE101	Heating	-0.060	-0.065	-0.067	-0.058	-0.066	-0.067	-0.070	-0.060	
AE203-AE103	Sensible Cooling	0.073	0.069	0.068	0.063	0.067	0.068	0.068	0.072	
AE204-AE104	Total Cooling	0.077	0.070	0.061	0.178	0.066	0.068	0.068	0.077	
	Sensible Cooling	0.077	0.070	0.065	0.069	0.067	0.067	0.067	0.077	
	Latent Cooling	0.000	-0.000	-0.004	0.109	-0.001	0.001	0.001	-0.000	

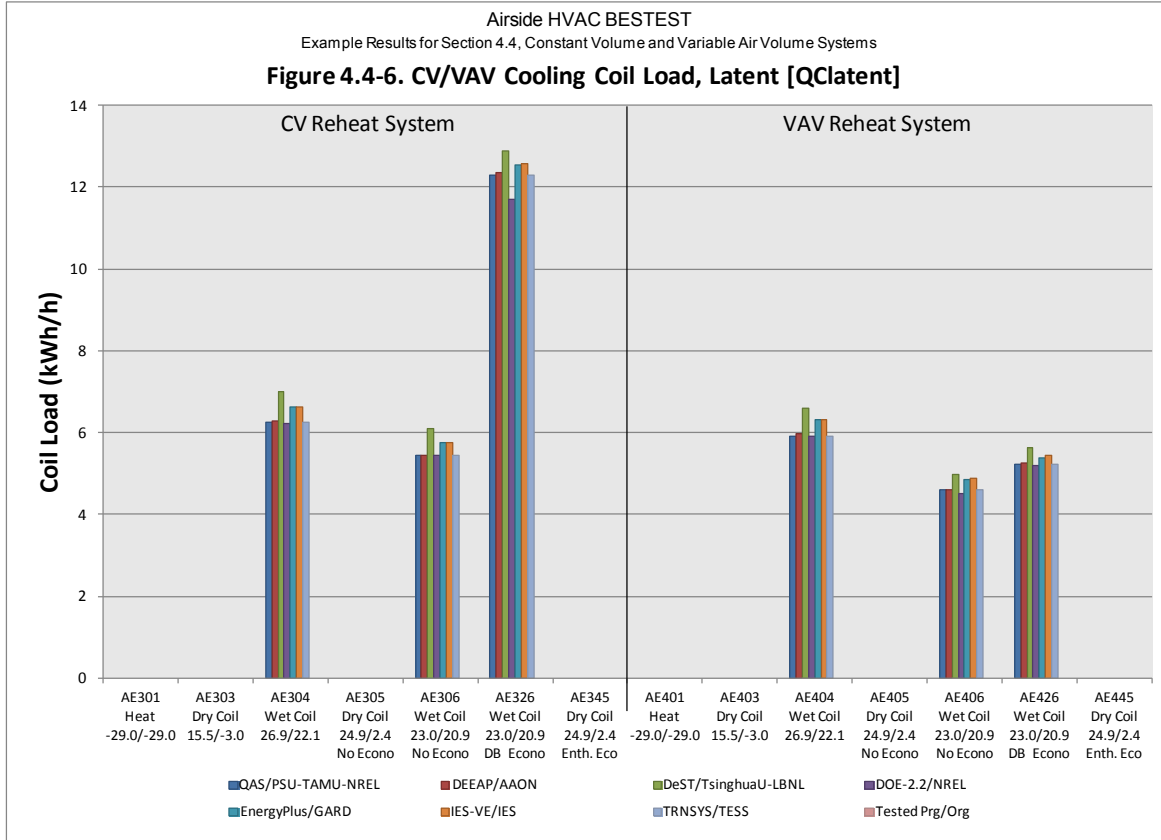
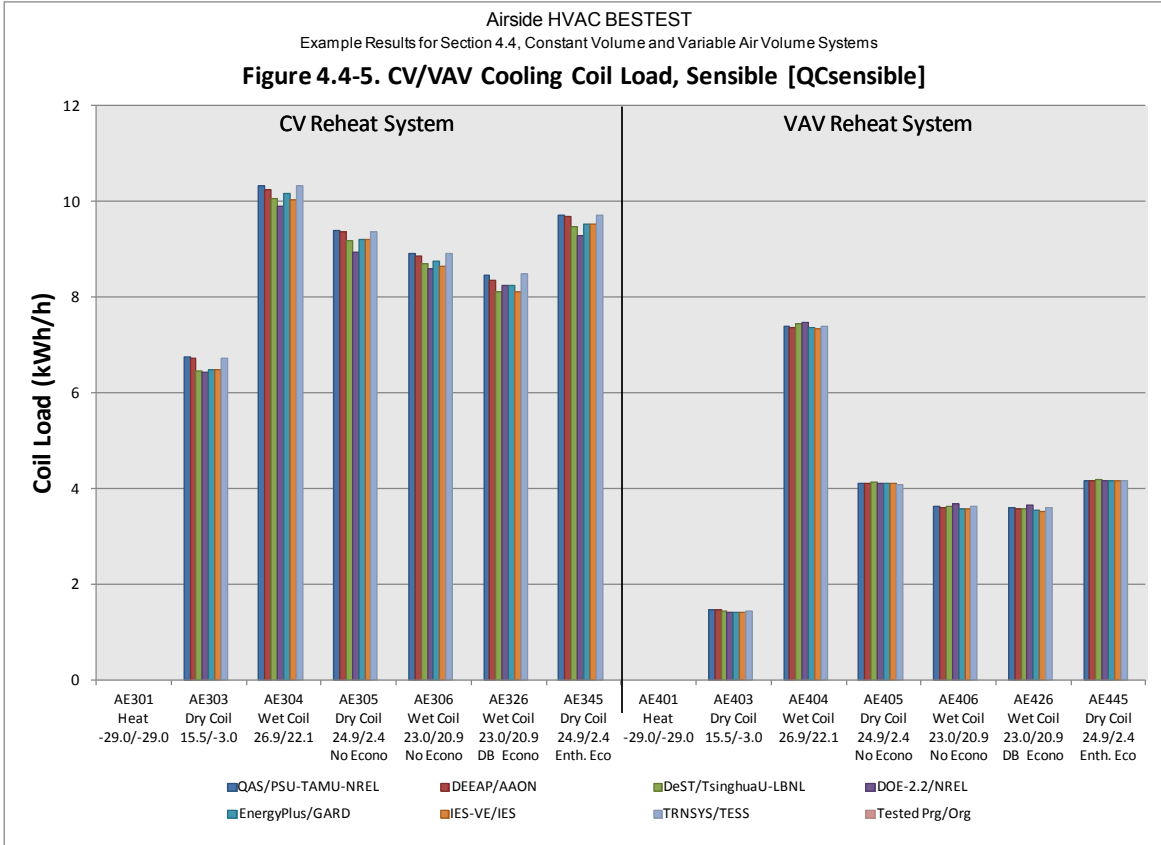
**Table 4.3-22. SZ Delta Coil Load: Economizer Operation (kWh/h)**

Test Case	Load	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	LCEM MLIT-TTE	TRNSYS TESS	Tested Prg Org
AE245-AE205	Total Cooling	0.291	0.292	0.287	0.279	0.295	0.295	0.291	0.296	
	Sensible Cooling	0.291	0.292	0.287	0.279	0.295	0.295	0.292	0.296	
	Latent Cooling	0.000	0.000	0.000	0.000	0.000	0.000	-0.000	0.000	
AE226-AE206	Total Cooling	0.685	0.671	0.825	0.673	0.680	0.681	0.692	0.684	
	Sensible Cooling	-0.145	-0.144	-0.138	-0.126	-0.142	-0.143	-0.142	-0.143	
	Latent Cooling	0.830	0.815	0.963	0.799	0.822	0.825	0.835	0.827	

## 4.4 Results Graphs and Tables for Constant Volume Reheat and Variable Air Volume Systems



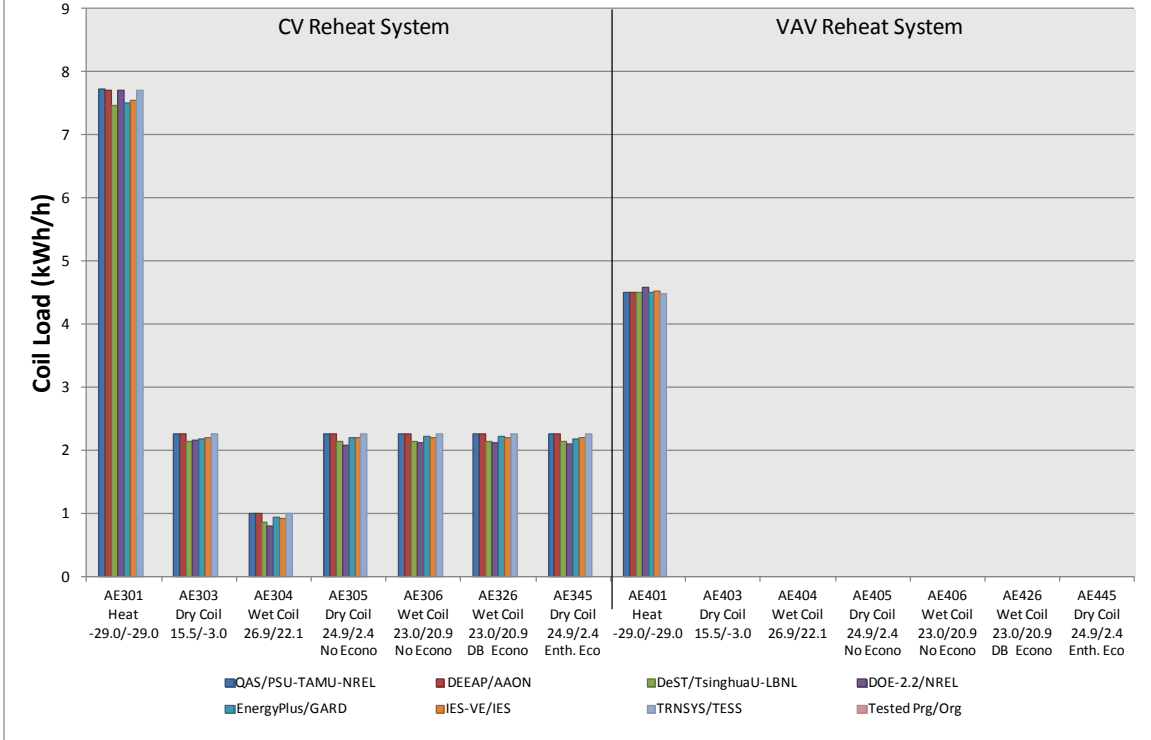






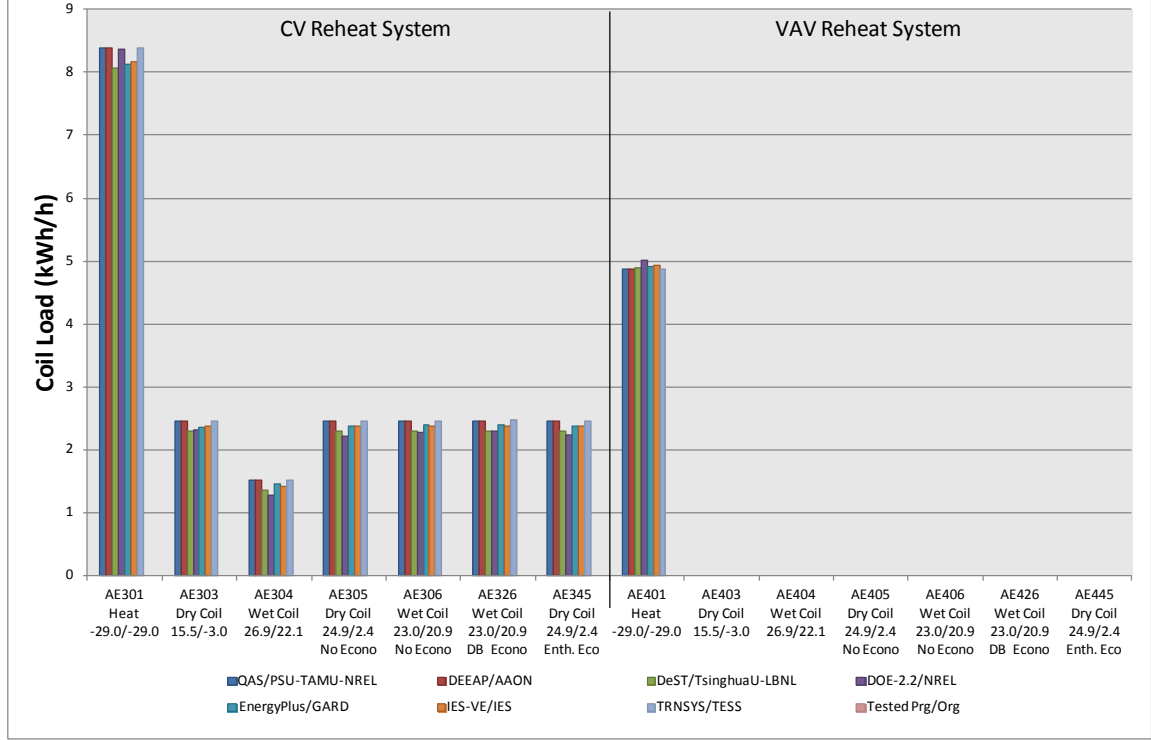
Airside HVAC BESTEST  
 Example Results for Section 4.4, Constant Volume and Variable Air Volume Systems

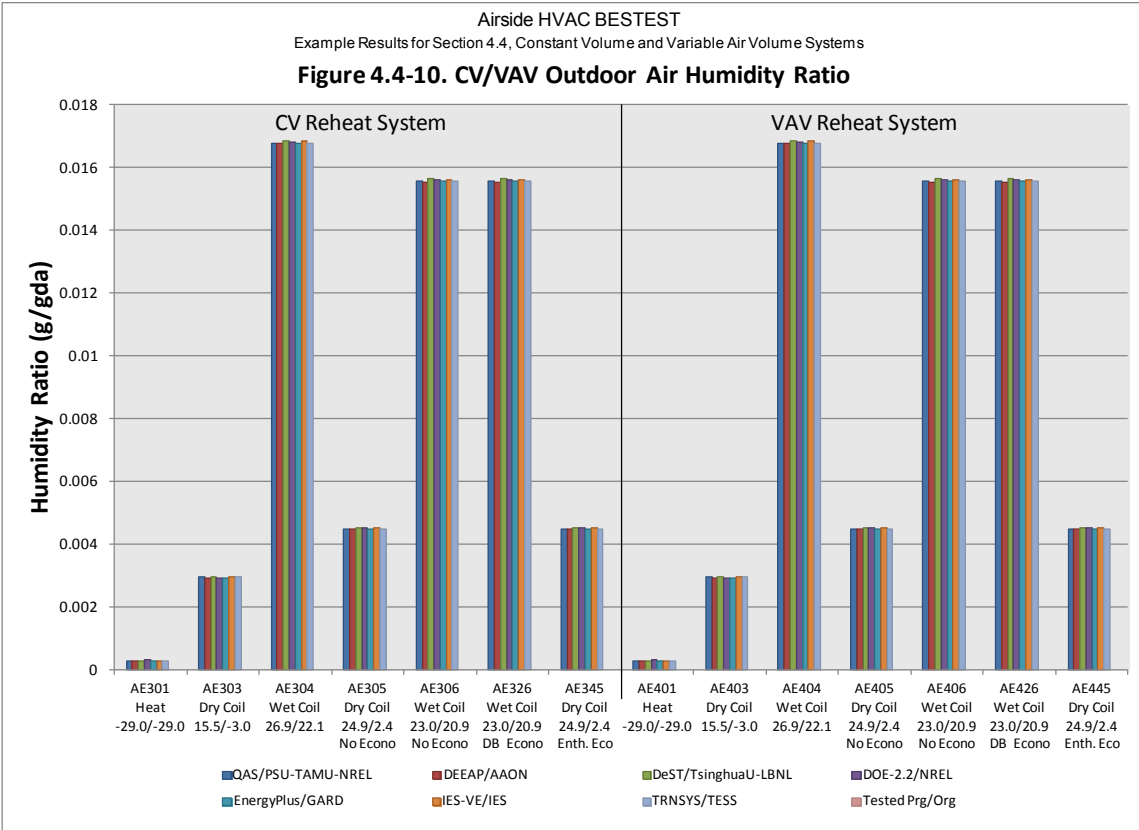
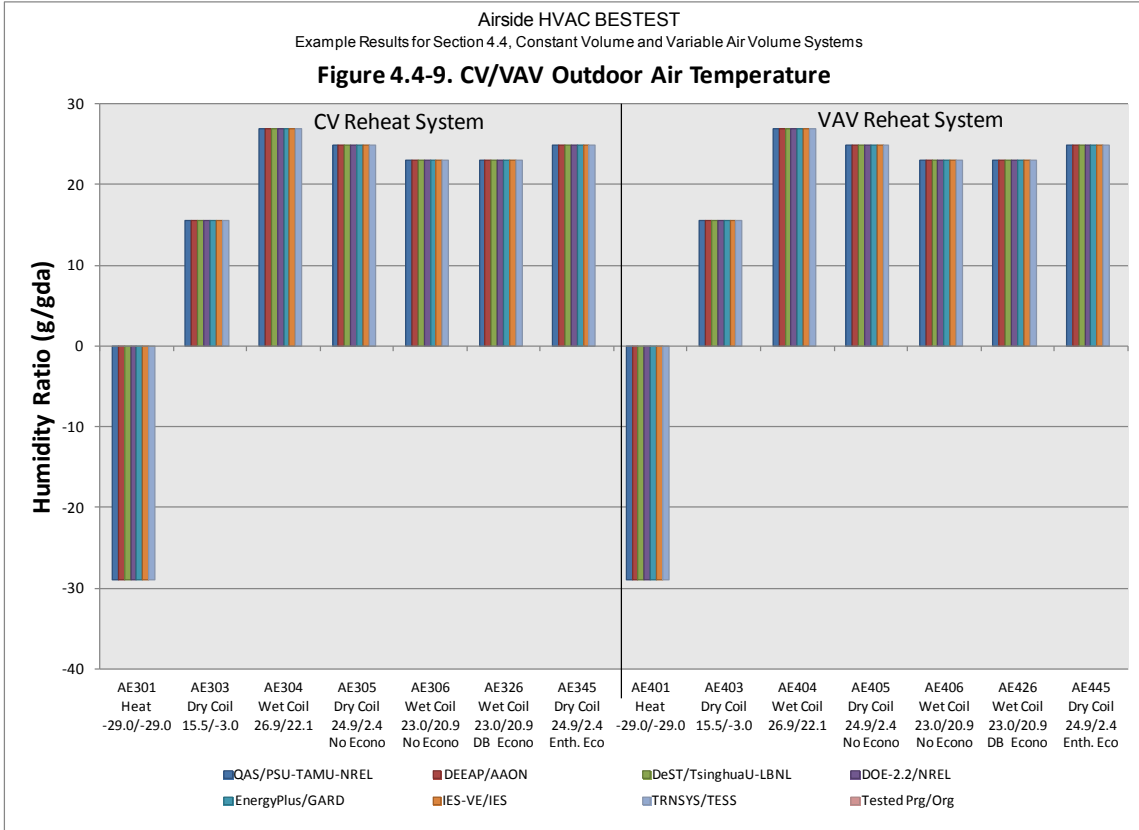
Figure 4.4-7. CV/VAV Zone 1 Reheat Load [QH1reheat]

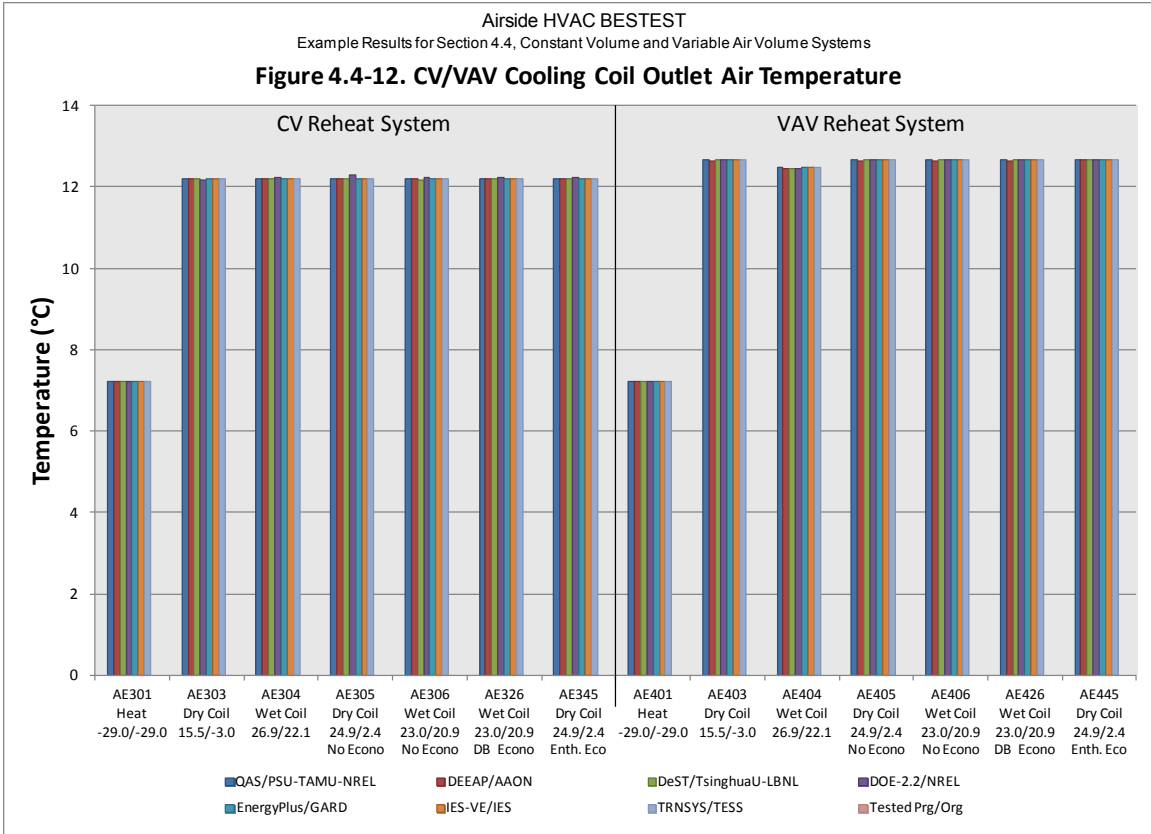
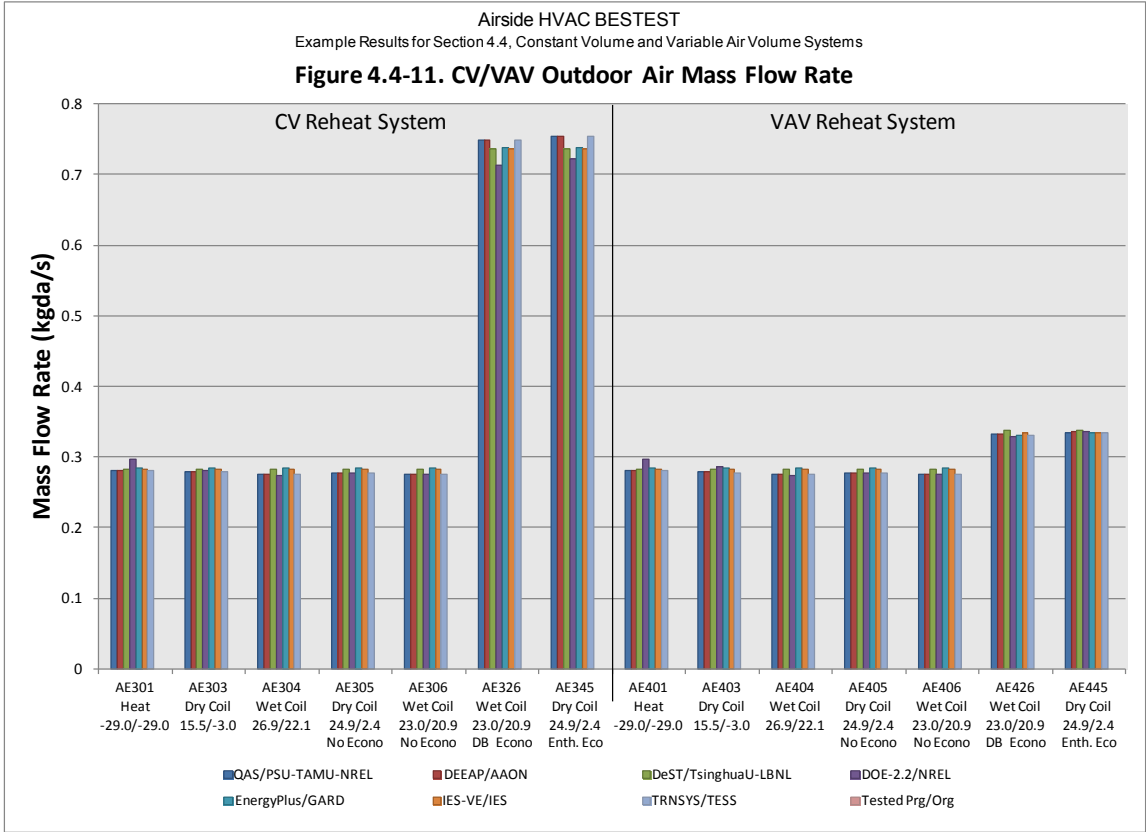


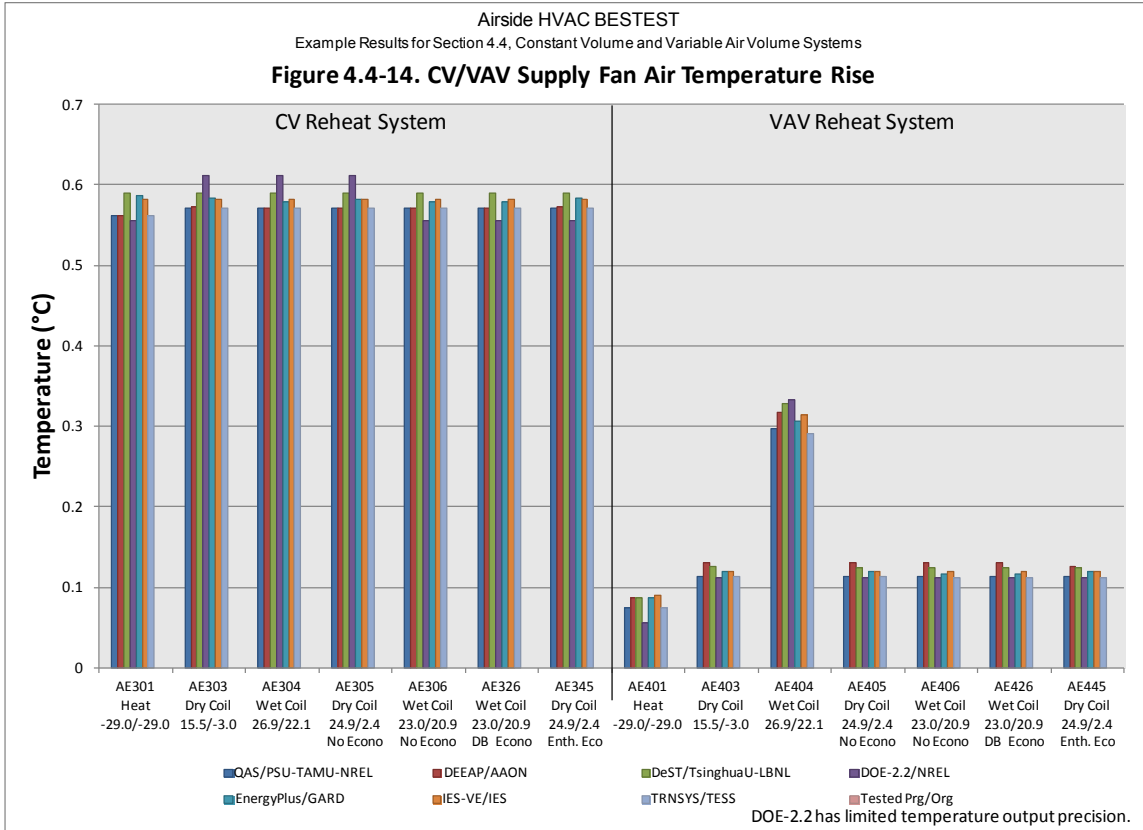
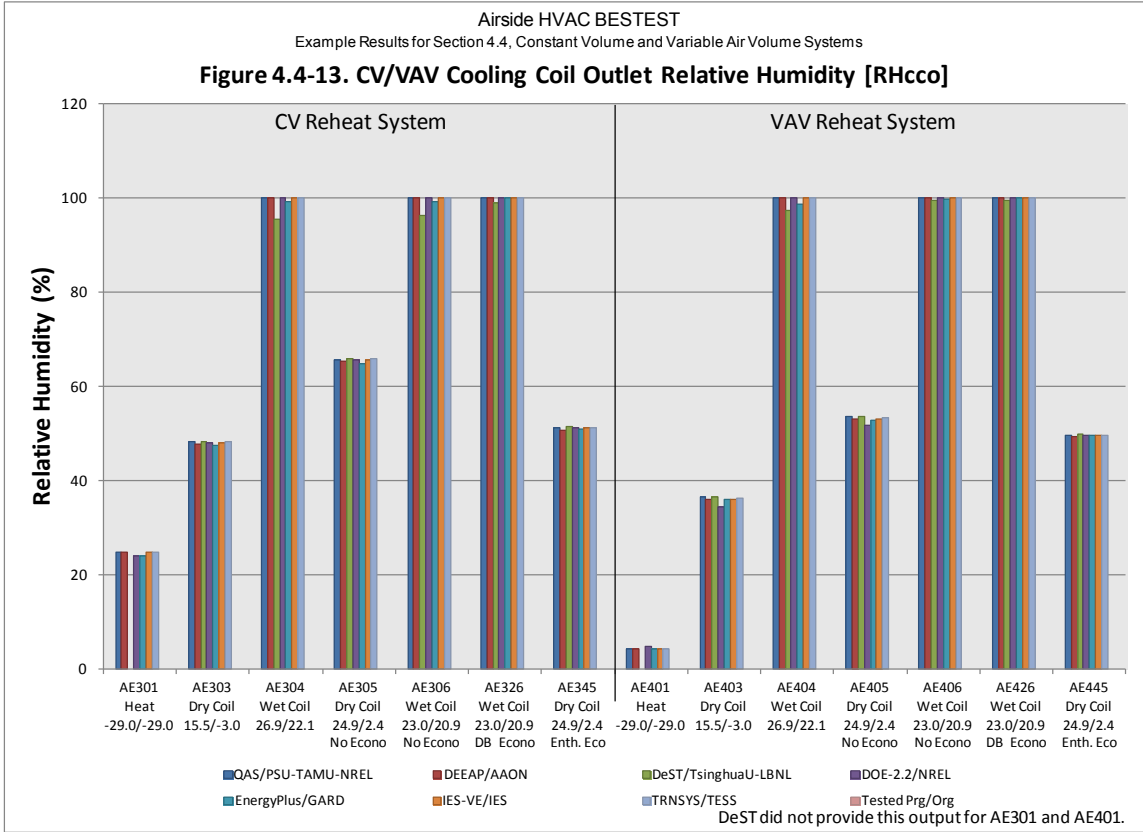
Airside HVAC BESTEST  
 Example Results for Section 4.4, Constant Volume and Variable Air Volume Systems

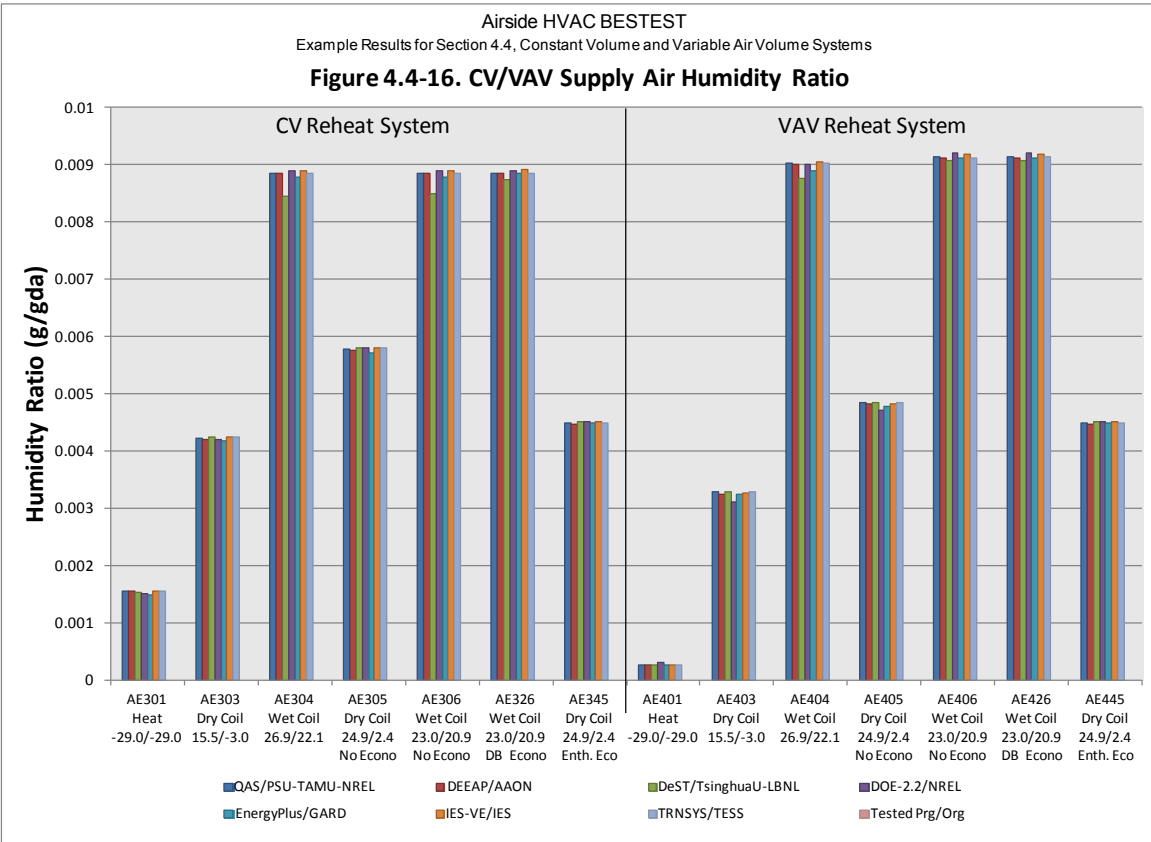
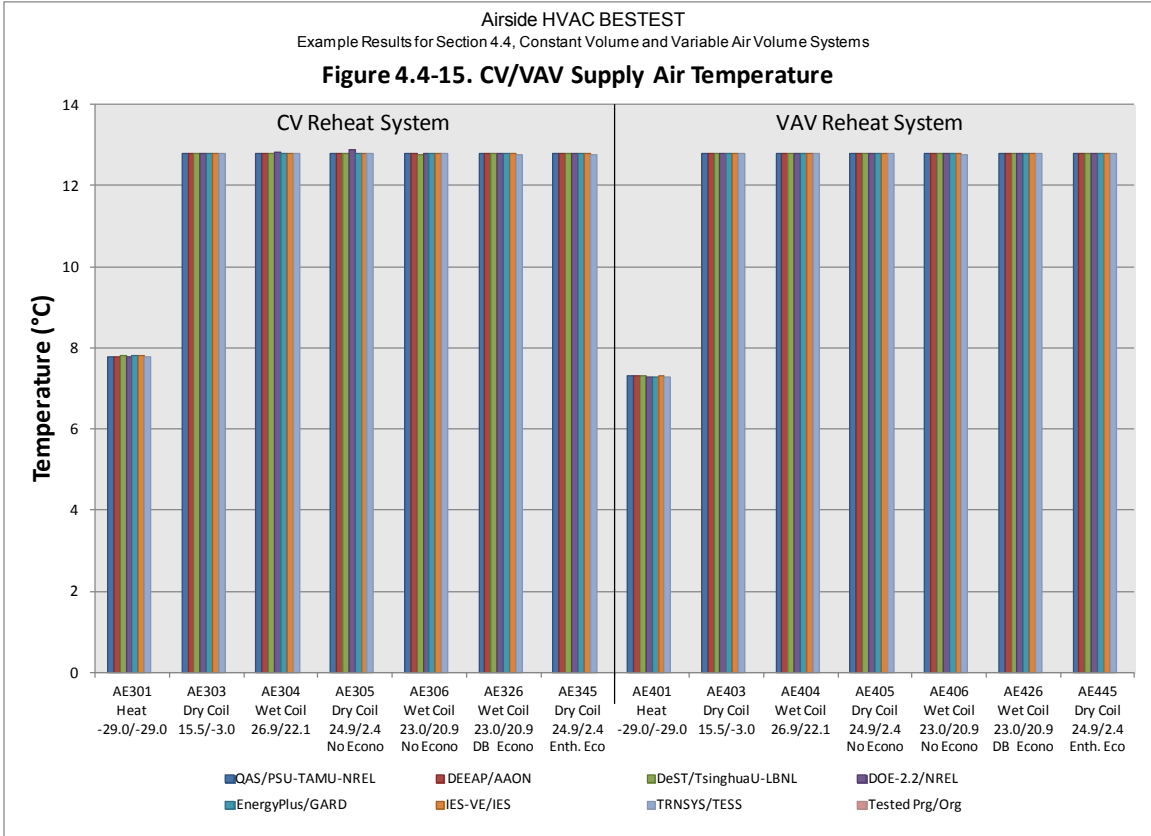
Figure 4.4-8. CV/VAV Zone 2 Reheat Load [QH2reheat]

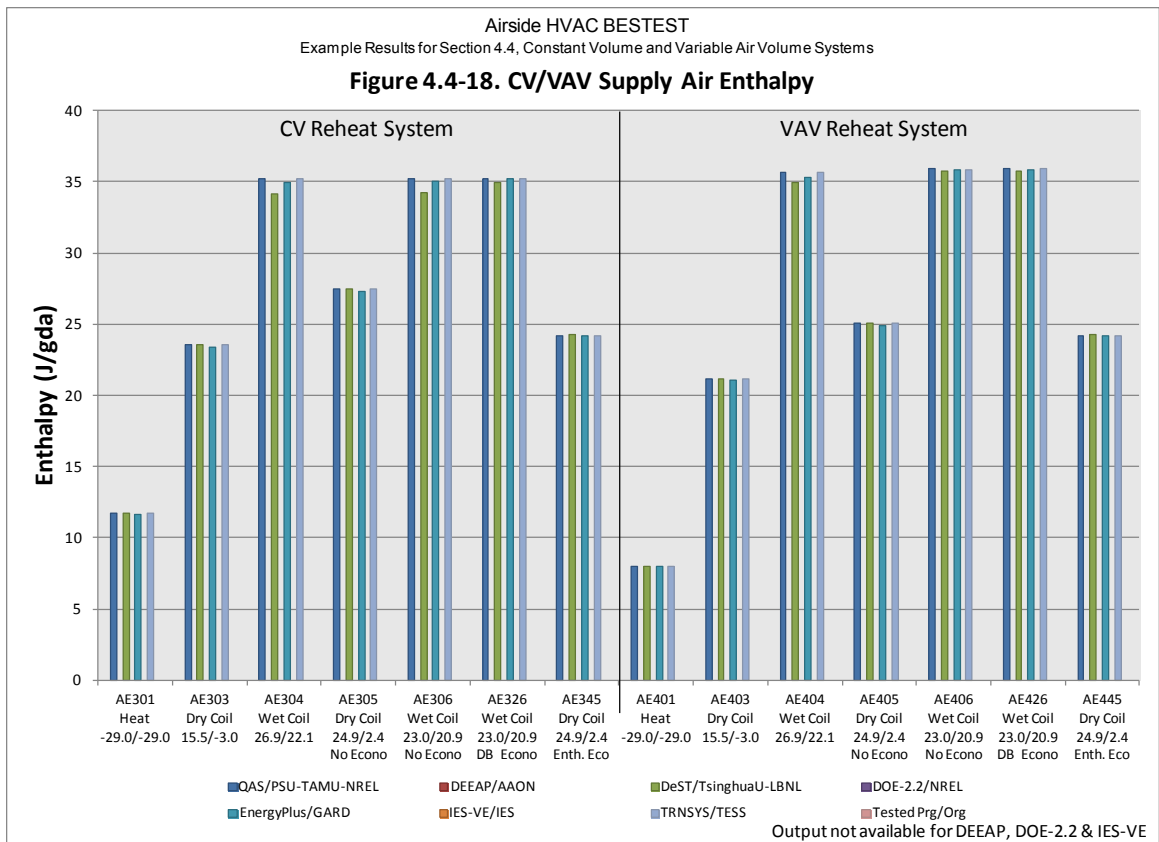
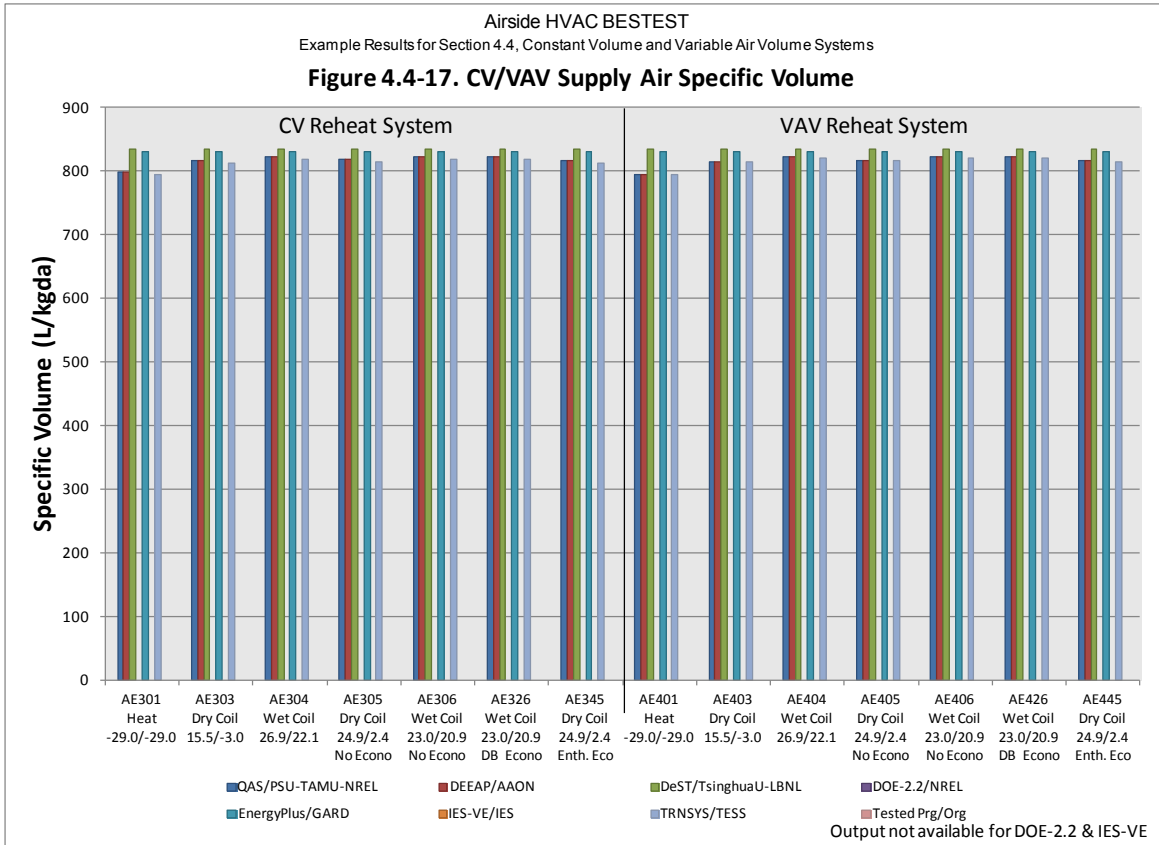


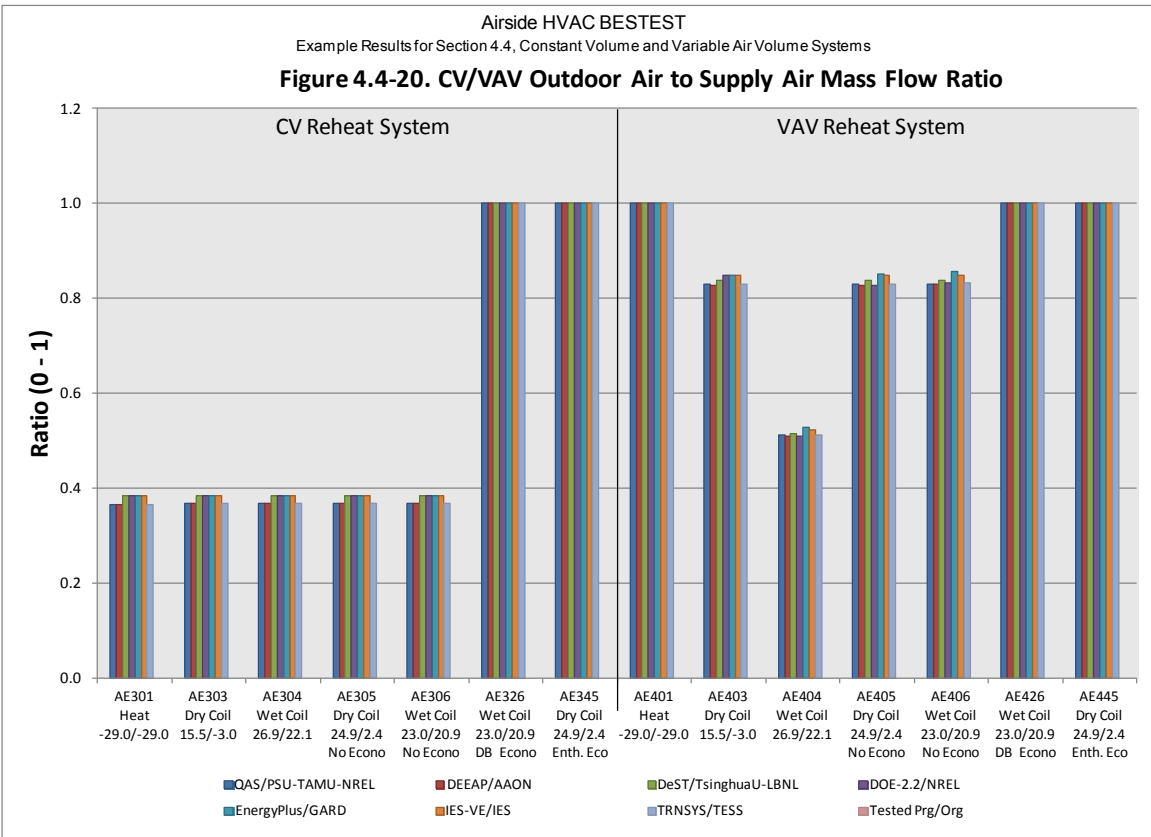
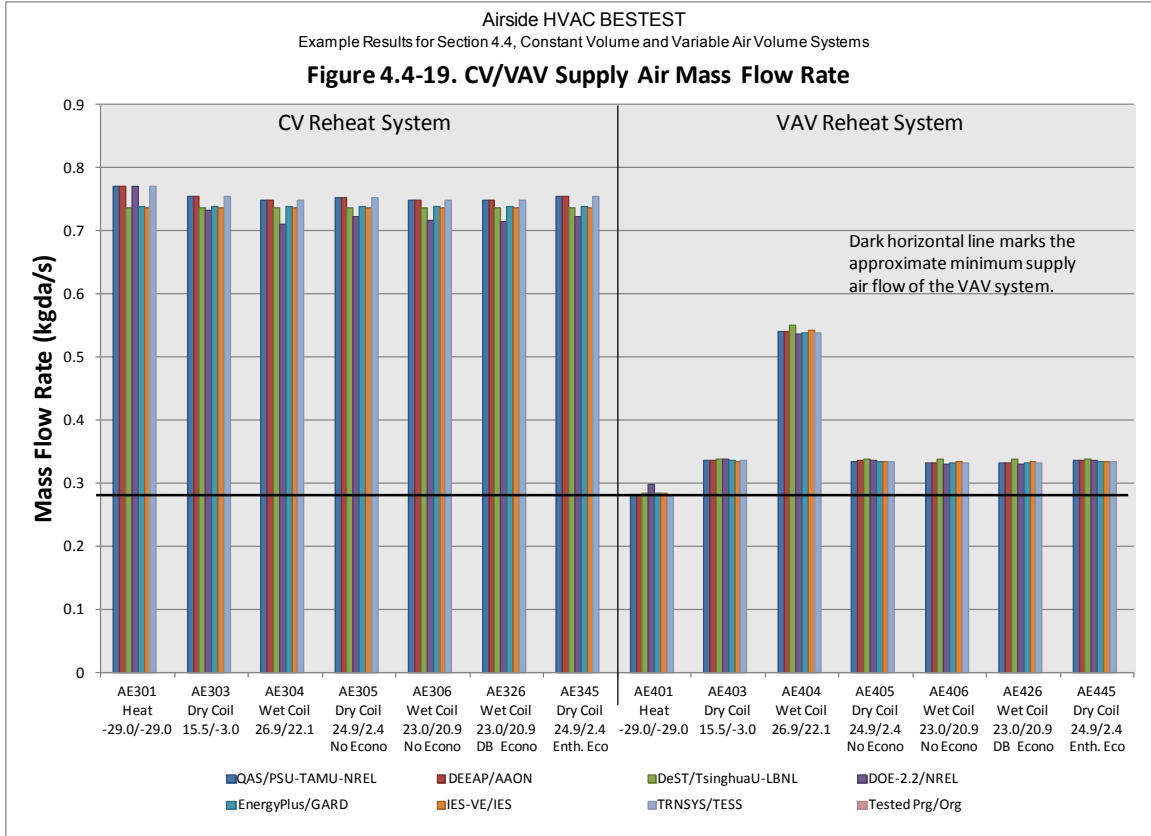


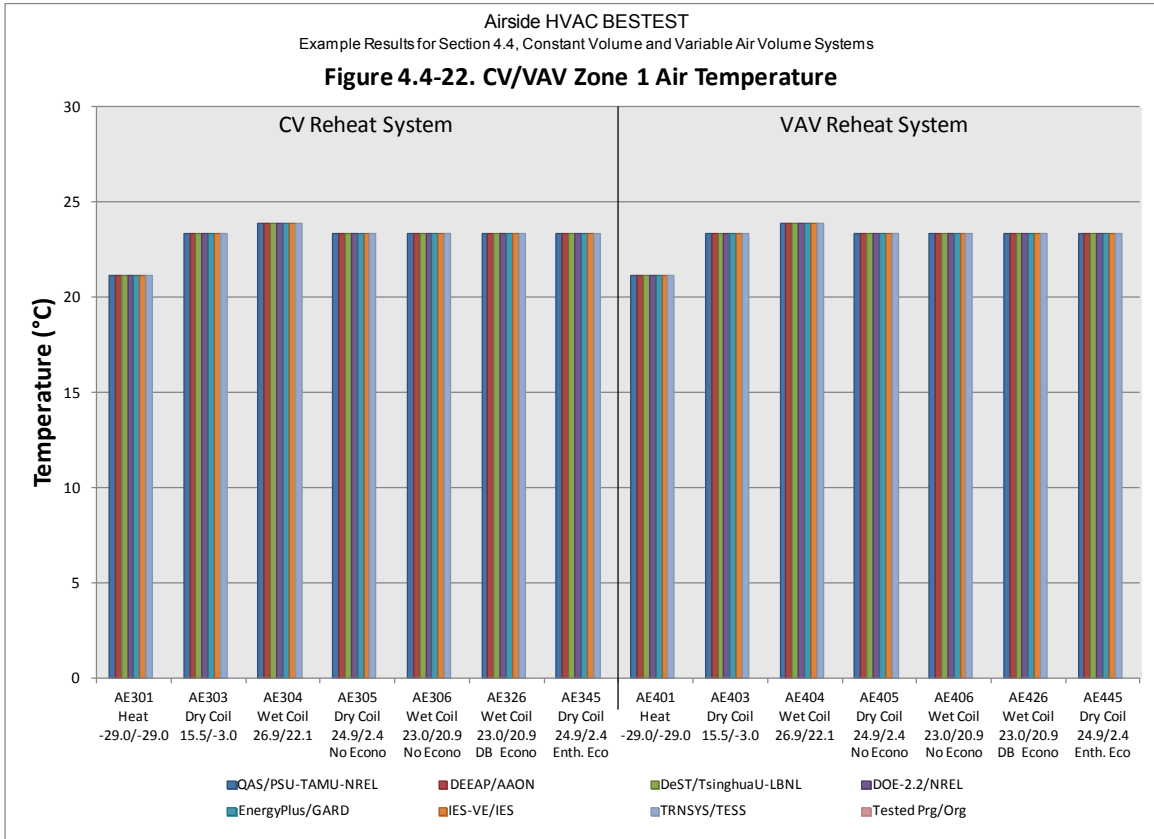
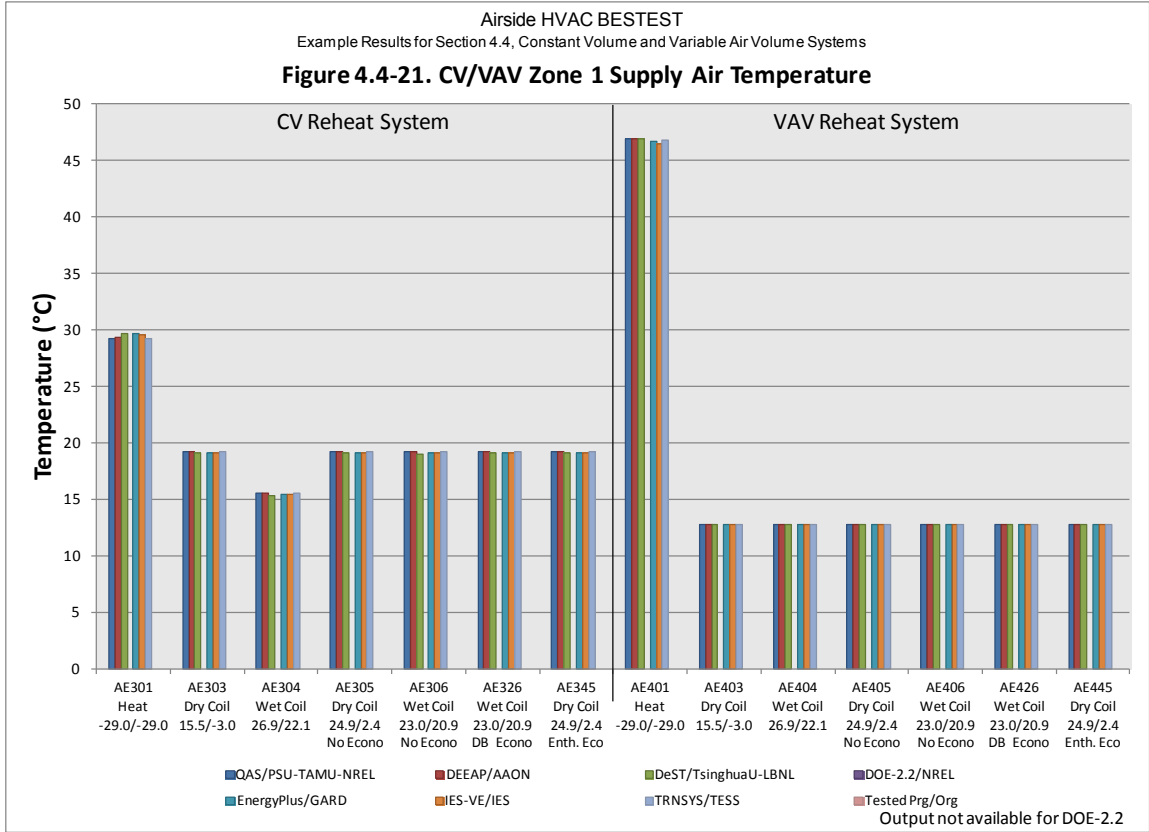




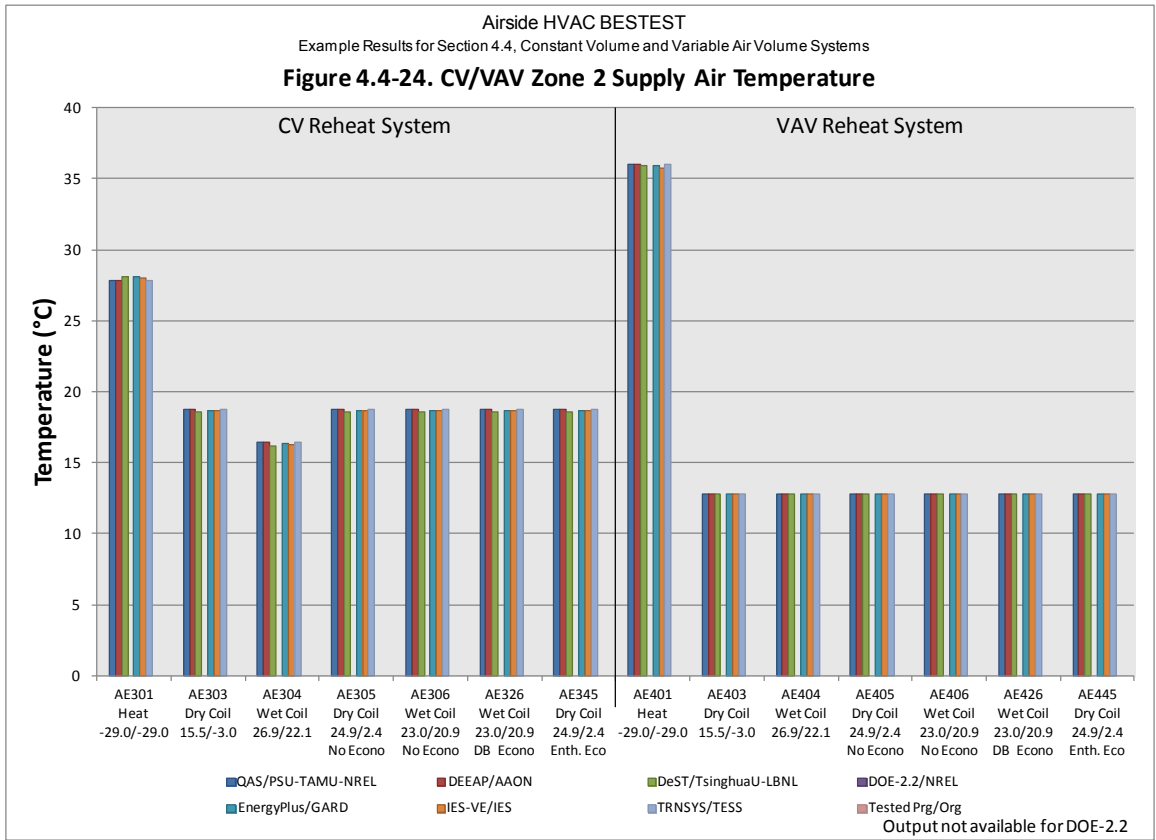
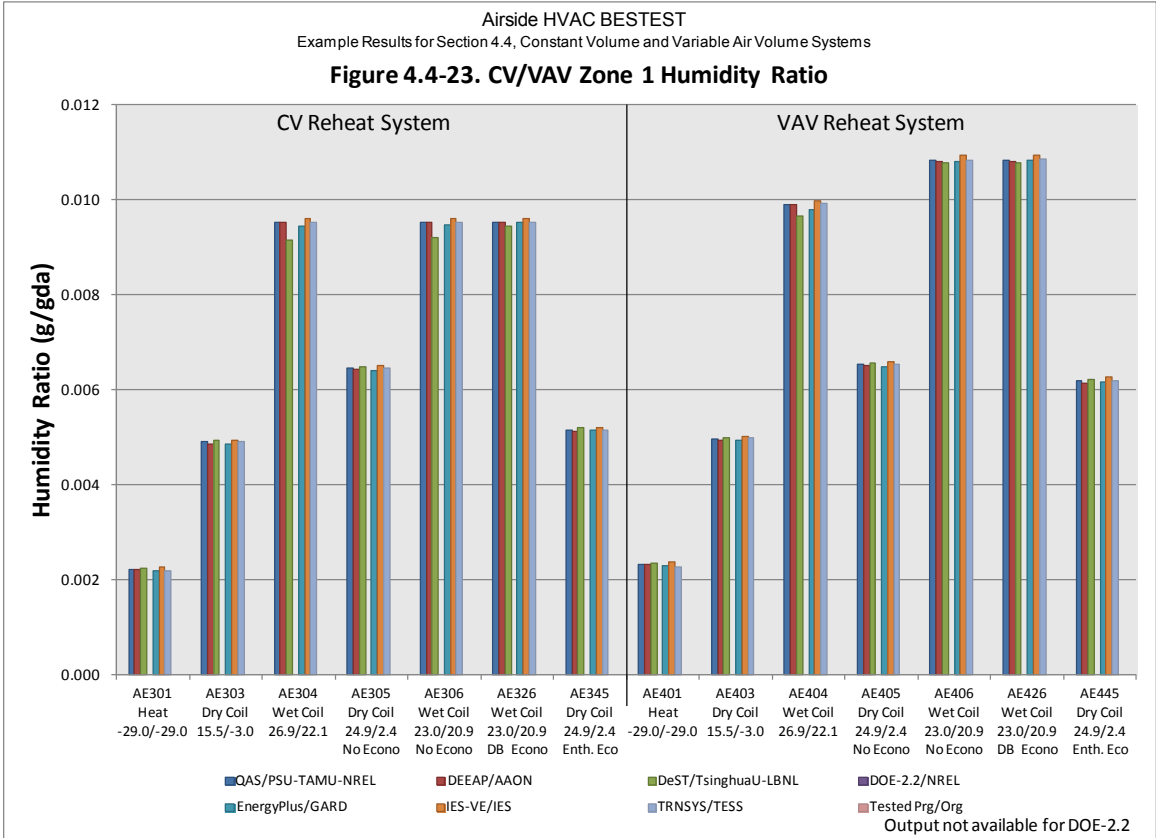






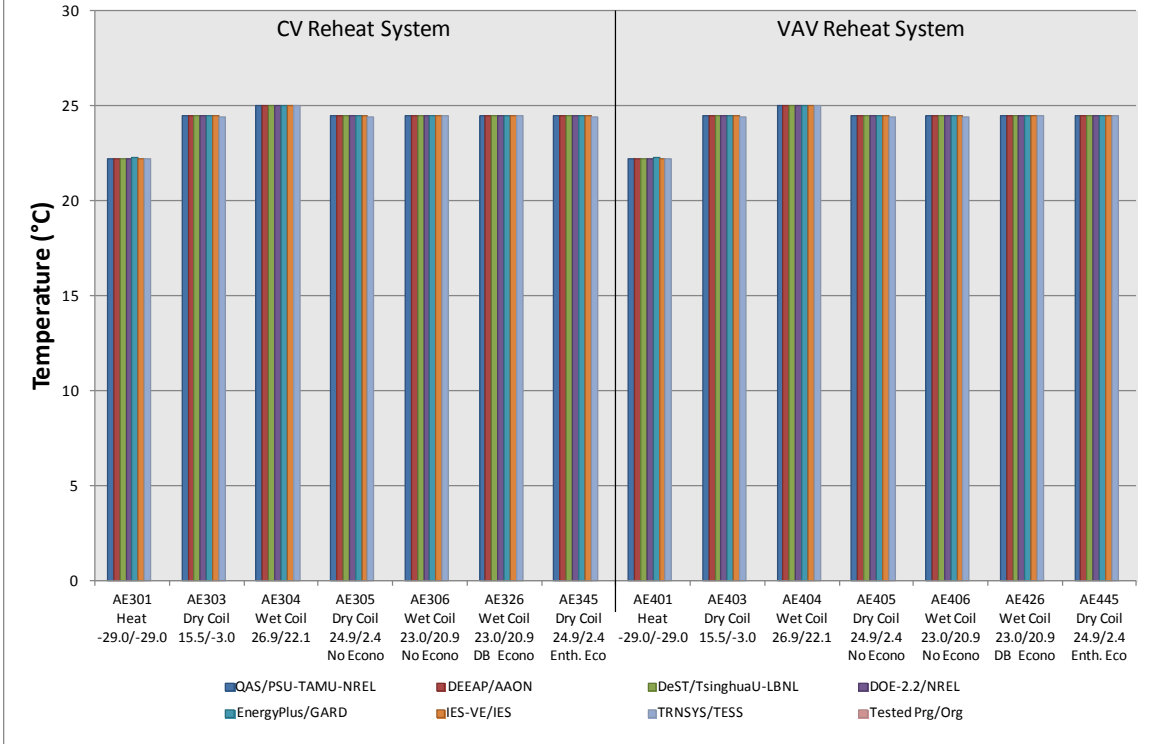






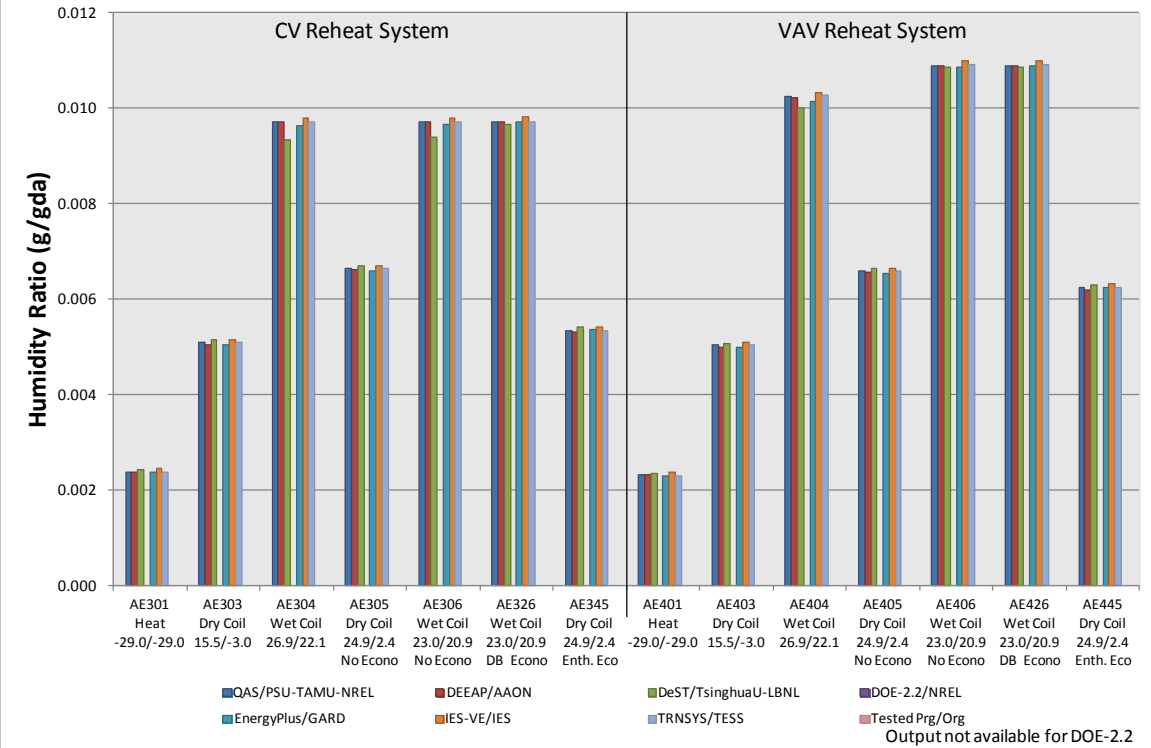
Airside HVAC BESTEST  
 Example Results for Section 4.4, Constant Volume and Variable Air Volume Systems

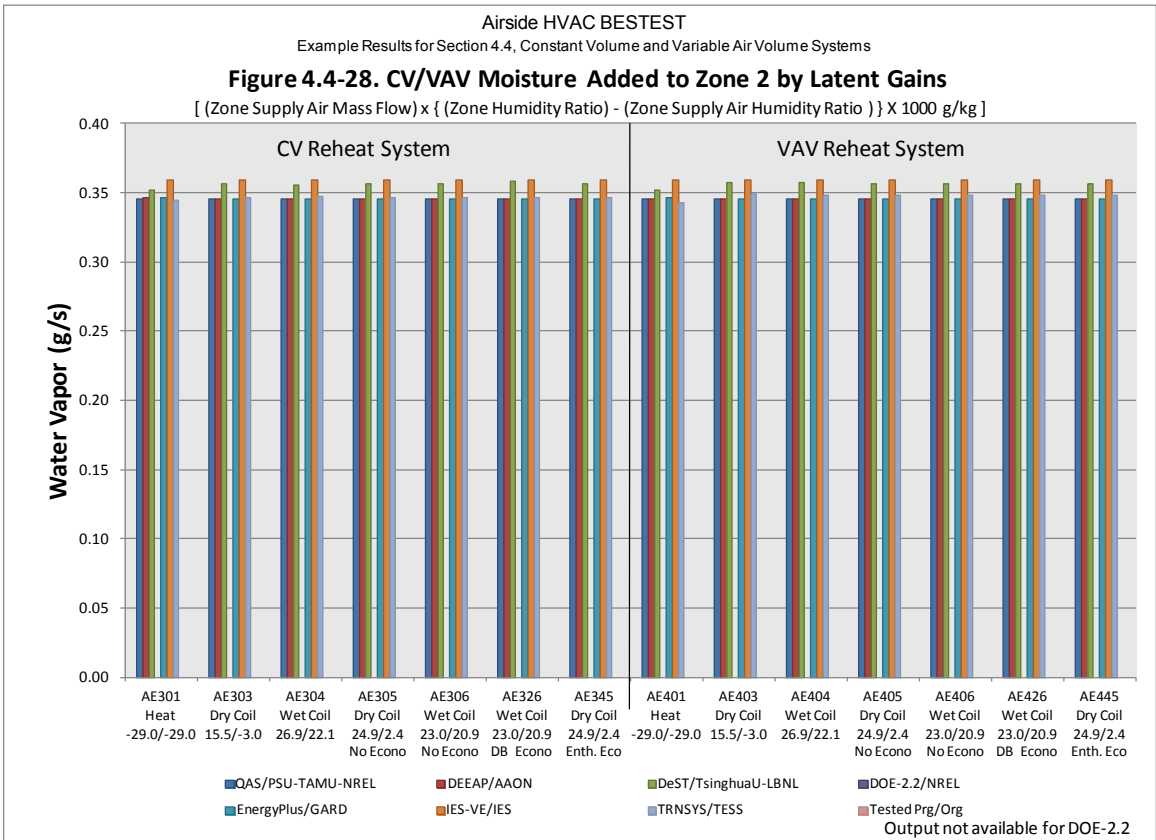
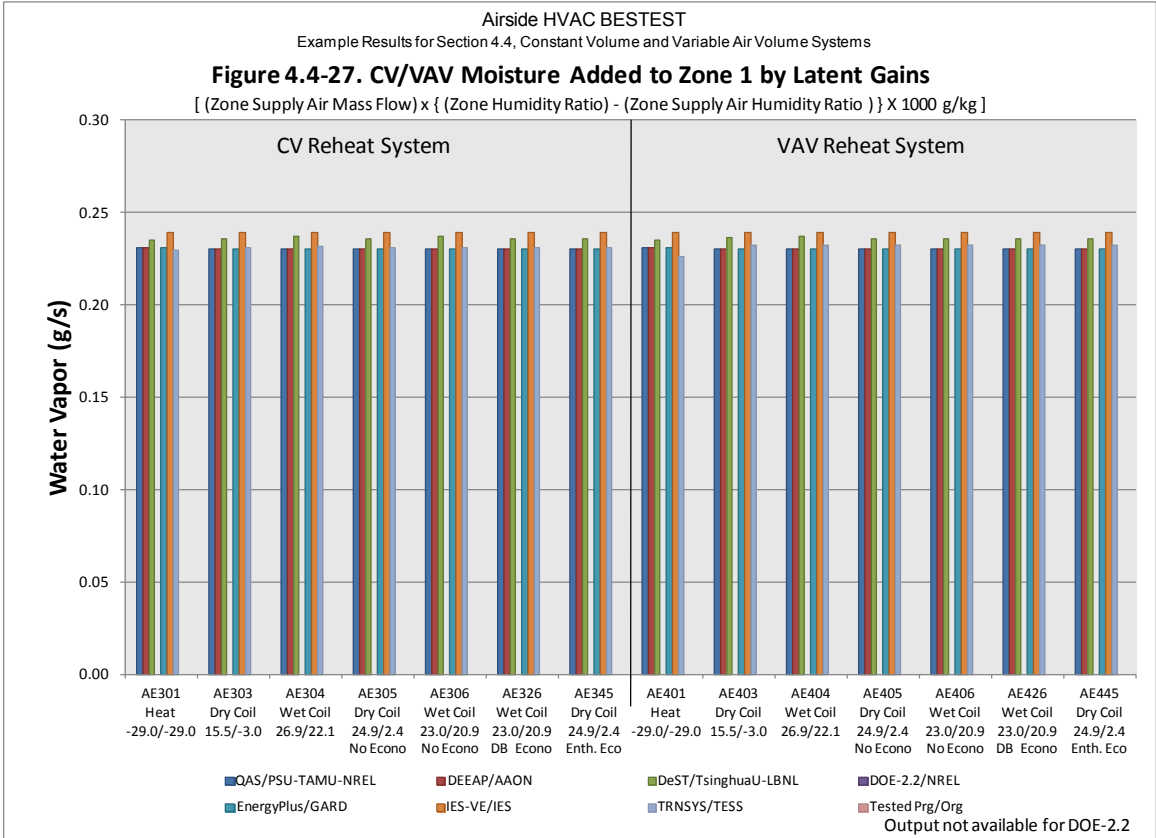
**Figure 4.4-25. CV/VAV Zone 2 Air Temperature**

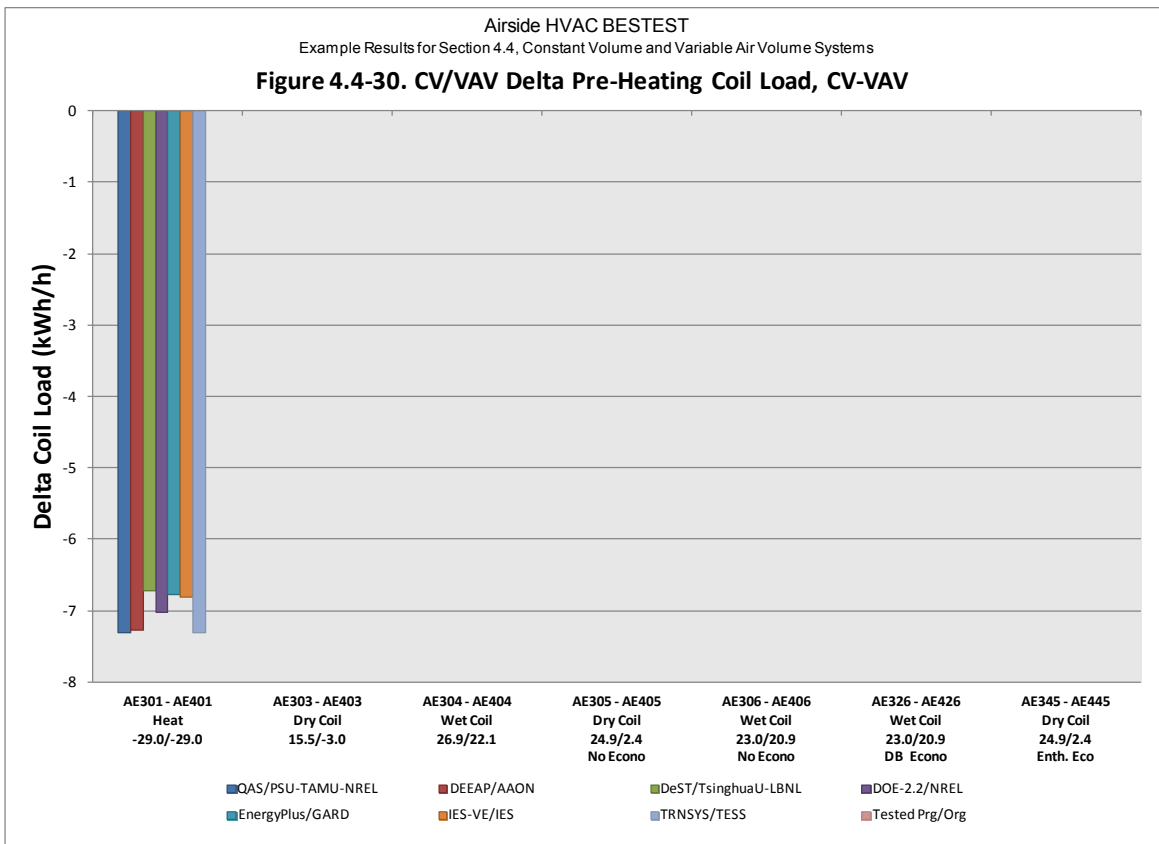
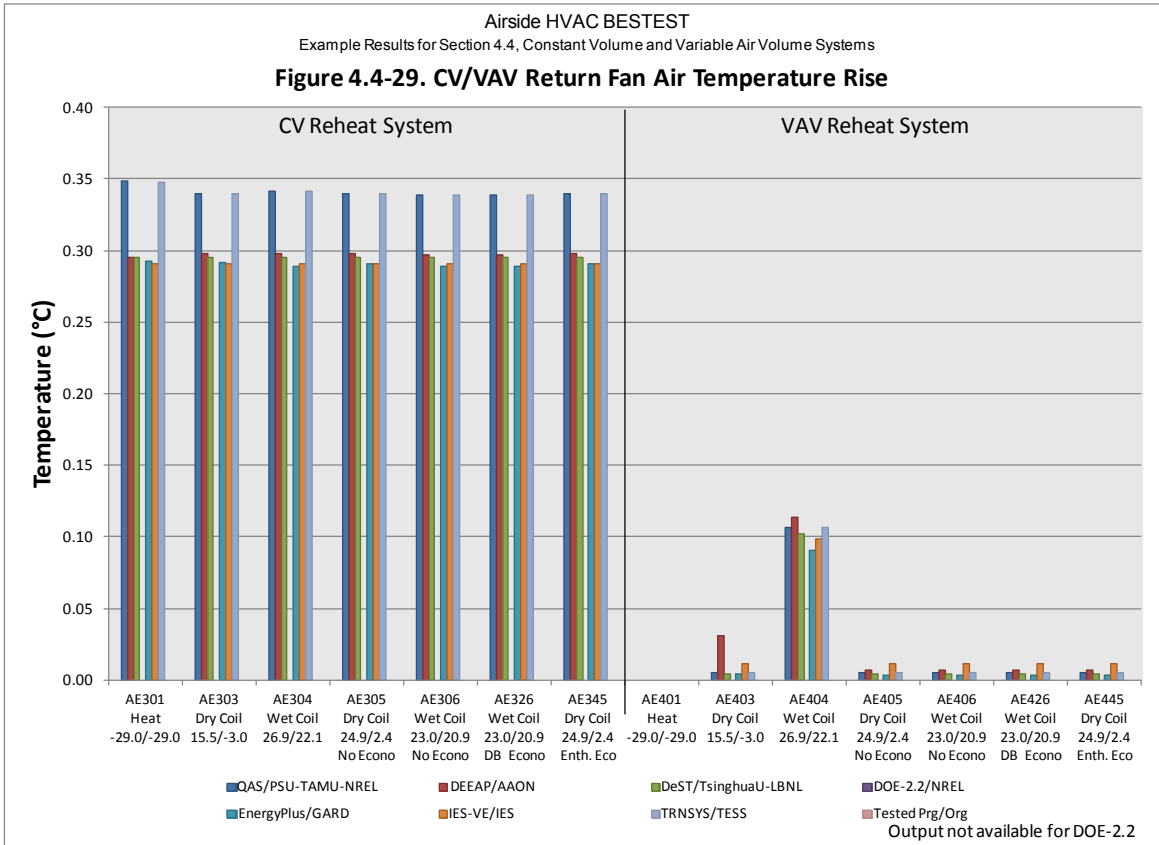


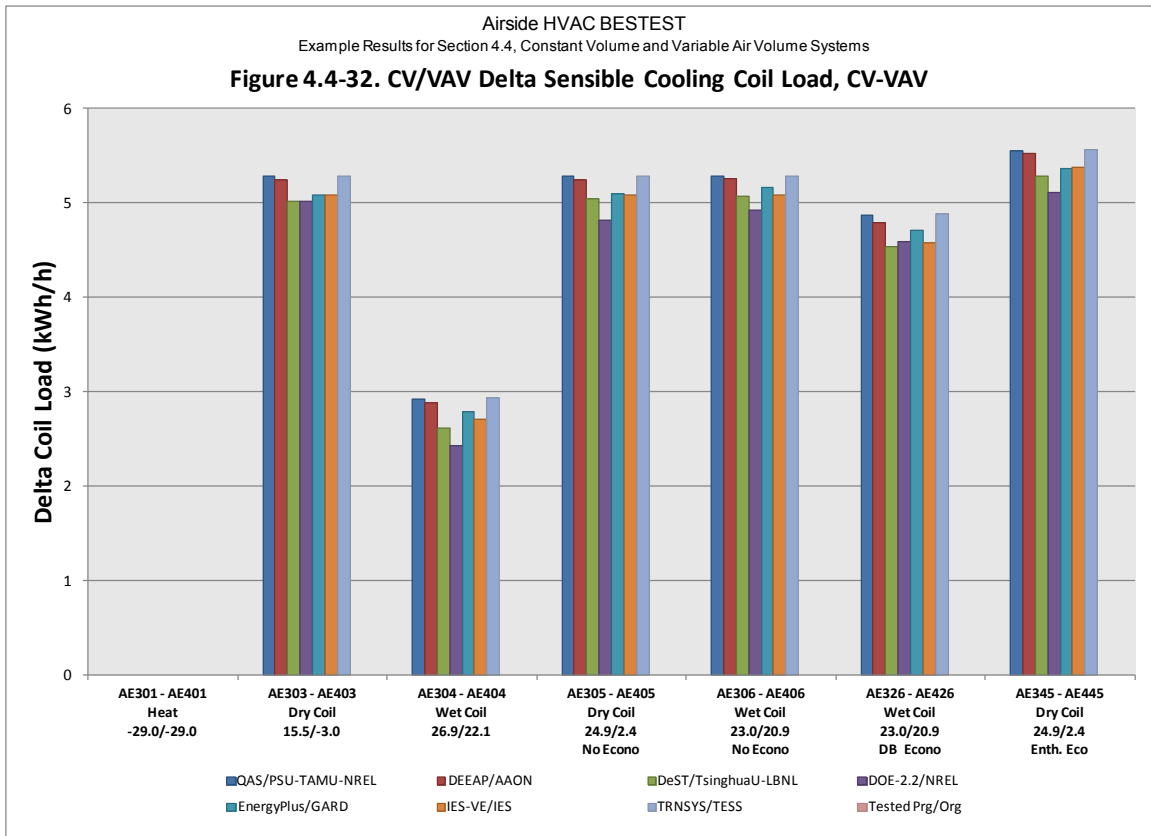
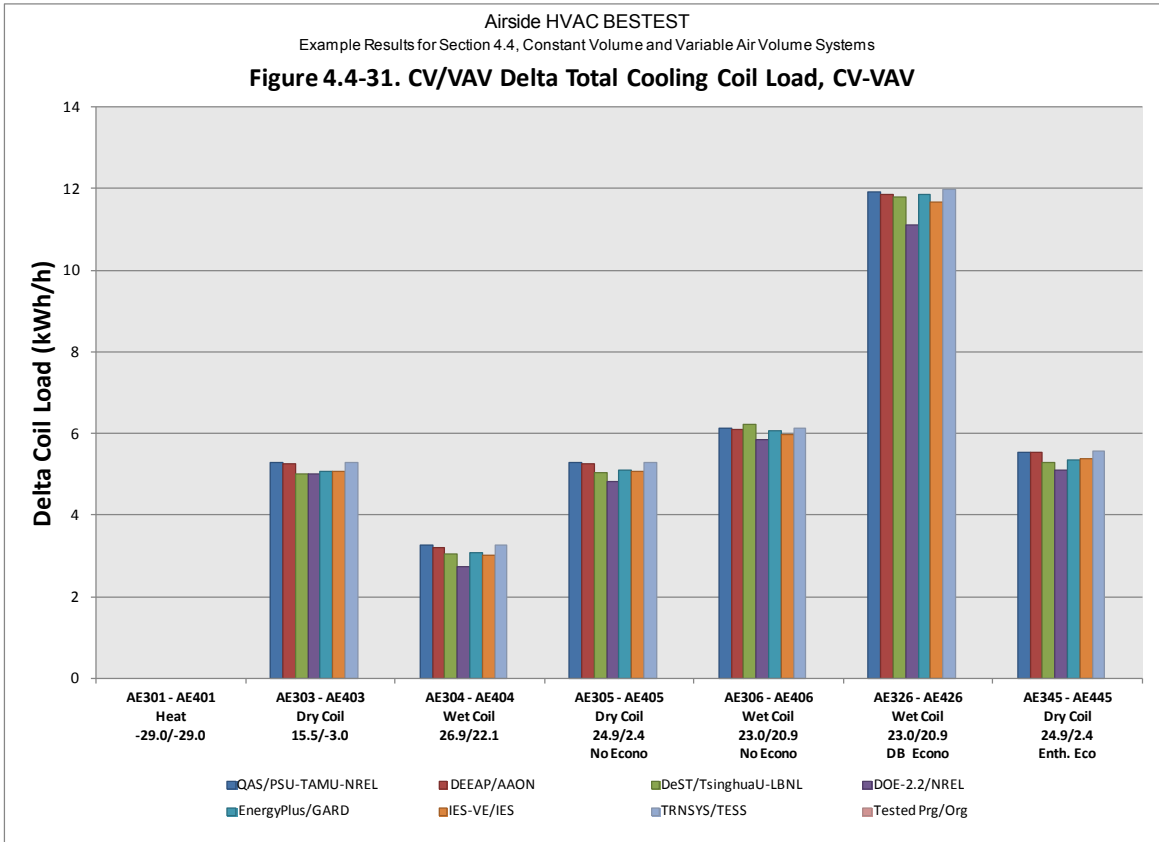
Airside HVAC BESTEST  
 Example Results for Section 4.4, Constant Volume and Variable Air Volume Systems

**Figure 4.4-26. CV/VAV Zone 2 Humidity Ratio**





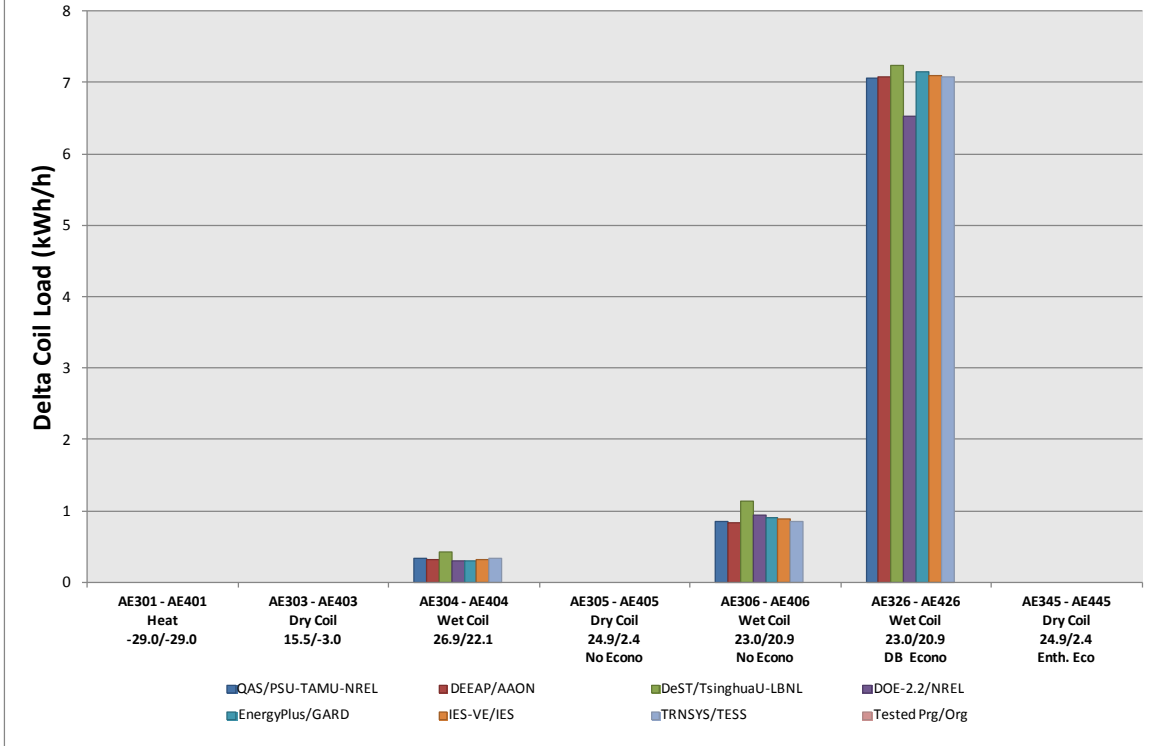




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Example Results for Section 4.4, Constant Volume and Variable Air Volume Systems

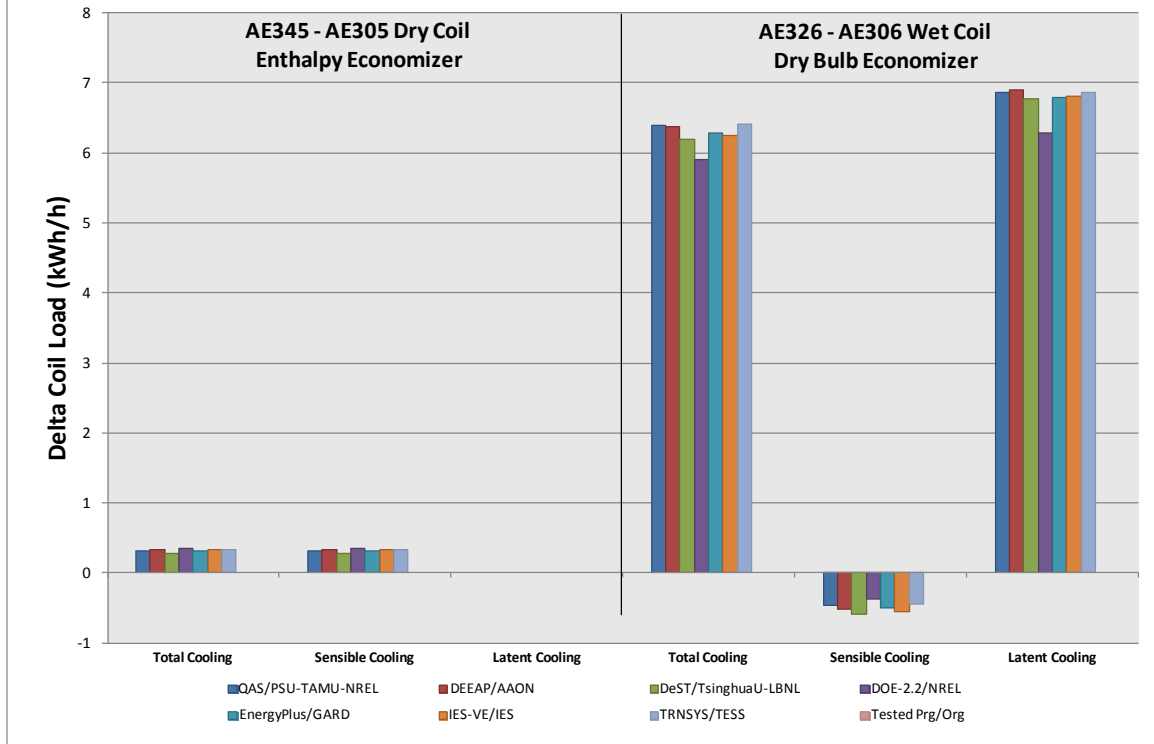
Figure 4.4-33. CV/VAV Delta Latent Cooling Coil Load, CV-VAV



Airside HVAC BESTEST

Example Results for Section 4.4, Constant Volume and Variable Air Volume Systems

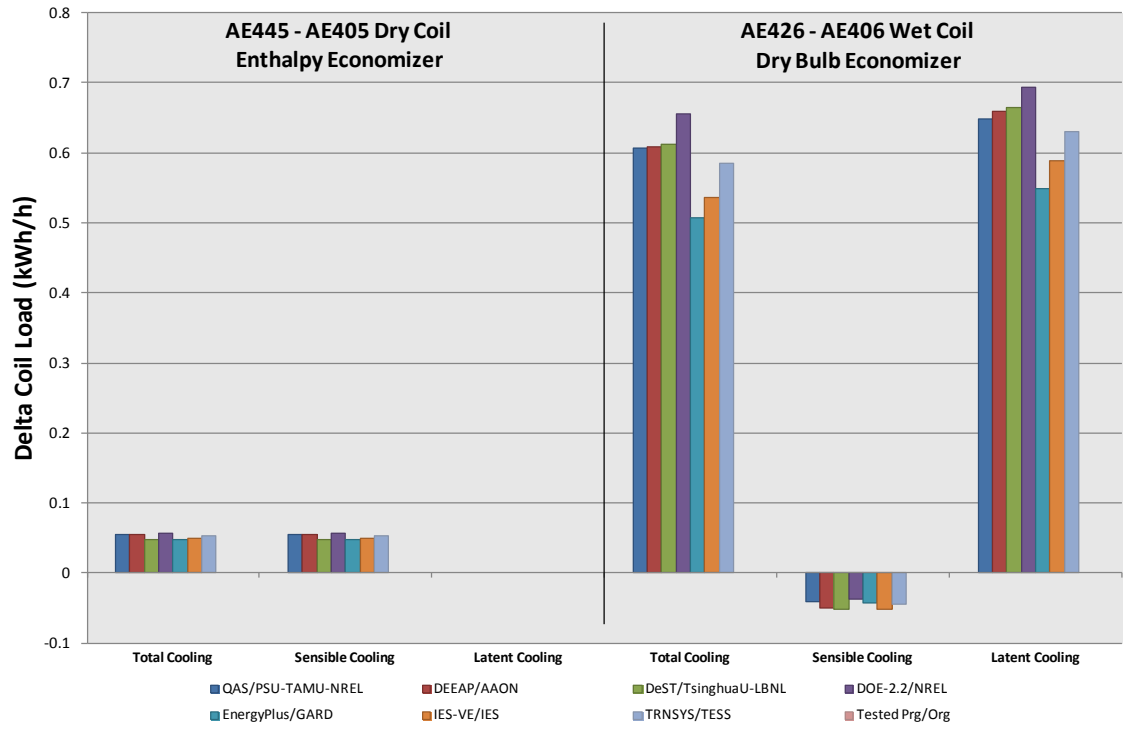
Figure 4.4-34. CV Delta Cooling Coil Load, Economizer Operation



Airside HVAC BESTEST

Example Results for Section 4.4, Constant Volume and Variable Air Volume Systems

Figure 4.4-35. VAV Delta Cooling Coil Load, Economizer Operation



**Table 4.4-1. CV/VAV Total Coil Load, Heating + Cooling, Sensible & Latent (kWh/h)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	19.05	19.06	19.13	19.87	19.21	19.36	19.04	
AE303	11.45	11.41	10.90	10.89	11.02	11.04	11.44	
AE304	19.09	19.05	19.28	18.17	19.18	19.00	19.09	
AE305	14.10	14.06	13.62	13.22	13.77	13.75	14.09	
AE306	19.09	19.03	19.24	18.41	19.11	18.96	19.08	
AE326	25.49	25.41	25.43	24.36	25.39	25.21	25.51	
AE345	14.42	14.39	13.91	13.62	14.06	14.08	14.43	
AE401	19.64	19.62	19.70	20.42	19.75	19.90	19.62	
AE403	1.46	1.47	1.44	1.40	1.40	1.40	1.45	
AE404	13.32	13.35	14.03	13.38	13.69	13.65	13.31	
AE405	4.10	4.10	4.14	4.11	4.10	4.11	4.09	
AE406	8.22	8.22	8.60	8.17	8.42	8.44	8.22	
AE426	8.83	8.82	9.21	8.83	8.93	8.97	8.80	
AE445	4.16	4.16	4.19	4.17	4.15	4.16	4.14	

**Table 4.4-2. CV/VAV Total Sensible Coil Load, Heating + Cooling (kWh/h)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	19.05	19.06	19.13	19.87	19.21	19.36	19.04	
AE303	11.45	11.41	10.90	10.89	11.02	11.04	11.44	
AE304	12.83	12.76	12.27	11.96	12.55	12.37	12.83	
AE305	14.10	14.06	13.62	13.22	13.77	13.75	14.09	
AE306	13.65	13.58	13.14	12.98	13.36	13.21	13.64	
AE326	13.19	13.07	12.55	12.65	12.86	12.65	13.21	
AE345	14.42	14.39	13.91	13.62	14.06	14.08	14.43	
AE401	19.64	19.62	19.70	20.42	19.75	19.90	19.62	
AE403	1.46	1.47	1.44	1.40	1.40	1.40	1.45	
AE404	7.40	7.37	7.45	7.46	7.37	7.33	7.39	
AE405	4.10	4.10	4.14	4.11	4.10	4.11	4.09	
AE406	3.64	3.61	3.63	3.68	3.58	3.57	3.63	
AE426	3.59	3.56	3.57	3.64	3.54	3.52	3.58	
AE445	4.16	4.16	4.19	4.17	4.15	4.16	4.14	

**Table 4.4-3. CV/VAV Pre-Heating Coil Load [QHpreheat] (kWh/h)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	2.95	2.97	3.58	3.79	3.57	3.65	2.95	
AE303	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE304	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE305	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE306	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE326	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE345	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE401	10.26	10.25	10.31	10.82	10.34	10.45	10.26	
AE403	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE404	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE405	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE406	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE426	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE445	0.00	0.00	0.00	0.00	0.00	0.00	0.00	



**Table 4.4-4. CV/VAV Total Cooling Coil Load [QCtotal] (kWh/h)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE303	6.74	6.71	6.46	6.42	6.48	6.48	6.73	
AE304	16.58	16.55	17.07	16.10	16.78	16.66	16.58	
AE305	9.38	9.35	9.18	8.93	9.20	9.20	9.37	
AE306	14.36	14.31	14.80	14.03	14.49	14.40	14.36	
AE326	20.76	20.69	20.99	19.94	20.78	20.65	20.78	
AE345	9.70	9.68	9.47	9.29	9.51	9.53	9.71	
AE401	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE403	1.46	1.47	1.44	1.40	1.40	1.40	1.45	
AE404	13.32	13.35	14.03	13.38	13.69	13.65	13.31	
AE405	4.10	4.10	4.14	4.11	4.10	4.11	4.09	
AE406	8.22	8.22	8.60	8.17	8.42	8.44	8.22	
AE426	8.83	8.82	9.21	8.83	8.93	8.97	8.80	
AE445	4.16	4.16	4.19	4.17	4.15	4.16	4.14	

**Table 4.4-5. CV/VAV Sensible Cooling Coil Load [QC\_sensible] (kWh/h)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE303	6.74	6.71	6.46	6.42	6.48	6.48	6.73	
AE304	10.32	10.26	10.06	9.89	10.16	10.03	10.31	
AE305	9.38	9.35	9.18	8.93	9.20	9.20	9.37	
AE306	8.92	8.87	8.70	8.60	8.74	8.65	8.91	
AE326	8.46	8.35	8.11	8.23	8.25	8.10	8.47	
AE345	9.70	9.68	9.47	9.29	9.51	9.53	9.71	
AE401	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE403	1.46	1.47	1.44	1.40	1.40	1.40	1.45	
AE404	7.40	7.37	7.45	7.46	7.37	7.33	7.39	
AE405	4.10	4.10	4.14	4.11	4.10	4.11	4.09	
AE406	3.64	3.61	3.63	3.68	3.58	3.57	3.63	
AE426	3.59	3.56	3.57	3.64	3.54	3.52	3.58	
AE445	4.16	4.16	4.19	4.17	4.15	4.16	4.14	

**Table 4.4-6. CV/VAV Latent Cooling Coil Load [QC\_latent] (kWh/h)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE303	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE304	6.26	6.29	7.01	6.21	6.62	6.63	6.26	
AE305	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE306	5.44	5.44	6.10	5.43	5.75	5.75	5.44	
AE326	12.30	12.34	12.88	11.71	12.53	12.56	12.31	
AE345	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE401	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE403	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE404	5.92	5.97	6.58	5.92	6.32	6.31	5.92	
AE405	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE406	4.59	4.61	4.97	4.49	4.84	4.87	4.59	
AE426	5.24	5.27	5.63	5.19	5.39	5.46	5.22	
AE445	0.00	0.00	0.00	0.00	0.00	0.00	0.00	

**Table 4.4-7. CV/VAV Zone 1 Reheat Coil Load [QH1reheat] (kWh/h)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	7.71	7.71	7.47	7.71	7.51	7.54	7.71	
AE303	2.26	2.25	2.14	2.15	2.18	2.19	2.26	
AE304	0.99	0.99	0.86	0.80	0.94	0.92	0.99	
AE305	2.26	2.26	2.14	2.07	2.19	2.19	2.26	
AE306	2.26	2.26	2.14	2.11	2.21	2.19	2.26	
AE326	2.26	2.26	2.14	2.13	2.21	2.19	2.27	
AE345	2.26	2.25	2.14	2.09	2.18	2.19	2.26	
AE401	4.50	4.50	4.50	4.58	4.50	4.52	4.49	
AE403	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE404	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE405	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE406	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE426	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE445	0.00	0.00	0.00	0.00	0.00	0.00	0.00	

**Table 4.4-8. CV/VAV Zone 2 Reheat Coil Load [QH2reheat] (kWh/h)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	8.39	8.38	8.08	8.38	8.13	8.17	8.38	
AE303	2.46	2.45	2.30	2.32	2.36	2.37	2.45	
AE304	1.52	1.51	1.35	1.27	1.46	1.42	1.52	
AE305	2.46	2.45	2.30	2.22	2.37	2.37	2.46	
AE306	2.46	2.46	2.30	2.27	2.40	2.37	2.46	
AE326	2.46	2.46	2.30	2.29	2.40	2.37	2.47	
AE345	2.46	2.45	2.30	2.24	2.36	2.37	2.46	
AE401	4.88	4.87	4.89	5.01	4.90	4.93	4.87	
AE403	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE404	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE405	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE406	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE426	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE445	0.00	0.00	0.00	0.00	0.00	0.00	0.00	

**Table 4.4-9. CV/VAV Outdoor Air Temperature (°F)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	-29.00	-29.00	-29.00	-29.00	-29.00	-29.00	-29.00	
AE303	15.50	15.50	15.50	15.50	15.50	15.50	15.50	
AE304	26.90	26.89	26.90	26.89	26.90	26.90	26.90	
AE305	24.90	24.89	24.90	24.89	24.90	24.90	24.90	
AE306	23.00	23.00	23.00	23.00	23.00	23.00	23.00	
AE326	23.00	23.00	23.00	23.00	23.00	23.00	23.00	
AE345	24.90	24.89	24.90	24.89	24.90	24.90	24.90	
AE401	-29.00	-29.00	-29.00	-29.00	-29.00	-29.00	-29.00	
AE403	15.50	15.50	15.50	15.50	15.50	15.50	15.50	
AE404	26.90	26.89	26.90	26.89	26.90	26.90	26.90	
AE405	24.90	24.89	24.90	24.89	24.90	24.90	24.90	
AE406	23.00	23.00	23.00	23.00	23.00	23.00	23.00	
AE426	23.00	23.00	23.00	23.00	23.00	23.00	23.00	
AE445	24.90	24.89	24.90	24.89	24.90	24.90	24.90	

**Table 4.4-10. CV/VAV Outdoor Air Humidity Ratio (g/gda)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.000259	0.000259	0.000260	0.000300	0.000259	0.000262	0.000259	
AE303	0.002936	0.002895	0.002948	0.002900	0.002933	0.002947	0.002936	
AE304	0.016774	0.016783	0.016850	0.016800	0.016772	0.016839	0.016774	
AE305	0.004491	0.004457	0.004510	0.004500	0.004481	0.004508	0.004491	
AE306	0.015556	0.015523	0.015630	0.015600	0.015565	0.015616	0.015556	
AE326	0.015556	0.015523	0.015630	0.015600	0.015565	0.015616	0.015556	
AE345	0.004491	0.004457	0.004510	0.004500	0.004481	0.004508	0.004491	
AE401	0.000259	0.000259	0.000260	0.000300	0.000259	0.000262	0.000259	
AE403	0.002936	0.002895	0.002948	0.002900	0.002933	0.002947	0.002936	
AE404	0.016774	0.016783	0.016850	0.016800	0.016772	0.016839	0.016774	
AE405	0.004491	0.004457	0.004510	0.004500	0.004481	0.004508	0.004491	
AE406	0.015556	0.015523	0.015630	0.015600	0.015565	0.015616	0.015556	
AE426	0.015556	0.015523	0.015630	0.015600	0.015565	0.015616	0.015556	
AE445	0.004491	0.004457	0.004510	0.004500	0.004481	0.004508	0.004491	

**Table 4.4-11. CV/VAV Outdoor Air Mass Flow Rate (kgda/s)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.2814	0.2814	0.2832	0.2965	0.2842	0.2832	0.2814	
AE303	0.2781	0.2781	0.2832	0.2815	0.2842	0.2832	0.2781	
AE304	0.2755	0.2756	0.2832	0.2735	0.2842	0.2832	0.2755	
AE305	0.2774	0.2774	0.2832	0.2774	0.2842	0.2832	0.2774	
AE306	0.2761	0.2761	0.2832	0.2754	0.2842	0.2832	0.2760	
AE326	0.7483	0.7484	0.7362	0.7137	0.7389	0.7362	0.7483	
AE345	0.7535	0.7537	0.7362	0.7217	0.7389	0.7362	0.7535	
AE401	0.2814	0.2814	0.2832	0.2971	0.2842	0.2832	0.2814	
AE403	0.2781	0.2781	0.2832	0.2860	0.2842	0.2832	0.2781	
AE404	0.2753	0.2754	0.2832	0.2729	0.2842	0.2832	0.2753	
AE405	0.2774	0.2775	0.2832	0.2776	0.2842	0.2832	0.2774	
AE406	0.2755	0.2756	0.2832	0.2747	0.2842	0.2832	0.2755	
AE426	0.3321	0.3325	0.3384	0.3294	0.3314	0.3335	0.3313	
AE445	0.3350	0.3354	0.3383	0.3359	0.3343	0.3335	0.3341	

**Table 4.4-12. CV/VAV Cooling Coil Outlet Air Temperature (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	7.22	7.22	7.22	7.22	7.22	7.22	7.22	
AE303	12.21	12.21	12.20	12.17	12.20	12.20	12.21	
AE304	12.21	12.21	12.20	12.22	12.20	12.20	12.21	
AE305	12.21	12.21	12.20	12.28	12.20	12.20	12.21	
AE306	12.21	12.21	12.18	12.22	12.20	12.20	12.21	
AE326	12.21	12.21	12.20	12.22	12.20	12.20	12.20	
AE345	12.21	12.21	12.20	12.22	12.20	12.20	12.20	
AE401	7.22	7.22	7.22	7.22	7.20	7.22	7.22	
AE403	12.66	12.65	12.67	12.67	12.66	12.66	12.67	
AE404	12.48	12.46	12.46	12.44	12.47	12.47	12.49	
AE405	12.66	12.65	12.67	12.67	12.66	12.66	12.67	
AE406	12.66	12.65	12.67	12.67	12.66	12.66	12.66	
AE426	12.66	12.65	12.66	12.67	12.66	12.66	12.67	
AE445	12.66	12.65	12.67	12.67	12.66	12.66	12.67	

**Table 4.4-13. CV/VAV Cooling Coil Outlet Relative Humidity [RHcco] (%)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	24.73	24.73		23.96	23.89	24.65	24.64	
AE303	48.15	47.69	48.22	47.92	47.46	48.02	48.18	
AE304	100.00	100.00	95.49	100.00	99.13	100.00	100.00	
AE305	65.72	65.34	65.84	65.53	64.91	65.57	65.75	
AE306	100.00	100.00	96.21	100.00	99.25	100.00	100.00	
AE326	100.00	100.00	98.89	100.00	99.93	100.00	100.00	
AE345	51.09	50.71	51.33	51.14	50.99	51.10	51.11	
AE401	4.14	4.15		4.80	4.15	4.17	4.15	
AE403	36.39	36.01	36.39	34.29	35.85	35.98	36.37	
AE404	100.00	100.00	97.24	100.00	98.71	100.00	100.00	
AE405	53.46	53.17	53.52	51.85	52.78	53.05	53.43	
AE406	100.00	100.00	99.38	100.00	99.76	100.00	100.00	
AE426	100.00	100.00	99.58	100.00	99.95	100.00	100.00	
AE445	49.58	49.24	49.78	49.66	49.46	49.58	49.56	

**Table 4.4-14. CV/VAV Supply Fan Air Temperature Rise (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.56	0.56	0.59	0.56	0.59	0.58	0.56	
AE303	0.57	0.57	0.59	0.61	0.58	0.58	0.57	
AE304	0.57	0.57	0.59	0.61	0.58	0.58	0.57	
AE305	0.57	0.57	0.59	0.61	0.58	0.58	0.57	
AE306	0.57	0.57	0.59	0.56	0.58	0.58	0.57	
AE326	0.57	0.57	0.59	0.56	0.58	0.58	0.57	
AE345	0.57	0.57	0.59	0.56	0.58	0.58	0.57	
AE401	0.07	0.09	0.09	0.06	0.09	0.09	0.07	
AE403	0.11	0.13	0.13	0.11	0.12	0.12	0.11	
AE404	0.30	0.32	0.33	0.33	0.31	0.31	0.29	
AE405	0.11	0.13	0.12	0.11	0.12	0.12	0.11	
AE406	0.11	0.13	0.12	0.11	0.12	0.12	0.11	
AE426	0.11	0.13	0.12	0.11	0.12	0.12	0.11	
AE445	0.11	0.13	0.12	0.11	0.12	0.12	0.11	

**Table 4.4-15. CV/VAV Supply Air Temperature (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	7.78	7.78	7.81	7.78	7.81	7.80	7.78	
AE303	12.78	12.78	12.79	12.78	12.78	12.78	12.79	
AE304	12.78	12.78	12.79	12.83	12.78	12.78	12.78	
AE305	12.78	12.78	12.79	12.89	12.78	12.78	12.78	
AE306	12.78	12.78	12.77	12.78	12.78	12.78	12.78	
AE326	12.78	12.78	12.79	12.78	12.78	12.78	12.77	
AE345	12.78	12.78	12.79	12.78	12.78	12.78	12.77	
AE401	7.30	7.31	7.31	7.28	7.29	7.31	7.29	
AE403	12.78	12.78	12.79	12.78	12.78	12.78	12.78	
AE404	12.78	12.78	12.79	12.78	12.78	12.78	12.78	
AE405	12.78	12.78	12.79	12.78	12.78	12.78	12.78	
AE406	12.78	12.78	12.79	12.78	12.78	12.78	12.77	
AE426	12.78	12.78	12.78	12.78	12.78	12.78	12.78	
AE445	12.78	12.78	12.79	12.78	12.78	12.78	12.78	

**Table 4.4-16. CV/VAV Supply Air Humidity Ratio (g/gda)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.001548	0.001548	0.001534	0.001500	0.001496	0.001549	0.001542	
AE303	0.004231	0.004190	0.004235	0.004200	0.004168	0.004234	0.004236	
AE304	0.008852	0.008852	0.008443	0.008900	0.008773	0.008891	0.008854	
AE305	0.005789	0.005755	0.005797	0.005800	0.005716	0.005795	0.005794	
AE306	0.008852	0.008852	0.008492	0.008900	0.008784	0.008888	0.008856	
AE326	0.008852	0.008852	0.008747	0.008900	0.008845	0.008909	0.008848	
AE345	0.004491	0.004457	0.004510	0.004500	0.004481	0.004508	0.004491	
AE401	0.000259	0.000259	0.000260	0.000300	0.000259	0.000262	0.000259	
AE403	0.003291	0.003252	0.003288	0.003100	0.003241	0.003265	0.003289	
AE404	0.009015	0.009004	0.008749	0.009000	0.008894	0.009056	0.009017	
AE405	0.004846	0.004814	0.004851	0.004700	0.004784	0.004826	0.004835	
AE406	0.009126	0.009116	0.009068	0.009200	0.009106	0.009175	0.009122	
AE426	0.009126	0.009116	0.009080	0.009200	0.009124	0.009184	0.009129	
AE445	0.004491	0.004457	0.004510	0.004500	0.004481	0.004508	0.004491	

**Table 4.4-17. CV/VAV Supply Air Specific Volume (L/kgda)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	797.83	797.73	833.33		830.35		793.92	
AE303	815.50	815.34	833.33		830.35		811.55	
AE304	821.52	821.42	833.33		830.35		817.53	
AE305	817.53	817.38	833.33		830.35		813.57	
AE306	821.52	821.42	833.33		830.35		817.53	
AE326	821.52	821.42	833.33		830.35		817.48	
AE345	815.84	815.69	833.33		830.35		811.84	
AE401	794.80	794.73	833.33		830.35		794.28	
AE403	814.28	814.12	833.33		830.35		813.51	
AE404	821.73	821.61	833.33		830.35		819.66	
AE405	816.31	816.16	833.33		830.35		815.52	
AE406	821.88	821.76	833.33		830.35		821.07	
AE426	821.88	821.76	833.33		830.35		821.11	
AE445	815.84	815.69	833.33		830.35		815.07	

**Table 4.4-18. CV/VAV Supply Air Enthalpy (J/gda)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	11.73		11.71		11.61		11.71	
AE303	23.54		23.55		23.37		23.56	
AE304	35.20		34.16		34.99		35.21	
AE305	27.47		27.49		27.27		27.49	
AE306	35.20		34.26		35.02		35.22	
AE326	35.20		34.93		35.17		35.19	
AE345	24.19		24.24		24.16		24.19	
AE401	7.99		8.00		7.98		7.99	
AE403	21.16		21.16		21.02		21.16	
AE404	35.62		34.93		35.30		35.62	
AE405	25.09		25.10		24.92		25.07	
AE406	35.90		35.74		35.83		35.88	
AE426	35.90		35.76		35.88		35.91	
AE445	24.19		24.24		24.16		24.20	

**Table 4.4-19. CV/VAV Supply Air Mass Flow Rate (kgda/s)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.7705	0.7706	0.7362	0.7709	0.7389	0.7362	0.7705	
AE303	0.7538	0.7540	0.7362	0.7319	0.7389	0.7362	0.7538	
AE304	0.7483	0.7484	0.7362	0.7111	0.7389	0.7362	0.7483	
AE305	0.7520	0.7521	0.7362	0.7213	0.7389	0.7362	0.7519	
AE306	0.7483	0.7484	0.7362	0.7160	0.7389	0.7362	0.7483	
AE326	0.7483	0.7484	0.7362	0.7137	0.7389	0.7362	0.7483	
AE345	0.7535	0.7537	0.7362	0.7217	0.7389	0.7362	0.7535	
AE401	0.2814	0.2814	0.2832	0.2971	0.2842	0.2832	0.2814	
AE403	0.3357	0.3361	0.3385	0.3369	0.3351	0.3335	0.3350	
AE404	0.5392	0.5399	0.5492	0.5354	0.5384	0.5414	0.5388	
AE405	0.3347	0.3352	0.3383	0.3359	0.3342	0.3335	0.3339	
AE406	0.3321	0.3325	0.3384	0.3297	0.3315	0.3335	0.3314	
AE426	0.3321	0.3325	0.3384	0.3294	0.3314	0.3335	0.3313	
AE445	0.3350	0.3354	0.3383	0.3359	0.3343	0.3335	0.3341	

**Table 4.4-20. CV/VAV Outdoor Air Mass Flow to Supply Air Mass Flow Ratio (fraction)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.365	0.365	0.385	0.385	0.385	0.385	0.365	
AE303	0.369	0.369	0.385	0.385	0.385	0.385	0.369	
AE304	0.368	0.368	0.385	0.385	0.385	0.385	0.368	
AE305	0.369	0.369	0.385	0.385	0.385	0.385	0.369	
AE306	0.369	0.369	0.385	0.385	0.385	0.385	0.369	
AE326	1.000	1.000	1.000	1.000	1.000	1.000	1.000	
AE345	1.000	1.000	1.000	1.000	1.000	1.000	1.000	
AE401	1.000	1.000	1.000	1.000	1.000	1.000	1.000	
AE403	0.828	0.828	0.837	0.849	0.848	0.849	0.830	
AE404	0.511	0.510	0.516	0.510	0.528	0.523	0.511	
AE405	0.829	0.828	0.837	0.826	0.850	0.849	0.831	
AE406	0.830	0.829	0.837	0.833	0.857	0.849	0.831	
AE426	1.000	1.000	1.000	1.000	1.000	1.000	1.000	
AE445	1.000	1.000	1.000	1.000	1.000	1.000	1.000	

**Table 4.4-21. CV/VAV Zone 1 Supply Air Temperature (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	29.28	29.29	29.67		29.65	29.58	29.27	
AE303	19.18	19.18	19.05		19.10	19.10	19.18	
AE304	15.59	15.58	15.31		15.48	15.42	15.59	
AE305	19.18	19.18	19.05		19.11	19.10	19.18	
AE306	19.18	19.18	19.02		19.14	19.10	19.19	
AE326	19.18	19.18	19.05		19.14	19.10	19.19	
AE345	19.18	19.18	19.05		19.10	19.10	19.18	
AE401	46.92	46.95	46.85		46.70	46.50	46.83	
AE403	12.78	12.78	12.79		12.78	12.78	12.78	
AE404	12.78	12.78	12.79		12.78	12.78	12.79	
AE405	12.78	12.78	12.79		12.78	12.78	12.78	
AE406	12.78	12.78	12.79		12.79	12.78	12.77	
AE426	12.78	12.78	12.78		12.79	12.78	12.78	
AE445	12.78	12.78	12.79		12.78	12.78	12.78	

**Table 4.4-22. CV/VAV Zone 1 Air Temperature (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	21.11	21.11	21.11	21.11	21.13	21.11	21.11	
AE303	23.33	23.33	23.33	23.33	23.34	23.33	23.33	
AE304	23.89	23.89	23.89	23.89	23.89	23.89	23.88	
AE305	23.33	23.33	23.33	23.33	23.34	23.33	23.33	
AE306	23.33	23.33	23.33	23.33	23.34	23.33	23.33	
AE326	23.33	23.33	23.33	23.33	23.34	23.33	23.33	
AE345	23.33	23.33	23.33	23.33	23.34	23.33	23.33	
AE401	21.11	21.11	21.11	21.11	21.15	21.11	21.12	
AE403	23.33	23.33	23.33	23.33	23.34	23.33	23.33	
AE404	23.89	23.89	23.89	23.89	23.90	23.89	23.88	
AE405	23.33	23.33	23.33	23.33	23.34	23.33	23.33	
AE406	23.33	23.33	23.33	23.33	23.34	23.33	23.32	
AE426	23.33	23.33	23.33	23.33	23.34	23.33	23.33	
AE445	23.33	23.33	23.33	23.33	23.34	23.33	23.33	

**Table 4.4-23. CV/VAV Zone 1 Humidity Ratio (g/gda)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.002197	0.002197	0.002225		0.002172	0.002253	0.002187	
AE303	0.004893	0.004852	0.004930		0.004844	0.004938	0.004900	
AE304	0.009518	0.009519	0.009140		0.009448	0.009595	0.009525	
AE305	0.006453	0.006419	0.006492		0.006391	0.006499	0.006460	
AE306	0.009519	0.009519	0.009190		0.009459	0.009592	0.009525	
AE326	0.009519	0.009519	0.009442		0.009520	0.009613	0.009517	
AE345	0.005153	0.005119	0.005204		0.005156	0.005212	0.005155	
AE401	0.002304	0.002303	0.002332		0.002289	0.002374	0.002269	
AE403	0.004970	0.004930	0.004996		0.004925	0.005021	0.004988	
AE404	0.009908	0.009895	0.009651		0.009788	0.009980	0.009921	
AE405	0.006530	0.006497	0.006557		0.006473	0.006582	0.006537	
AE406	0.010823	0.010812	0.010775		0.010803	0.010931	0.010838	
AE426	0.010823	0.010812	0.010787		0.010825	0.010940	0.010844	
AE445	0.006174	0.006139	0.006216		0.006169	0.006264	0.006192	

**Table 4.4-24. CV/VAV Zone 2 Supply Air Temperature (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	27.82	27.83	28.08		28.09	28.03	27.81	
AE303	18.75	18.74	18.57		18.64	18.64	18.75	
AE304	16.46	16.46	16.19		16.36	16.29	16.48	
AE305	18.75	18.74	18.57		18.66	18.64	18.75	
AE306	18.75	18.75	18.54		18.69	18.64	18.76	
AE326	18.75	18.75	18.57		18.69	18.64	18.76	
AE345	18.75	18.74	18.57		18.64	18.64	18.75	
AE401	36.04	36.05	35.95		35.89	35.77	35.99	
AE403	12.78	12.78	12.79		12.78	12.78	12.78	
AE404	12.78	12.78	12.79		12.78	12.78	12.79	
AE405	12.78	12.78	12.79		12.78	12.78	12.78	
AE406	12.78	12.78	12.79		12.79	12.78	12.77	
AE426	12.78	12.78	12.78		12.79	12.78	12.78	
AE445	12.78	12.78	12.79		12.78	12.78	12.78	

**Table 4.4-25. CV/VAV Zone 2 Air Temperature (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	22.22	22.22	22.22	22.22	22.25	22.22	22.22	
AE303	24.44	24.44	24.44	24.44	24.45	24.44	24.44	
AE304	25.00	25.00	25.00	25.00	25.00	25.00	25.00	
AE305	24.44	24.44	24.44	24.44	24.45	24.44	24.44	
AE306	24.44	24.44	24.44	24.44	24.45	24.44	24.44	
AE326	24.44	24.44	24.44	24.44	24.45	24.44	24.44	
AE345	24.44	24.44	24.44	24.44	24.45	24.44	24.44	
AE401	22.22	22.22	22.22	22.22	22.26	22.22	22.22	
AE403	24.44	24.44	24.44	24.44	24.46	24.44	24.44	
AE404	25.00	25.00	25.00	25.00	25.02	25.00	25.00	
AE405	24.44	24.44	24.44	24.44	24.46	24.44	24.44	
AE406	24.44	24.44	24.44	24.44	24.46	24.44	24.43	
AE426	24.44	24.44	24.44	24.44	24.46	24.44	24.44	
AE445	24.44	24.44	24.44	24.44	24.46	24.44	24.44	

**Table 4.4-26. CV/VAV Zone 2 Humidity Ratio (g/gda)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.002382	0.002382	0.002421		0.002365	0.002454	0.002372	
AE303	0.005081	0.005040	0.005135		0.005036	0.005139	0.005090	
AE304	0.009708	0.009708	0.009340		0.009640	0.009796	0.009716	
AE305	0.006642	0.006608	0.006697		0.006584	0.006700	0.006650	
AE306	0.009709	0.009709	0.009390		0.009652	0.009793	0.009716	
AE326	0.009709	0.009709	0.009650		0.009713	0.009814	0.009708	
AE345	0.005342	0.005308	0.005409		0.005349	0.005413	0.005345	
AE401	0.002310	0.002310	0.002330		0.002287	0.002374	0.002289	
AE403	0.005029	0.004989	0.005071		0.004985	0.005084	0.005049	
AE404	0.010241	0.010229	0.009998		0.010124	0.010327	0.010261	
AE405	0.006590	0.006556	0.006633		0.006533	0.006645	0.006598	
AE406	0.010883	0.010871	0.010849		0.010863	0.010994	0.010900	
AE426	0.010883	0.010871	0.010861		0.010885	0.011003	0.010906	
AE445	0.006234	0.006197	0.006292		0.006229	0.006327	0.006253	

**Table 4.4-27. CV/VAV Moisture Added to Zone 1 by Latent Gains (g/s)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.231	0.231	0.235		0.231	0.239	0.229	
AE303	0.230	0.230	0.236		0.230	0.239	0.231	
AE304	0.230	0.230	0.237		0.230	0.239	0.232	
AE305	0.230	0.230	0.236		0.230	0.239	0.231	
AE306	0.230	0.230	0.237		0.230	0.239	0.231	
AE326	0.230	0.230	0.236		0.230	0.239	0.231	
AE345	0.230	0.230	0.236		0.230	0.239	0.231	
AE401	0.231	0.231	0.235		0.231	0.239	0.226	
AE403	0.230	0.230	0.236		0.230	0.239	0.232	
AE404	0.230	0.230	0.237		0.230	0.239	0.232	
AE405	0.230	0.230	0.236		0.230	0.239	0.232	
AE406	0.230	0.230	0.236		0.230	0.239	0.232	
AE426	0.230	0.230	0.236		0.230	0.239	0.232	
AE445	0.230	0.230	0.236		0.230	0.239	0.232	

[ (Zone Supply Air Mass Flow) x { (Zone Humidity Ratio) - (Zone Supply Air Humidity Ratio) } X 1000 g/kg ]



**Table 4.4-28. CV/VAV Moisture Added to Zone 2 by Latent Gains (g/s)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.346	0.346	0.352		0.346	0.359	0.344	
AE303	0.345	0.345	0.356		0.345	0.359	0.347	
AE304	0.345	0.345	0.356		0.345	0.359	0.347	
AE305	0.345	0.345	0.356		0.345	0.359	0.347	
AE306	0.345	0.345	0.356		0.345	0.359	0.347	
AE326	0.345	0.345	0.358		0.345	0.359	0.347	
AE345	0.345	0.345	0.356		0.345	0.359	0.347	
AE401	0.346	0.346	0.352		0.346	0.359	0.342	
AE403	0.345	0.345	0.357		0.345	0.359	0.349	
AE404	0.345	0.345	0.358		0.345	0.359	0.348	
AE405	0.345	0.345	0.356		0.345	0.359	0.348	
AE406	0.345	0.345	0.356		0.345	0.359	0.349	
AE426	0.345	0.345	0.356		0.345	0.359	0.348	
AE445	0.345	0.345	0.357		0.345	0.359	0.348	

[ (Zone Supply Air Mass Flow) x { (Zone Humidity Ratio) - (Zone Supply Air Humidity Ratio) } X 1000 g/kg ]

**Table 4.4-29. CV/VAV Return Fan Air Temperature Rise (°C)**

Test Case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301	0.35	0.30	0.29		0.29	0.29	0.35	
AE303	0.34	0.30	0.29		0.29	0.29	0.34	
AE304	0.34	0.30	0.29		0.29	0.29	0.34	
AE305	0.34	0.30	0.29		0.29	0.29	0.34	
AE306	0.34	0.30	0.29		0.29	0.29	0.34	
AE326	0.34	0.30	0.29		0.29	0.29	0.34	
AE345	0.34	0.30	0.29		0.29	0.29	0.34	
AE401								
AE403	0.00	0.03	0.00		0.00	0.01	0.00	
AE404	0.11	0.11	0.10		0.09	0.10	0.11	
AE405	0.00	0.01	0.00		0.00	0.01	0.00	
AE406	0.00	0.01	0.00		0.00	0.01	0.00	
AE426	0.00	0.01	0.00		0.00	0.01	0.00	
AE445	0.00	0.01	0.00		0.00	0.01	0.00	

**Table 4.4-30. CV/VAV Delta Preheat Coil Load, CV - VAV (kWh/h)**

Test case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301 - AE401	-7.31	-7.28	-6.73	-7.03	-6.77	-6.80	-7.31	
AE303 - AE403	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE304 - AE404	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE305 - AE405	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE306 - AE406	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE326 - AE426	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE345 - AE445	0.00	0.00	0.00	0.00	0.00	0.00	0.00	

**Table 4.4-31. CV/VAV Delta Sensible Cooling Coil Load, CV - VAV (kWh/h)**

Test case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301 - AE401	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE303 - AE403	5.28	5.24	5.02	5.02	5.08	5.08	5.28	
AE304 - AE404	2.92	2.88	2.61	2.43	2.79	2.70	2.93	
AE305 - AE405	5.28	5.24	5.04	4.81	5.10	5.08	5.28	
AE306 - AE406	5.29	5.26	5.07	4.92	5.16	5.09	5.29	
AE326 - AE426	4.87	4.79	4.54	4.59	4.71	4.58	4.89	
AE345 - AE445	5.55	5.52	5.28	5.11	5.36	5.37	5.56	

**Table 4.4-32. CV/VAV Delta Latent Cooling Coil Load, CV - VAV (kWh/h)**

Test case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301 - AE401	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE303 - AE403	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE304 - AE404	0.33	0.32	0.43	0.30	0.30	0.32	0.34	
AE305 - AE405	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE306 - AE406	0.86	0.84	1.13	0.94	0.91	0.88	0.85	
AE326 - AE426	7.07	7.08	7.25	6.52	7.14	7.10	7.09	
AE345 - AE445	0.00	0.00	0.00	0.00	0.00	0.00	0.00	

**Table 4.4-33. CV/VAV Delta Total Cooling Coil Load, CV - VAV (kWh/h)**

Test case	QAS PSU-TAMU-NREL	DEEAP AAON	DeST TsinghuaU-LBNL	DOE-2.2 NREL	EnergyPlus GARD	IES-VE IES	TRNSYS TESS	Tested Prg Org
AE301 - AE401	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE303 - AE403	5.28	5.24	5.02	5.02	5.08	5.08	5.28	
AE304 - AE404	3.26	3.20	3.04	2.73	3.09	3.02	3.27	
AE305 - AE405	5.28	5.24	5.04	4.81	5.10	5.08	5.28	
AE306 - AE406	6.14	6.10	6.21	5.86	6.07	5.97	6.14	
AE326 - AE426	11.93	11.87	11.79	11.11	11.85	11.68	11.97	
AE345 - AE445	5.55	5.52	5.28	5.11	5.36	5.37	5.56	

**Table 4.4-34. CV Delta Cooling Coil Load, Economizer Operation (kWh/h)**

Test case	Load	QAS	DEEAP	DeST	DOE-2.2	EnergyPlus	IES-VE	TRNSYS	Tested Prg Org
		PSU-TAMU-NREL	AAON	TsinghuaU-LBNL	NREL	GARD	IES	TESS	
AE345-AE305	Total Cooling	0.32	0.33	0.29	0.36	0.31	0.33	0.33	
	Sensible Cooling	0.32	0.33	0.29	0.36	0.31	0.33	0.33	
	Latent Cooling	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE326 - AE306	Total Cooling	6.40	6.38	6.19	5.91	6.29	6.25	6.42	
	Sensible Cooling	-0.46	-0.52	-0.58	-0.37	-0.50	-0.56	-0.44	
	Latent Cooling	6.86	6.90	6.78	6.28	6.78	6.81	6.86	

**Table 4.4-35. VAV Delta Cooling Coil Load, Economizer Operation (kWh/h)**

Test case	Load	QAS	DEEAP	DeST	DOE-2.2	EnergyPlus	IES-VE	TRNSYS	Tested Prg Org
		PSU-TAMU-NREL	AAON	TsinghuaU-LBNL	NREL	GARD	IES	TESS	
AE445-AE405	Total Cooling	0.06	0.05	0.05	0.06	0.05	0.05	0.05	
	Sensible Cooling	0.06	0.05	0.05	0.06	0.05	0.05	0.05	
	Latent Cooling	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
AE426 - AE406	Total Cooling	0.61	0.61	0.61	0.66	0.51	0.54	0.59	
	Sensible Cooling	-0.04	-0.05	-0.05	-0.04	-0.04	-0.05	-0.04	
	Latent Cooling	0.65	0.66	0.66	0.69	0.55	0.59	0.63	