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Technical Report

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International Energy Agency Building Energy Simulation Test and Diagnostic Method for Heating, Ventilating, and Air-Conditioning Equipment Models (HVAC BESTEST)

Volume 1: Cases E100–E200

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Preface

Background and Introduction to the International Energy Agency

The International Energy Agency (IEA) was established in 1974 as an autonomous agency within the framework of the Organization for Economic Cooperation and Development (OECD) to carry out a comprehensive program of energy cooperation among its 24 member countries and the Commission of the European Communities.

An important part of the Agency's program involves collaboration in the research, development, and demonstration of new energy technologies to reduce excessive reliance on imported oil, increase long-term energy security, and reduce greenhouse gas emissions. The IEA's R&D activities are headed by the Committee on Energy Research and Technology (CERT) and supported by a small Secretariat staff, headquartered in Paris. In addition, three Working Parties are charged with monitoring the various collaborative energy agreements, identifying new areas for cooperation, and advising the CERT on policy matters.

Collaborative programs in the various energy technology areas are conducted under Implementing Agreements, which are signed by contracting parties (government agencies or entities designated by them). There are currently 40 Implementing Agreements covering fossil fuel technologies, renewable energy technologies, efficient energy end-use technologies, nuclear fusion science and technology, and energy technology information centers.

Solar Heating and Cooling Program

The Solar Heating and Cooling Program was one of the first IEA Implementing Agreements to be established. Since 1977, its 21 members have been collaborating to advance active solar, passive solar, and photovoltaic technologies and their application in buildings.

The members are:

Australia	France	Norway
Austria	Germany	Portugal
Belgium	Italy	Spain
Canada	Japan	Sweden
Denmark	Mexico	Switzerland
European Commission	Netherlands	United Kingdom
Finland	New Zealand	United States

A total of 26 Tasks have been initiated, 17 of which have been completed. Each Task is managed by an Operating Agent from one of the participating countries. Overall control of the program rests with an Executive Committee comprised of one representative from each contracting party to the Implementing Agreement. In addition, a number of special ad hoc activities—working groups, conferences, and workshops—have been organized.

The Tasks of the IEA Solar Heating and Cooling Programme, both completed and current, are as follows:

Completed Tasks:

Task 1	Investigation of the Performance of Solar Heating and Cooling Systems
Task 2	Coordination of Solar Heating and Cooling R&D
Task 3	Performance Testing of Solar Collectors
Task 4	Development of an Insolation Handbook and Instrument Package
Task 5	Use of Existing Meteorological Information for Solar Energy Application
Task 6	Performance of Solar Systems Using Evacuated Collectors
Task 7	Central Solar Heating Plants with Seasonal Storage
Task 8	Passive and Hybrid Solar Low Energy Buildings
Task 9	Solar Radiation and Pyranometry Studies
Task 10	Solar Materials R&D
Task 11	Passive and Hybrid Solar Commercial Buildings
Task 12	Building Energy Analysis and Design Tools for Solar Applications
Task 13	Advanced Solar Low Energy Buildings
Task 14	Advanced Active Solar Energy Systems
Task 16	Photovoltaics in Buildings
Task 17	Measuring and Modeling Spectral Radiation
Task 18	Advanced Glazing and Associated Materials for Solar and Building Applications
Task 19	Solar Air Systems
Task 20	Solar Energy in Building Renovation
Task 21	Daylight in Buildings

Current Tasks and Working Groups:

Task 22	Building Energy Analysis Tools
Task 23	Optimization of Solar Energy Use in Large Buildings
Task 24	Solar Procurement
Task 25	Solar Assisted Cooling Systems for Air Conditioning of Buildings
Task 26	Solar Combisystems Working Group Materials in Solar Thermal Collectors
Task 27	Performance Assessment of Solar Building Envelope Components
Task 28	Solar Sustainable Housing
Task 29	Solar Crop Drying
Task 30	Solar Cities

Task 22: Building Energy Analysis Tools

Goal and Objectives of the Task

The overall goal of Task 22 is to establish a sound technical basis for analyzing solar, low-energy buildings with available and emerging building energy analysis tools. This goal will be pursued by accomplishing the following objectives:

- Develop methods to assess the accuracy of available building energy analysis tools in predicting the performance of widely used solar and low-energy concepts;
- Collect and document engineering models of widely used solar and low-energy concepts for use in the next generation building energy analysis tools;
- Assess and document the impact (value) of improved building analysis tools in analyzing solar, low-energy buildings; and
- Widely disseminate research results and analysis tools to software developers, industry associations, and government agencies.

Scope of the Task

This Task will investigate the availability and accuracy of building energy analysis tools and engineering models to evaluate the performance of solar and low-energy buildings. The scope of the Task is limited to whole building energy analysis tools, including emerging modular type tools, and to widely used solar and low-energy design concepts. Tool evaluation activities will include analytical, comparative, and empirical methods, with emphasis given to blind empirical validation using measured data from test rooms of full-scale buildings. Documentation of engineering models will use existing standard reporting formats and procedures. The impact of improved building energy analysis will be assessed from a building owner perspective.

The audience for the results of the Task is building energy analysis tool developers. However, tool users, such as architects, engineers, energy consultants, product manufacturers, and building owners and managers, are the ultimate beneficiaries of the research, and will be informed through targeted reports and articles.

Means

In order to accomplish the stated goal and objectives, the Participants will carry out research in the framework of two Subtasks:

Subtask A: Tool Evaluation

Subtask B: Model Documentation

Participants

The participants in the Task are: Finland, France, Germany, Spain, Sweden, Switzerland, United Kingdom, and United States. The United States serves as Operating Agent for this Task, with Michael J. Holtz of Architectural Energy Corporation providing Operating Agent services on behalf of the U.S. Department of Energy.

This report documents work carried out under Subtask A.2, Comparative and Analytical Verification Studies.

Table of Contents

Zipped CD Files

Electronic Media Contents

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

README.DOC: Electronic media contents

HVBT294.TM2: TMY2 weather data with constant ODB = 29.4°C

HVBT294.TMY: TMY weather data with constant ODB = 29.4°C

HVBT350.TM2: TMY2 weather data with constant ODB = 35.0°C

HVBT350.TMY: TMY weather data with constant ODB = 35.0°C

HVBT406.TM2: TMY2 weather data with constant ODB = 40.6°C

HVBT406.TMY: TMY weather data with constant ODB = 40.6°C

HVBT461.TM2: TMY2 weather data with constant ODB = 46.1°C

HVBT461.TMY: TMY weather data with constant ODB = 46.1°C

HVBTOUT.XLS: Raw output data spreadsheet used by the IEA participants

PERFMAP.XLS: Performance data (Tables 1-6a through 1-6f)

RESULTS.DOC: Documentation for navigating RESULTS.XLS

RESULTS.XLS: Results spreadsheet to assist users with plotting their results versus analytical solution results and other simulation results

\INPDECKS subdirectory (IEA SHC Task 22 participant simulation input decks)

\DOE2-CIEMAT

\DOE2-NREL

\ENERGYPLUS

\TRNSYS-TUD

Note: Final input decks were not submitted for CASIS, CLIM2000, or PROMETHEUS.

Executive Summary

Objectives

This report describes the Building Energy Simulation Test for Heating, Ventilating, and Air-Conditioning Equipment Models (HVAC BESTEST) project conducted by the Tool Evaluation and Improvement International Energy Agency (IEA) Experts Group. The group was composed of experts from the Solar Heating and Cooling (SHC) Programme, Task 22, Subtask A. The current test cases, E100–E200, represent the beginning of work on mechanical equipment test cases; additional cases that would expand the current test suite have been proposed for future development.

The objective of the tool evaluation subtask has been to develop practical procedures and data for an overall IEA validation methodology that the National Renewable Energy Laboratory (NREL) has been developing since 1981 (Judkoff et al. 1983; Judkoff 1988), with refinements contributed by representatives of the United Kingdom (Lomas 1991; Bloomfield 1989) and the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (American National Standards Institute [ANSI]/ASHRAE Standard 140-2001). The methodology combines empirical validation, analytical verification, and comparative analysis techniques and is discussed in detail in the following Background section. This report documents an analytical verification and comparative diagnostic procedure for testing the ability of whole-building simulation programs to model the performance of unitary space cooling equipment that is typically modeled using manufacturer design data presented in the form of empirically derived performance maps. The report also includes results from analytical solutions as well as from simulation programs that were used in field trials of the test procedure. Other projects conducted by Task 22, Subtask A and reported elsewhere, included work on empirical validation (Guyon and Moinard 1999; Palomo and Guyon 1999; Travesi et al 2001) and analytical verification (Tuomaala 1999; San Isidro 2000). In addition, Task 22, Subtask B has produced a report on the application of the Neutral Model Format in building energy simulation programs (Bring et al 1999).

In this project the BESTEST method, originally developed for use with envelope models in IEA SHC Task 12 (Judkoff and Neymark 1995a), was extended for testing mechanical system simulation models and diagnosing sources of predictive disagreements. Cases E100–E200, described in this report, apply to unitary space cooling equipment. Testing of additional cases, which cover other aspects of HVAC equipment modeling, is planned for the future. Field trials of HVAC BESTEST were conducted with a number of detailed state-of-the-art simulation programs from the United States and Europe including: CA-SIS, CLIM2000, DOE-2, ENERGYPLUS, PROMETHEUS, and TRNSYS. The process was iterative in that executing the simulations led to the refining of HVAC BESTEST, and the results of the tests led to improving and debugging the programs.

HVAC BESTEST consists of a series of steady-state tests using a carefully specified mechanical cooling system applied to a highly simplified near-adiabatic building envelope. Because the mechanical equipment load is driven by sensible and latent internal gains, the sensitivity of the simulation programs to a number of equipment performance parameters is explored. Output values for the cases such as compressor and fan electricity consumption, cooling coil sensible and latent loads, coefficient of performance (COP), zone temperature, and zone humidity ratio are compared and used in conjunction with a formal diagnostic method to determine the algorithms responsible for predictive differences. In these steady-state cases, the following parameters are varied: sensible internal gains, latent internal gains, zone thermostat set point (entering dry-bulb temperature), and outdoor dry-bulb temperature (ODB). To obtain steady-state ODB, ambient dry-bulb temperatures were held constant in the weather data files provided with the test cases. Parametric variations isolate the effects of the parameters singly and in various combinations, as well as the influence of: part-loading of equipment, varying sensible heat ratio, "dry" coil (no latent load) versus "wet" coil (with dehumidification) operation, and operation at typical Air-Conditioning and Refrigeration Institute (ARI) rating conditions. In this way the models are tested in various domains of the performance map.

As a BESTEST user, if you have not already tested your software's ability to model envelope loads, we strongly recommend that you run the envelope-load tests in addition to HVAC BESTEST. A set of envelope loads tests is included in ASHRAE Standard 140 (ANSI/ASHRAE 2001); the Standard 140 test cases are based on IEA BESTEST (Judkoff and Neymark 1995a). Another set of envelope-load test cases, which were designed to test simplified tools such as those currently used for home energy rating systems, is included in HERS BESTEST (Judkoff and Neymark 1995b; Judkoff and Neymark 1997). HERS BESTEST has a more realistic base building than IEA BESTEST; however, its ability to diagnose sources of differences among results is not as detailed (Neymark and Judkoff 1997).

Significance of the Analytical Solution Results

A methodological difference between this work and the envelope BESTEST work of Task 12 is that this work includes analytical solutions. In general, it is difficult to develop worthwhile test cases that can be solved analytically, but such solutions are extremely useful when possible. The analytical solutions represent a "mathematical truth standard" for cases E100–E200. Given the underlying physical assumptions in the case definitions, there is a mathematically provable and deterministic solution for each case. In this context, the underlying physical assumptions about the mechanical equipment (as defined in cases E100–E200) are representative of typical manufacturer data. These data, with which many whole-building simulation programs are designed to work, are normally used by building design practitioners. It is important to understand the difference between a mathematical truth standard and an "absolute truth standard." In the former, we accept the given underlying physical assumptions while recognizing that these assumptions represent a simplification of physical reality. The ultimate or absolute validation standard would be a comparison of simulation results with a *perfectly performed* empirical experiment, the inputs for which are *perfectly specified* to the simulationists. In reality an experiment is performed and the experimental object is specified within some acceptable band of uncertainty. Such experiments are possible but fairly expensive. In the section on future work, we recommend developing a set of empirical validation experiments.

Two of the participating organizations independently developed analytical solutions that were submitted to a third party for review. Comparison of the results indicated some disagreements, which were then resolved by allowing the solvers to review the comments from the third party reviewers, and to also review and critique each other's solution techniques. As a result of this process, both solvers made logical and non-arbitrary changes to their solutions such that their final results are mostly well within a <1% range of disagreement. Remaining minor differences in the analytical solutions are due in part to the difficulty of completely describing boundary conditions. In this case the boundary conditions are a compromise between full reality and some simplification of the real physical system that is analytically solvable. Therefore, the analytical solutions have some element of interpretation of the exact nature of the boundary conditions, which causes minor disagreement in the results. For example, in the modeling of the controller, one group derived an analytical solution for an "ideal" controller, while another group developed a numerical solution for a "realistic" controller. Each solution yields slightly different results, but both are correct in the context of this exercise. Although this may be less than perfect from a mathematician's viewpoint, it is quite acceptable from an engineering perspective.

The remaining minor disagreements among analytical solutions are small enough to allow identification of bugs in the software that would not otherwise be apparent from comparing software only to other software. Therefore, having cases that are analytically solvable when possible improves the diagnostic capabilities of the test procedure. As test cases become more complex, it is rarely possible to solve them analytically.

Field Trial Results

Disagreement among Simulation Programs

After correcting software errors using HVAC BESTEST diagnostics, the mean results of COP and total energy consumption for the simulation programs are on average within <1% of the analytical solution results, with average variations of up to 2% for the low part load ratio (PLR) dry-coil cases (E130 and E140). Ranges of disagreement are further summarized in Table ES-1for predictions of various outputs, disaggregated for dry-coil performance (no dehumidification) and for wet-coil performance (dehumidification moisture condensing on the coil). This range of disagreement for each case is based on the difference between each simulation result versus the mean of the analytical solution results, divided by the mean of the analytical solution results. This summary excludes results for one of the participants who suspected an error(s) in their software, but were not able to correct their results or complete the project.

Table ES-1. Ranges of Disagreement among Simulation Results

Cases	Dry Coil (E100-E140)	Wet Coil (E150-E200)
COP and Total Electric Consumption	0% - 6%[a]	0% - 3%[a]
Zone Humidity Ratio	0% - 11%[a]	0% - 7%[a]
Zone Temperature	0.0°C - 0.7°C (0.1°C)[b]	0.0°C - 0.5°C (0.0°C – 0.1°C)[b]

[a] Percentage disagreement for each case is based on the difference between each simulation result (excluding one simulation participant that could not finish the project) versus the mean of the analytical solution results, divided by the mean of the analytical solution results.
[b] Excludes results for TRNSYS-TUD with realistic controller.

In Table ES-1, the higher level of disagreement in the dry-coil cases occurs for the case with lowest PLR; further discussion of specific disagreements is included in Part III (e.g., Sections 3.4 and 3.5). The disagreement in zone temperature results is primarily from one simulation that applies a realistic controller on a short time step (36 seconds); all other simulation results apply ideal control.

Based on results after "HVAC BESTESTing," the programs appear reliable for performance-map modeling of space cooling equipment when the equipment is operating close to design conditions. In the future, HVAC BESTEST cases will explore modeling at "off-design" conditions and the effects of using more realistic control schemes.

Bugs Found in Simulation Programs

The results generated with the analytical solution techniques and the simulation programs are intended to be useful for evaluating other detailed or simplified building energy prediction tools. The group's collective experience has shown that when a program exhibits major disagreement with the analytical solution results given in Part II, the underlying cause is usually a bug, a faulty algorithm, or a documentation problem. During the field trials, the HVAC BESTEST diagnostic methodology was successful at exposing such problems in every one of the simulation programs tested. The most notable examples are listed below; a full listing appears in the conclusions section of Part III.

- Isolation and correction of a coding error related to calculation of minimum supply air temperature in the DOE-2.1E mechanical system model "RESYS2"; this affected base case efficiency estimates by 36%. (Until recently, DOE-2 was the main building energy analysis program sponsored by the U.S. Department of Energy [DOE]; many of its algorithms are being incorporated into DOE's next-generation simulation software, ENERGYPLUS.)
- Isolation and correction of a problem associated with using single precision variables, rather than double precision variables, in an algorithm associated with modeling a realistic controller in TRNSYS-TUD; this affected compressor power estimates by 14% at medium PLR, and by 45% at low PLR. (TRNSYS is the main program for active solar systems analysis in the U.S.; TRNSYS-TUD is a version with custom algorithms developed by Technische Universität Dresden, Germany.)
- Isolation of a missing algorithm to account for degradation of COP with decreasing PLR in CLIM2000 and in PROMETHEUS, and later inclusion of this algorithm in CLIM2000; this affected compressor power estimates by up to 20% at low PLR. (CLIM2000, which is primarily dedicated to research and development studies, is the most detailed of the building energy analysis programs sponsored by the French national electric utility Electricité de France; PROMETHEUS is a detailed hourly simulation program sponsored by the architectural engineering firm Klimasystemtechnik, Germany.)
- Isolation and correction of problems in CA-SIS associated with neglecting to include the fan heat with the coil load. Neglecting the fan heat on coil load caused up to 4% underestimation of total energy consumption. (CA-SIS, which is based on TRNSYS, was developed by Electricité de France for typical energy analysis studies.)
- Isolation and correction of a coding error in ENERGYPLUS that excluded correction for run time during cycling operation from reported coil loads. This caused reported coil loads to be overestimated by a factor of up to 25 for cases with lowest PLR; there was negligible effect on energy consumption and COP from this error. (ENERGYPLUS has recently been released by DOE as the building energy simulation program that will be supported by DOE.)
- Isolation of problems in CA-SIS, DOE-2.1E and ENERGYPLUS (which were corrected in CA-SIS and ENERGYPLUS) associated with neglecting to account for the effect of degradation of COP (increased run time) with decreasing PLR on the indoor and outdoor fan energy consumptions. Neglecting the PLR effect on fan run time caused a 2% underestimation of total energy consumption at mid-range PLRs.

Additionally, Electricité de France software developers used this project to check on model improvements to CLIM2000 begun before this project began, and completed during the beginning of the project. They demonstrated up to a 50% improvement in COP predictions over results of their previous version.

Conclusions

An advantage of BESTEST is that a program is examined over a broad range of parametric interactions based on a variety of output types, minimizing the possibility for concealment of problems by compensating errors. Performance of the tests resulted in quality improvements to all of the building energy simulation programs used in this study. Many of the errors found during the project stemmed from incorrect code implementation. Some of these bugs may well have been present for many years. The fact that they have just now been uncovered shows the power of BESTEST and also suggests the importance of continuing to develop formalized validation methods.

Checking a building energy simulation program with HVAC BESTEST requires about one person-week for an experienced user. Because the simulation programs have taken many years to produce, HVAC BESTEST provides a very cost-effective way of testing them. As we continue to develop new test cases, we will adhere to the principle of parsimony so that the entire suite of BESTEST cases may be implemented by users within a reasonable time span.

Architects, engineers, program developers, and researchers can use the HVAC BESTEST method in a number of different ways, such as:

- To compare output from building energy simulation programs to a set of analytical solutions that constitute a reliable set of theoretical results given the underlying physical assumptions in the case definitions
- To compare several building energy simulation programs to determine the degree of disagreement among them
- To diagnose the algorithmic sources of prediction differences among several building energy simulation programs
- To compare predictions from other building energy programs to the analytical solutions and simulation results in this report
- To check a program against a previous version of itself after internal code modifications to ensure that only the intended changes actually resulted
- To check a program against itself after a single algorithmic change to understand the sensitivity among algorithms.

In general, the current generation of programs appears reliable for performance-map modeling of space cooling equipment when the equipment is operating close to design conditions. However, the current cases check extrapolation only to a limited extent. Additional cases have been defined for future work to further explore the issue of modeling equipment performance at off-design conditions, which are not typically included within the performance data provided in manufacturer catalogs. As buildings become more energy efficient, either through conservation or via the integration of solar technology, the relative importance of equipment operation at off-design conditions increases. The tendency among some practitioners to oversize equipment, as well as the importance of simulation for designing equipment retrofits and upgrades, also emphasizes the importance of accurate off-design equipment modeling. For the state of the art in annual simulation of mechanical equipment to improve, manufacturers need to either readily provide expanded data sets on the performance of their equipment or improve existing equipment selection software to facilitate the generation of such data sets.

Future Work: Recommended Additional Cases

The previous IEA BESTEST envelope test cases (Judkoff and Neymark 1995a) have been code-language adapted and formally approved by ANSI and ASHRAE as a Standard Method of Test, ANSI/ASHRAE Standard 140 (ANSI/ASHRAE Standard 140-2001). The BESTEST procedures are also being used as teaching tools for simulation courses at universities in the United States and Europe.

The addition of mechanical equipment tests to the existing envelope tests gives building energy software developers and users an expanded ability to test a program for reasonableness of results and to determine if a program is appropriate for a particular application. Cases E100–E200 emphasize the testing of a program's modeling capabilities with respect to the building's mechanical equipment on the working-fluid side of the coil. These cases represent just the beginning of possible work on validation of mechanical equipment models. In the course of this work it became apparent, based on comments from the participants and observations of the test specification authors, that some additional test cases would be useful for improved error trapping and diagnostics with HVAC BESTEST. In future work, the following extensions to the test suite should be considered. (A more complete listing is included in Part III, Section 3.5.1.)

The following cases have been defined for an extension of Task 22 (Neymark and Judkoff 2001). These cases will not be possible to solve analytically, but will be developed as comparative test cases. They are dynamic test cases that utilize unrevised dynamic annual site weather data. They also help to scale the significance of disagreements that are less obvious with steady-state test cases. The cases test the ability to model:

- Quasi-steady-state performance using dynamic boundary conditions (dynamic internal gains loading and dynamic weather data)
- Latent loading from infiltration
- Outside air mixing
- Periods of operation away from typical design conditions
- Thermostat setup (dynamic operating schedule)
- Undersized system performance
- Economizer with a variety of control schemes
- Variation of PLR (using dynamic weather data)
- Outdoor dry-bulb temperature and entering dry-bulb temperature performance sensitivities (using dynamic loading and weather data)

Additional cases and improvements that are also being considered for development as part of the Task 22 extension are:

- Various control schemes including minimum on/off, hysteresis, and proportional controls
- Variation of part load performance based on more detailed data
- Variable air volume fan performance and control
- Heating equipment such as furnaces and heat pumps
- Radiant heating and cooling systems
- Multizone envelope heat transfer
- Update of simulation reference results for IEA (envelope) BESTEST and HERS BESTEST cases with floor slab and below-grade wall ground-coupled heat transfer.

Longer-term work would include developing test cases for:

- Thermal storage equipment
- Air-to-air heat exchanger
- More complex systems associated with larger buildings, such as:
    - Large chillers
    - Chilled water loops

- Cooling towers and related circulation loops
  - More complex air handling systems
  - Other "plant" equipment

- Equivalent inputs for highly detailed unitary system primary loop component models of, for example, compressors, condensers, evaporators, and expansion valves.

More envelope modeling test cases could be included such as:

- Improved ground coupling cases
- Expanded infiltration tests (e.g., testing algorithms that vary infiltration with wind speed)
- Vary radiant fraction of heat sources
- Moisture adsorption/desorption.

ASHRAE is also conducting related work to develop tests related to the airside of the mechanical equipment in commercial buildings (Yuill 2001).

Closing Remarks

The previous IEA BESTEST procedure (Judkoff and Neymark 1995a), developed in conjunction with IEA SHC Task 12, has been code-language-adapted and approved by ANSI and ASHRAE as a Standard Method of Test for evaluating building energy analysis computer programs (ANSI/ASHRAE Standard 140-2001). This method primarily tests envelope-modeling capabilities. We anticipate that after code language adaptation, HVAC BESTEST will be added to that Standard Method of Test. In the United States, the National Association of State Energy Officials (NASEO) Residential Energy Services Network (RESNET) has also adopted HERS BESTEST (Judkoff and Neymark 1995b) as the basis for certifying software to be used for Home Energy Rating Systems under the association's national guidelines. The BESTEST procedures are also being used as teaching tools for simulation courses at universities in the United States and Europe. We hope that as the procedures become better known, developers will automatically run the tests as part of their normal in-house quality control efforts. The large number of requests (more than 800) that we have received for the envelope BESTEST reports indicates that this is beginning to happen. Developers should also include the test input and output files with their respective software packages to be used as part of the standard benchmarking process.

Clearly, there is 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 (see the following Background section), including:

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

Future work should therefore encompass:

- Continued production of a standard set of analytical verification tests
- Development of a sequentially ordered series of high-quality data sets for empirical validation
- Development of a set of diagnostic comparative tests that emphasize the modeling issues important in large commercial buildings, such as zoning and more tests for heating, ventilation, and air-conditioning systems.

Continued support of model development and validation activities is essential because occupied buildings are not amenable to classical controlled, repeatable experiments. The energy, comfort, and lighting performance of buildings depends on the interactions among a large number of energy transfer mechanisms, components, and systems. Simulation is the only practical way to bring a systems integration problem of this magnitude within the grasp of designers. Greatly reducing the energy intensity of buildings through better design is possible with the use of such simulation tools. However, building energy simulation programs will not come into widespread use unless the design and engineering communities have confidence in these programs. Confidence can best be encouraged by a rigorous development and validation effort, combined with friendly user interfaces to minimize human error and effort.

Finally, the authors wish to acknowledge that the expertise available through IEA and the dedication of the participants were essential to the success of this project. For example, when the test cases were developed, they were originally intended as comparative tests, so that there would be simulation results but not analytical solution results. However, after initial development of the steady-state tests, it became apparent to us that analytical solutions would be possible. The participating countries provided the expertise to derive two independent sets of analytical solutions and a third party to examine the results of the two original solvers. Also, over the 3-year field trial effort, there were several revisions to the HVAC BESTEST specifications and subsequent re-executions of the computer simulations. This iterative process led to the refining of HVAC BESTEST, and the results of the tests led to improving and debugging of the programs. The process underscores the leveraging of resources for the IEA countries participating in this project. Such extensive field trials, and resulting enhancements to the tests, were much more cost effective with the participation of the IEA SHC Task 22 experts.

Final Report Structure

This report is divided into four parts. Part I is a user's manual that furnishes instructions on how to apply the HVAC BESTEST procedure. Part II describes what two of the working group participants did to develop analytical solutions independently, including a third-party comparison. After the third-party comparison and comments, there was intense follow-up comparison and discussion among the initial solvers to revise the analytical solutions so a high level of agreement was achieved. The last section of Part II also includes a tabulation of the analytical solution results by each solver along with disagreement statistics. Part II will be useful to those wanting to understand the physical theories and assumptions underlying the test cases. However, with the exception of the last section, which includes the final analytical results tables, it is not necessary to read Part II to implement the test procedure.

Part III describes what the working group members did to field-test HVAC BESTEST and produce a set of results using several state-of-the-art, detailed whole-building energy simulation programs with time steps of 1 hour or less. This includes a summary compilation of significant bugs found in all the simulation programs as a result of their testing in the field trials. Part III is helpful for understanding how other simulationists implemented the test procedure and applied the diagnostic logic. However, it is not necessary to read Part III to implement the test procedure.

Part IV presents both the analytical solution and simulation program results in tables and graphs along with disagreement statistics comparing the simulation programs to each other and to the analytical solutions. These data can be used to compare results from other programs to Part IV results, and to observe the range of disagreement among the simulation programs used for the field trials versus the analytical solutions.

The report includes electronic media that contains the weather data and all the analytical solution and simulation program results in a common spreadsheet format, along with participants' simulation input data.

Additionally, we have included with the front matter the following Background section that includes an overview of validation methodology and a summary of previous NREL, IEA-related, and other validation work related to software that analyzes energy use in buildings.

Background

This section summarizes some of the work that preceded this BESTEST effort and describes the overall methodological and historical context for BESTEST.

The increasing power and attractive pricing of personal computers has engendered a proliferation of building energy analysis software. An on-line directory sponsored by DOE (*Building Energy Tools Directory* 2001) lists more than 200 building energy software tools that have thousands of users worldwide. Such tools utilize a number of different approaches to calculating building energy usage (Gough 1999). There is little if any objective quality control of much of this software. An early evaluation of a number of design tools conducted in IEA's SHC Programme Task 8 showed large unexplained predictive differences between these tools, even when run by experts (Rittelmann and Ahmed 1985). More recent work to develop software testing and evaluation procedures indicates that the causes of predictive differences can be isolated, and that bugs that may be causing anomalous differences can be found and fixed, resulting in program improvements. However, even with improved capabilities for testing and evaluating software, predictive differences remain (Judkoff and Neymark 1995a, 1995b).

Users of building energy simulation tools must have confidence in their utility and accuracy because the use of such tools offers a great potential for energy savings and comfort improvements. Validation and testing is a necessary part of any software development process, and is intended to stimulate the confidence of the user. In recognition of the benefits of testing and validation, an effort was begun under IEA SHC Task 8, and continued in SHC Task 12 Subtask B and Buildings and Community Systems (BCS) Annex 21 Subtask C, to develop a quantitative procedure for evaluating and diagnosing building energy software (Judkoff et al. 1988; Bloomfield 1989). The procedure that resulted from that effort is called the Building Energy Simulation Test (BESTEST) and Diagnostic Method (Judkoff and Neymark 1995a). This initial version of BESTEST focused on evaluating a simulation program's ability to model building envelope heat transfer, and to model basic thermostat controls and mechanical ventilation. As part of SHC Task 22, the BESTEST work was expanded to include more evaluation of heating, ventilating and air-conditioning (HVAC) equipment models. This new procedure, which is the subject of this report, is called HVAC BESTEST.

Before the inception of IEA SHC Task 8, NREL (then the Solar Energy Research Institute) had begun working on a comprehensive validation methodology for building energy simulation programs (Judkoff et al. 1983). This effort was precipitated by two comparative studies that showed considerable disagreement between four simulation programs—DOE-2, BLAST, DEROB, and SUNCAT—when given equivalent input for a simple direct-gain solar building with a high and low heat capacitance parametric option (Judkoff, Wortman, and Christensen 1980; Judkoff, Wortman, and O'Doherty 1981). These studies clearly indicated the need for a validation effort based on a sound methodological approach.

Validation Methodology

A typical building energy simulation program contains hundreds of variables and parameters. The number of possible cases that can be simulated by varying each of these parameters in combination is astronomical and cannot practically be fully tested. For this reason the NREL validation methodology required three different kinds of tests (Judkoff et al. 1983):

- Empirical Validation—in which calculated results from a program, subroutine, or algorithm are compared to monitored data from a real building, test cell, or laboratory experiment.

- Analytical Verification—in which outputs from a program, subroutine, or algorithm are compared to results from a known analytical solution or generally accepted numerical method for isolated heat transfer mechanisms under very simple and highly defined boundary conditions
- Comparative Testing—in which a program is compared to itself, or to other programs that may be considered better validated or more detailed and, presumably, more physically correct.

Table 1-1 shows the advantages and disadvantages of these three techniques. Defining two terms is useful in interpreting Table 1-1. Here a “model” is the representation of reality for a given physical behavior. For example, one way to model heat transfer through a wall is by using a simplifying assumption of one-dimensional conduction. An alternative (more detailed) model for wall heat transfer could employ two-dimensional conduction. The “solution process” is a term that encompasses the mathematics and computer coding to solve a given model (e.g., a finite difference approximation to solve a differential equation) and the technique for integrating individual models and boundary conditions into an overall solution methodology—such as an iterative energy balance through layers of a single wall, over all the surfaces of a given zone, or between a zone(s) and its related mechanical system(s). The solution process for a model can be perfect, while the model remains faulty or inappropriate for a given physical situation or purpose; for example, using a one-dimensional conduction model where two-dimensional conduction dominates.

Table 1-1. Validation Techniques

Technique	Advantages	Disadvantages
Empirical Test of model and solution process	• Approximate truth standard within experimental accuracy • Any level of complexity	• Experimental uncertainties: – Instrument calibration, spatial/ temporal discretization – Imperfect knowledge/specification of the experimental object (building) being simulated • Detailed measurements of high quality are expensive and time consuming • Only a limited number of test conditions are practical
Analytical Test of solution process	• No input uncertainty • Exact mathematical truth standard for the given model • Inexpensive	• No test of model validity • Limited to highly constrained cases for which analytical solutions can be derived
Comparative Relative test of model and solution process	• No input uncertainty • Any level of complexity • Many diagnostic comparisons possible • Inexpensive and quick	• No truth standard

Empirical validation is required when establishing a truth standard for evaluating the ability of a program to analyze real physical behavior. However, empirical validation is only possible within the range of measurement uncertainty, including instrument uncertainty and spatial and temporal discretization uncertainty. Test cells and buildings are large and relatively complex experimental objects. It is difficult to know the exact nature of the construction details, material properties, and actual construction implementation in the field. The simulationist is therefore left with a degree of uncertainty about the inputs that accurately represent the experimental object. Meticulous care is required to describe the experimental apparatus as clearly as possible to modelers to minimize this uncertainty. This includes experimental determination of as many material properties as possible, including overall building properties such as steady-state heat loss coefficient and effective thermal mass, among others.

The NREL methodology subdivided empirical validation into different levels. This was necessary because many of the empirical validation efforts conducted before then had produced results that could not support definitive conclusions despite considerable expenditure of resources. The levels of validation depend on the degree of control exercised over the possible sources of error in a *simulation*. These error sources consist of seven types divided into two groups.

External Error Types

- Differences between the actual microclimate that affects the building and the weather input used by the program
- Differences between the actual schedules, control strategies, and effects of occupant behavior and those assumed by the program user
- User error in deriving building input files
- Differences between the actual thermal and physical properties of the building including its HVAC systems and those input by the user.

Internal Error Types

- Differences between the actual thermal transfer mechanisms taking place in the real building and its HVAC systems and the simplified model of those physical processes in the simulation
- Errors or inaccuracies in the mathematical solution of the models
- Coding errors.

At the most simplistic level, the actual long-term energy use of a building is compared to that calculated by a computer program, with no attempt to eliminate sources of discrepancy. Because this level is similar to how a simulation tool would actually be used in practice, it is favored by many representatives of the building industry. However, it is difficult to interpret the results of this kind of validation exercise because all possible error sources are simultaneously operative. Even if good agreement is obtained between measured and calculated performance, the possibility of offsetting errors prevents a definitive conclusion about the model's accuracy. More informative levels of validation can be achieved by controlling or eliminating various combinations of error types and by increasing the number of output-to-data comparisons; for example, comparing temperature and energy results at various time scales ranging from sub-hourly to annual values. At the most detailed level, all known sources of error are controlled to identify and quantify unknown error sources, and to reveal cause and effect relationships associated with the error sources.

This same general principle applies to comparative and analytical methods of validation. The more realistic the test case, the more difficult it is to establish cause and effect and to diagnose problems. The simpler and more controlled the test case, the easier it is to pinpoint the source(s) of error or inaccuracy. It is useful to methodically build up to realistic cases for testing the interaction between algorithms that model linked mechanisms.

Each comparison between measured and calculated performance represents a small region in an immense N-dimensional parameter space. We are constrained to exploring relatively few regions within this space, yet we would like to be assured that the results are not coincidental and do represent the validity of the simulation elsewhere in the parameter space. The analytical and comparative techniques minimize the uncertainty of the extrapolations we must make around the limited number of empirical domains it is possible to sample. Table 1-2 classifies these extrapolations.

Table 1-2. Types of Extrapolation

Obtainable Data Points	Extrapolation
A few climates	Many climates
Short-term total energy usage	Long-term total energy usage or vice versa
Short-term (hourly) temperatures and/or fluxes	Long-term total energy usage or vice versa
A few equipment performance points	Many equipment performance points
A few buildings representing a few sets of variable mixes	Many buildings representing many sets of variable mixes
Small-scale: simple test cells, buildings, and mechanical systems; and laboratory experiments	Large-scale complex buildings with complex HVAC systems or vice versa

Figure 1-1 shows one process by which we may use the analytical, empirical, and comparative techniques together. In actuality, these three techniques may be used together in a number of ways. For example, intermodel comparisons may be done before an empirical validation exercise to better define the experiment and to help estimate experimental uncertainty by propagating all known sources of uncertainty through one or several whole-building energy simulation programs (Hunn et al. 1982; Martin 1991; Lomas et al. 1994).

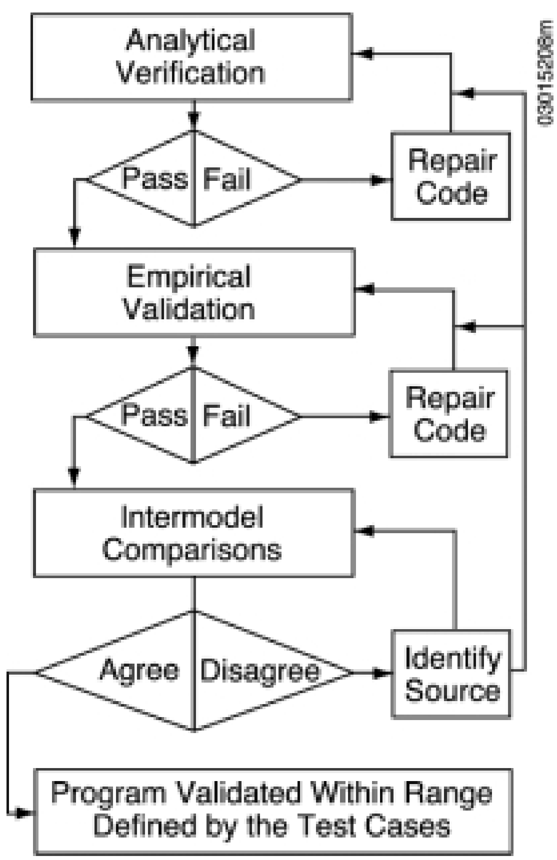

Figure 1-1. Validation method

For the path shown in Figure 1-1, the first step is to run the code against analytical test cases. This checks the mathematical solution of major heat transfer models in the code. If a discrepancy occurs, the source of the difference must be corrected before any further validation is done.

The second step is to run the code against high-quality empirical validation data and to correct errors. However, diagnosing error sources can be quite difficult, and is an area of research in itself as described below. Comparative techniques can be used to create diagnostic procedures (Judkoff, Wortman, and Burch 1983; Judkoff 1985a, 1985b, 1988; Judkoff and Wortman 1984; Morck 1986; Judkoff and Neymark 1995a) and to better define the empirical experiments.

The third step involves checking the agreement of several different programs with different thermal solution and modeling approaches (which have passed through steps 1 and 2) in a variety of representative cases. Cases for which the program predictions diverge indicate areas for further investigation. This utilizes the comparative technique as an extrapolation tool. When programs have successfully completed these three stages, we consider them to be validated for the domains in which acceptable agreement was achieved. That is, the codes are considered validated for the range of building and climate types represented by the test cases.

Once several detailed simulation programs have satisfactorily passed through the procedure, other programs and simplified design tools can be tested against them. A validated code does not necessarily represent truth. It does represent a set of algorithms that have been shown, through a repeatable procedure, to perform according to the current state of the art.

The NREL methodology for validating building energy simulation programs has been generally accepted by the IEA (Irving 1988) and elsewhere with a number of methodological refinements suggested by subsequent researchers (Bowman and Lomas 1985a; Lomas and Bowman 1987; Lomas 1991; Lomas and Eppel 1992; Bloomfield 1985, 1988, 1999; Bloomfield, Lomas, and Martin 1992; Allen et al. 1985; Irving 1988; Bland and Bloomfield 1986; Bland 1992; Izquierdo et al. 1995; Guyon and Palomo 1999a). Additionally, the Commission of European Communities has conducted considerable work under the PASSYS program (Jensen 1993; Jensen and van de Perre 1991; Jensen 1989).

Summary of Previous NREL, IEA-Related, and Other Validation Work

Beginning in 1980, NREL conducted several analytical, empirical, and comparative studies in support of the validation methodology. These studies focused on heat transfer phenomena related to the building envelope. Validation work has been continued and expanded by NREL and others as discussed below.

Analytical Verification

At NREL, a number of analytical tests were derived and implemented including wall conduction, mass charging and decay resulting from a change in temperature, glazing heat transfer, mass charging and decay resulting from solar radiation, and infiltration heat transfer. These tests and several comparative studies facilitated the detection and diagnosis of a convergence problem in the DEROB-3 program, which was then corrected in DEROB-4 (Wortman, O'Doherty, and Judkoff 1981; Burch 1980; Judkoff, Wortman, and Christensen 1980; Judkoff, Wortman, and O'Doherty 1981). These studies also showed DOE2.1, BLAST-3, SUNCAT-2.4, and DEROB-4 to be in good agreement with the analytical solutions even though considerable disagreement was observed among them in some of the comparative studies. This confirmed the need for both analytical and comparative tests as part of the overall validation methodology.

Further development of the analytical testing approach has occurred in Europe, and has been collected in an IEA working document of analytical tests (Tuomaala 1999). This collection includes work on conduction tests (Bland and Bloomfield 1986; Bland 1993); infrared radiation tests (Pinney and Bean 1988; Stefanizzi, Wilson, and Pinney 1988); multizone air flow tests (Walton 1989); solar shading tests (Rodriguez and

Alvarez 1991); building level conduction, solar gains, and solar/mass interaction tests (Wortman, O'Doherty, and Judkoff 1981); and conduction, long-wave radiation exchange, solar shading, and whole-building zone temperature calculation tests (Comité Européen de la Normalisation [CEN] 1997). The IEA analytical test collection also includes field trial modeler reports by some of the Task 22 participants that critique the utility of the tests. Further field trial details are included in other papers (Guyon and Palomo 1999a; San Isidro 2000; Tuomaala et al. 1999). Other work includes a study of convection coefficients that compares whole-building simulation results to pure analytical and computational fluid dynamics solutions as convective coefficients are varied, and includes comparisons with convective coefficient empirical data (Beausoleil-Morrison 2000).

ASHRAE has sponsored work under ASHRAE 1052-RP on the analytical testing approach. A set of building level tests has been completed. These tests cover convection, steady-state and dynamic conduction (including ground coupling), solar radiation, glazing transmittance, shading, interior solar distribution, infiltration, interior and exterior infrared radiative exchange, and internal heat gains. (Spitler, Rees, and Dongyi 2001) That work incorporates and expands on the previous IEA work cited above, and also includes new test cases. Testing related to airside mechanical equipment in commercial buildings is also nearing completion (Yuill 2001, ASHRAE 865-RP).

Empirical Validation

Several major empirical validation studies have been conducted including:

- NREL (formerly SERI) Direct Gain Test House near Denver, Colorado
- National Research Council of Canada (NRCan) Test House in Ottawa, Canada
- Los Alamos National Laboratory Sunspace Test Cell in Los Alamos, New Mexico
- Building Research Establishment Test Rooms in Cranfield, England
- Electricité de France 'ETNA' and 'GENEC' Test Cells in France
- Iowa Energy Resource Station (ERS) near Des Moines, Iowa.

Data were collected from the NREL Test House during the winters of 1982 and 1983, and two studies were conducted using the DOE-2.1A, BLAST-3.0, and SERIRES computer programs (Burch et al. 1985). In the first study, based on the 1982 data, nine cases were run, beginning with a base case (case 1) in which only "handbook" input values were used, and ending with a final case (case 9) in which measured input values were used for infiltration, ground temperature, ground albedo, set point, and opaque envelope and window conductances (Judkoff, Wortman, and Burch 1983). Simulation heating energy predictions were high by 59%–66% for the handbook case. Simulation heating energy predictions were low by 10%–17% when input inaccuracies were eliminated using measured values. However, root mean square (rms) temperature prediction errors were actually greater for case 9, which indicated the existence of compensating errors in some of the programs.

In the second study, based on the 1983 data, a comparative diagnostic approach was used to determine the sources of disagreement among the computer programs (25%) and between the programs and the measured data (±13%) (Judkoff and Wortman 1984). The diagnostics showed that most of the disagreement was caused by the solar and ground-coupling algorithms. Also, the change in the range of disagreement caused by the difference between the 1982 and 1983 weather periods confirmed the existence of compensating errors.

The Canadian direct gain study and the Los Alamos Sunspace study were both done in the context of IEA SHC Task 8 (Judkoff 1985a, 1985b, 1986; Barakat 1983; Morck 1986; McFarland 1982). In these studies a combination of empirical, comparative, and analytical techniques was used to diagnose the sources of difference among simulation predictions, and between simulation predictions and measurements. These studies showed that disagreement increases in cases where the solar forcing function is greater, and decreases in cases where one-dimensional conduction is the dominant heat-transfer mechanism.

The BRE study was done in the context of IEA Energy Conservation in Buildings and Community Systems (ECBCS) Program Annex 21. Twenty-five sets of results from 17 different simulation programs were compared (Lomas et al. 1994). Most of the simulation programs underpredicted the energy consumption with considerable variation among the simulation programs. The modeling of internal convection and the influence of temperature stratification were indicated as two of the primary causes for the discrepancies. These data were used in subsequent research to check the appropriateness of various internal convection models for various zone air conditions (Beausoleil-Morrison and Strachan 1999).

The French data from the ETNA and GENEC test cells were used for IEA SHC Task 22 (Guyon and Moinard 1999). In all, ten different simulation programs were compared to measured results over three separate experiments. In the first two experiments using the ETNA cells, an ideal purely convective heat source was compared to a typical zonal electric convective heater. In the first experiment the simulations predicted zone temperature based on given heater power; in the second experiment zone thermostat set points were given, and the simulations predicted heater energy consumption. Both experiments incorporated pseudo-random variation of heater power and thermostat set points, respectively, and were used to test a new technique for diagnosing modeling errors in building thermal analysis (Guyon and Palomo 1999b). In the second experiment the simulated energy consumption predictions were about 10%–30% lower than the measurements in both test cells, which was consistent with higher simulated than measured zone temperatures in the first experiment. The simulations (which generally assume an ideal purely convective heat source) gave better agreement with the empirical results of the typical convective heater than with the ideal heater. Possible reasons for this unexpected outcome include higher than specified building loss coefficients, and higher interior film coefficients caused by high mixing from the ideal heater.

In the third experiment with the GENEC test cells the objective was to validate the calculation of solar gains through glazed surfaces by estimating resulting free float temperatures. In this experiment simulation results showed less agreement with measured data than for the ETNA experiments, but the simulation results were roughly equivalent with each other.

In the ERS tests the goal of the project was to assess the accuracy of predicting the performance of a realistic commercial building with realistic operating conditions and HVAC equipment. Four simulation programs were compared to empirical results for constant air volume and variable air volume HVAC systems. Conclusions indicate that after improvements to the models and test specifications, simulation results had generally good agreement with measured data within the uncertainty of the experiments (Travesi et al 2001).

In general, these studies demonstrated the importance of designing validation studies with a very high degree of control over the previously mentioned external error sources. For this reason, the NREL methodology emphasized the following points for empirical validation:

- Start with very simple test objects, before progressing to more complex buildings.
- Use a detailed mechanism level approach to monitoring that would ideally allow a measured component-by-component energy balance to be determined.
- Experimentally measure important overall building macro-parameters, such as the overall steady-state heat loss coefficient (UA) and the effective thermal mass, to crosscheck the building specifications with the as-built thermal properties.
- Use a diversity of climates, building types, and modes of operation to sample a variety of domains within the parameter space.

- Compare measured data to calculated outputs at a variety of time scales, and on the basis of both intermediate and final outputs including temperature and power values.

The studies also showed the diagnostic power of using comparative techniques in conjunction with empirical validation methods. These are especially useful for identifying compensating errors in a program.

European work on empirical validation included a comprehensive review of empirical validation data sets (Lomas 1986; Lomas and Bowman 1986); a critical review of previous validation studies (Bowman and Lomas 1985b); the construction and monitoring of a group of test cells and several validation studies using the test cell data (Martin 1991); and methodological work on data analysis techniques (Bloomfield et al 1995; Eppel and Lomas 1995; Guyon and Palomo 1999b; Izquierdo et al. 1995; Lomas and Eppel 1992; Rahni et al. 1999).

For convective surface coefficient component models, a number of studies that include data and correlations have been conducted. These have been useful for empirical testing of previous assumptions, along with developing improvements to algorithms where needed (Beausoleil-Morrison 2000; Fisher and Pedersen 1997; Spitler, Pedersen, and Fisher 1991; Yazdanian and Klems 1994).

Component models for unitary air-conditioning equipment have been compared to laboratory data in an ASHRAE research project. (LeRoy, Groll, and Braun 1997; LeRoy, Groll, and Braun 1998)

Additional summaries of numerous whole-building simulation and individual model empirical validation studies can be found in the proceedings of the International Building Performance Simulation Association (IBPSA) and elsewhere including building load related studies (e.g., Ahmad and Szokolay 1993; Boulkroune et al. 1993; David 1991; Guyon and Rahni 1997; Guyon, Moinard, and Ramdani 1999) and mechanical (including solar) equipment related studies (e.g., Nishitani et al.1999; Trombe, Serres, and Mavroulakis 1993; Walker, Siegel, and Degenetais 2001; Zheng et al.1999). A summary of empirical validation studies applied to one simulation program has also been published (Sullivan 1998).

Intermodel Comparative Testing: The BESTEST Approach

Two major comparative testing and diagnostics procedures were developed before the work described in this report was conducted. IEA BESTEST, the first of these procedures, was developed in conjunction with IEA SHC Task 12b/ECBCS Annex 21c. It is designed to test a program's ability to model the building envelope, along with some mechanical equipment features, and provides a formal diagnostic method to determine sources of disagreement among programs (Judkoff and Neymark 1995a). The 5-year research effort to develop IEA BESTEST involved field trials by 9 countries using 10 simulation programs. Important conclusions of the IEA BESTEST effort include:

- The BESTEST method trapped bugs and faulty algorithms in every program tested.
- The IEA Task 12b/21c experts unanimously recommended that no energy simulation program be used until it is "BESTESTed."
- BESTEST is an economic means of testing, in several days, software that has taken many years to develop.
- Even the most advanced whole-building energy models show a significant range of disagreement in the calculation of basic building physics.
- Improved modeling of building physics is as important as improved user interfaces.

Home Energy Rating Systems (HERS) BESTEST is the second of these procedures. It was designed to test simplified tools such as those currently used for home energy rating systems (Judkoff and Neymark 1995b). It also tests the ability of analysis tools to model the building envelope in hot/dry and cold/dry climates. A similar version of HERS BESTEST was also developed for a hot/humid climate. (Judkoff and Neymark 1997). Although HERS BESTEST has a more realistic base building than IEA BESTEST, its ability to diagnose sources of differences among results is not as detailed. Additional discussion comparing IEA BESTEST and HERS BESTEST can be found in a previous paper (Neymark and Judkoff 1997).

In addition to the original IEA BESTEST field trial modeler reports, papers from other software authors who documented their experiences indicate specific problems in software uncovered by the IEA and HERS BESTEST procedures (Fairey et al 1998; Haddad and Beausoleil-Morrison 2001; Haltrecht and Fraser 1997; Judkoff and Neymark 1998; Mathew and Mahdavi 1998; Soubdhan et al. 1999). IEA BESTEST has recently been adapted for use in The Netherlands; this adaptation includes rerunning the results for the region's weather, redesigning some of the test cases, and translating the test specification into Dutch (Institut voor Studie en Stimulering van Onderzoekop Het Gebied van Gebouwinstallaties [ISSO] 2000; Plokker 2001). The list of IEA BESTEST and HERS BESTEST users continues to grow, and NREL has received requests for and sent out about 800 copies of the test procedures worldwide.

The Japanese government has developed its own tests for the evaluation of building energy analysis computer programs (NRCan 2000; Sakamoto 2000). The Japanese test suite (as translated into English by NRCan) is somewhat comparable to HERS BESTEST. However, the Japanese tests have fewer test cases (parametric variations), and less detail included in the test specification (e.g., less detail on optical properties of windows), which precludes the possibility of generating reference results using highly detailed simulations.

Software Tests Applied to Codes and Standards

ASHRAE Standard 140-2001 (Standard Method of Test for the Evaluation of Building Energy Analysis Computer Programs) is based on IEA BESTEST described above (ANSI/ASHRAE 2001; Judkoff and Neymark 1999).

The HERS BESTEST test cases represent the Tier 1 and Tier 2 Tests for Certification of Rating Tools as described in the *U.S. Code of Federal Regulations* (DOE 10 CFR Part 437) and the HERS Council Guidelines for Uniformity (HERS Council 1995). The NASEO Board of Directors, in a joint effort with RESNET, has issued procedures that require home energy rating software programs used by a given HERS to have passed HERS BESTEST using example acceptability ranges set forth in HERS BESTEST Appendix H (NASEO/RESNET 2000). The U.S. Environmental Protection Agency's (EPA) Energy Star Homes program requires HERS BESTESTed software to be used for developing EPA ratings (Bales and Tracey 1999).

Two European standards that include procedures for validating software, PrEN 13791 and PrEN 13792 have been approved (CEN 1999; CEN 2000). These standards define detailed and simplified calculation techniques and validation procedures for building energy simulation software based on calculation of internal temperatures in a single room. The references for these tests include many of the analytical verification procedures also collected by IEA SHC Task 22, and some of the tests used in PrEN 13791 were run as part of IEA SHC Task22a (Tuomaala 1999). Although the CEN cases are useful, one comment is that the CEN approach assumes its physical model is correct and therefore is too restrictive in terms of acceptance of detailed modeling approaches. Two more CEN standards are under development, including Working Items 89040.1 and 89040.2 on cooling load calculations and cooling energy calculations, respectively (IEA SHC Task 22 2001).

IEA SHC Task 22 and HVAC BESTEST

The objective of IEA SHC Task 22 has been to further develop practical implementation procedures and data for the overall validation methodology. The task has therefore proceeded on three tracks, with the analytical verification approach led by Finland, the empirical validation approach led by France and Spain, and the intermodel comparative testing approach led by the United States. The United States has also served as the chair for the IEA SHC Task 22A, the Tool Evaluation Experts Group.

The procedures presented in this report take the "analytical verification" approach. Later tests will have more realistic dynamic boundary conditions for which analytical solutions will not be possible. They will therefore require the comparative approach. Here, a set of carefully specified cases is described so that equivalent input files can be easily defined for a variety of detailed and simplified whole-building energy simulation programs. The given analytical solutions represent a mathematical truth standard. That is, given the underlying physical assumptions in the case definitions, then there is a mathematically provable and deterministic solution for each case. It is important to understand the difference between a "mathematical truth standard" and an "absolute truth standard". In the former we accept the given underlying physical assumptions while recognizing that these assumptions represent a simplification of physical reality. The ultimate or "absolute" validation standard would be comparison of simulation results with a *perfectly performed* empirical experiment, the inputs for which are *perfectly specified* to the simulationists. In reality for empirical studies, an experiment is performed and the experimental object is specified within some acceptable band of uncertainty. This set of analytical solutions is based on the assumption of a performance map approach to the modeling of mechanical systems. For unitary systems, manufacturers do not typically provide the detailed level of information required for "first-principles" modeling. They generally supply performance maps derived from a limited set of empirical data points, developed primarily for HVAC system designers to use when selecting equipment. Because these are the types of data that are easiest to acquire and use, most current detailed whole-building simulation programs commonly used by design practitioners take the performance map approach to HVAC modeling. In the future, it may be possible to develop a more detailed set of equivalent inputs that could be used for testing first-principles models side-by-side with performance-map models.

Although the analytical solution results do not represent absolute truth, they do represent a mathematically correct solution of the performance map modeling approach for each test case. A high degree of confidence in these solutions is merited because of the process by which the mathematical implementation of the solutions has been derived. This involved three steps. First, two separate groups worked independently to derive their solutions. Second, a third party review was conducted of both sets of solutions. Third, both groups worked together to resolve any remaining differences. At the end of this process the analytical solutions derived by each group agreed generally well within <1% difference. Therefore, the only source of legitimate disagreement between simulation program results and analytical solution results is the model within the simulation program. A program that disagrees with the analytical solution results in this report may not be incorrect, but it does merit scrutiny.

The field trial modeler reports of Part III and experience from other analytical verification tests have shown that the underlying cause of such discrepancies is usually a bug or a faulty algorithm (Judkoff et al. 1988; Bloomfield 1989; Spitler, Rees, and Dongyi 2001). Although they are not a perfect solution to the validation problem, we hope that these cases and the accompanying set of results will be useful to software developers and to designers attempting to determine the appropriateness of a program for a particular application.

The test cases presented here expand the BESTEST work conducted in IEA SHC Task 12 by adding analytical verification tests for mechanical equipment based on performance map models. We hope that as this test procedure becomes better known, all software developers will use it, along with the previously developed BESTEST procedures, as part of their standard quality control function. We also hope that they will include the input and output files for the tests as sample problems with their software packages.

The next section, Part I, is a User's Manual that fully describes the test cases, as well as how to use the test cases and the diagnostic procedures.

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PART I:

User's Manual

1.0 Part I: Heating, Ventilating, and Air-Conditioning (HVAC) BESTEST User's Manual—
Procedure and Specification

1.1 General Description of Test Cases

An analytical verification and comparative diagnostic procedure was developed to test the ability of whole-building simulation programs to model the performance of unitary space-cooling equipment. Typically, this modeling is conducted using manufacturer design data presented as empirically derived performance maps. This section contains a uniform set of unambiguous test cases for comparing simulation results to analytical solutions, and for diagnosing possible sources of disagreement. Because no two programs require exactly the same input information, we have tried to describe the test cases in a fashion that allows many different building simulation programs (representing different degrees of modeling complexity) to be tested.

As summarized in Table 1-1a (metric units) and Table 1-1b (English units), there are 14 cases in all. Terms used in Tables 1-1a and 1-1b are defined in Appendix H. Cases E100 through E200 represent a set of fundamental mechanical equipment tests. These cases test a program's ability to model unitary space-cooling equipment performance under controlled load and weather conditions. Given the underlying physical assumptions in the case definitions, there is a mathematically provable and deterministic solution for each case; Part II of this report describes these analytical solutions.

The configuration of the base-case building (Case E100) is a near-adiabatic rectangular single zone with only user-specified internal gains to drive cooling loads. We purposely kept the geometric and materials specifications as simple as possible to minimize the opportunity for input errors on the user's part. Mechanical equipment specifications represent a simple unitary vapor-compression cooling system, or more precisely a split-system, air-cooled condensing unit with an indoor evaporator coil. As Tables 1-1a and 1-1b show, only the following parameters are varied to develop the remaining cases:

- Internal sensible gains
- Internal latent gains
- Thermostat set point (indoor dry bulb temperature)
- Outdoor dry bulb temperature.

The electronic media included with this document contains the following:

- Typical meteorological year (TMY) format weather: HVBT294.TMY; HVBT350.TMY; HVBT406.TMY; HVBT461.TMY
- Typical meteorological year 2 (TMY2) format weather: HVBT294.TM2; HVBT350.TM2; HVBT406.TM2; HVBT461.TM2
- PERFMAP.XLS (performance data, described later in Section 1.3.2.2.3)
- RESULTS.XLS (analytical solution results and International Energy Agency [IEA] participant simulation results)
- RESULTS.DOC (text file for help with navigating RESULTS.XLS)
- HVBTOUT.XLS (spreadsheet used by IEA participants for recording output
- IEA participants' simulation input files.

Table 1-1a. HVAC BESTEST Case Descriptions (Metric Units).

	Zone			Weather	
	Internal Gains*		Setpoint		
	Sensible	Latent	EDB	ODB	
Case #	(W)	(W)	(°C)	(°C)	Comments
dry zone series					
E100	5400	0	22.2	46.1	Base case, dry coil. High PLR.
E110	5400	0	22.2	29.4	High PLR. Tests low ODB versus E100.
E120	5400	0	26.7	29.4	High PLR. Tests high EDB versus E110. Tests ODB & EDB interaction versus E100.
E130	270	0	22.2	46.1	Low PLR test versus E100.
E140	270	0	22.2	29.4	Tests ODB at low PLR vs E130. Tests PLR at low ODB vs E110.
humid zone series					
E150	5400	1100	22.2	29.4	High PLR. High SHR. Tests latent load versus E110.
E160	5400	1100	26.7	29.4	High PLR. High SHR. Tests EDB versus E150.
E165	5400	1100	23.3	40.6	High PLR. High SHR. Tests ODB & EDB interaction with latent load versus E160.
E170	2100	1100	22.2	29.4	Mid PLR. Mid SHR. Tests low sensible load versus E150.
E180	2100	4400	22.2	29.4	High PLR. Low SHR. Tests SHR versus E150. Tests high latent load versus E170.
E185	2100	4400	22.2	46.1	High PLR. Low SHR. Tests ODB versus E180.
E190	270	550	22.2	29.4	Low PLR. Low SHR Tests low PLR at constant SHR vs E180. Tests latent load at low PLR versus E140.
E195	270	550	22.2	46.1	Low PLR. Low SHR Tests ODB at low PLR & SHR versus E190. Tests low PLR at constant SHR vs E185. Tests latent load at low PLR versus E130.
full load test at ARI conditions					
E200	6120	1817	26.7	35.0	Tests for ARI indoor wetbulb temperature at full sensible and latent loads.

Abbreviations: PLR = Part Load Ratio; ODB = outdoor drybulb temperature;
EDB = entering drybulb temperature; vs = versus;
SHR = Sensible Heat Ratio; ARI = Air Conditioning and Refrigeration Institute.

*Internal Gains are internally generated sources of heat and humidity that are not related to operation of the mechanical cooling system or its air distribution fan.

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Table 1-1b. HVAC BESTEST Case Descriptions (English Units).

	Zone			Weather	
	Internal Gains*		Setpoint		
	Sensible	Latent	EDB	ODB	
Case #	(Btu/h)	(Btu/h)	(°F)	(°F)	Comments
dry zone series					
E100	18430	0	72.0	115.0	Base case, dry coil. High PLR.
E110	18430	0	72.0	85.0	High PLR. Tests low ODB versus E100.
E120	18430	0	80.0	85.0	High PLR. Tests high EDB versus E110. Tests ODB & EDB interaction versus E100.
E130	922	0	72.0	115.0	Low PLR test versus E100.
E140	922	0	72.0	85.0	Tests ODB at low PLR vs E130. Tests PLR at low ODB vs E110.
humid zone series					
E150	18430	3754	72.0	85.0	High PLR. High SHR. Tests latent load versus E110.
E160	18430	3754	80.0	85.0	High PLR. High SHR. Tests EDB versus E150.
E165	18430	3754	74.0	105.0	High PLR. High SHR. Tests ODB & EDB interaction with latent load versus E160.
E170	7166	3754	72.0	85.0	Mid PLR. Mid SHR. Tests low sensible load versus E150.
E180	7166	15018	72.0	85.0	High PLR. Low SHR. Tests SHR versus E150. Tests high latent load versus E170.
E185	7166	15018	72.0	115.0	High PLR. Low SHR. Tests ODB versus E180.
E190	922	1877	72.0	85.0	Low PLR. Low SHR Tests low PLR at constant SHR vs E180. Tests latent load at low PLR versus E140.
E195	922	1877	72.0	115.0	Low PLR. Low SHR Tests ODB at low PLR & SHR versus E190. Tests low PLR at constant SHR vs E185. Tests latent load at low PLR versus E130.
full load test at ARI conditions					
E200	20890	6200	80.0	95.0	Tests for ARI indoor wetbulb temperature at full sensible and latent loads.

Abbreviations: PLR = Part Load Ratio; ODB = outdoor drybulb temperature;
EDB = entering drybulb temperature; vs = versus;
SHR = Sensible Heat Ratio; ARI = Air Conditioning and Refrigeration Institute.

*Internal Gains are internally generated sources of heat and humidity that are not related to operation of the mechanical cooling system or its air distribution fan.

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1.2 Performing the Tests

1.2.1 Input Requirements

Building input data are organized case by case. Section 1.3.2 contains the base building and mechanical system description (Case E100), with additional cases presented in Sections 1.3.3 and 1.3.4. The additional cases are organized as modifications to the base case and ordered in a manner designed to facilitate test implementation. In many instances (e.g., Case E110), a case developed from modifications to Case E100 will also serve as the base case for other cases.

Tables 1-1a and 1-1b (metric and English units, respectively) summarize the various parametric cases contained in HVAC BESTEST. These tables are furnished only as an overview; use Section 1.3 to generate specific input decks. We recommend a quick look back at Table 1-1a or Table 1-1b now to briefly review the base building and other cases.

We used four sets of weather data. See Section 1.3.1 for more details on weather data.

1.2.2 Modeling Rules

1.2.2.1 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 software gives a choice of methods for modeling indoor air distribution fans, use the same indoor fan modeling method for all cases. For the purpose of generating the example results, the IEA Solar Heating and Cooling (SHC) Task 22 participants used the most detailed level of modeling their programs allowed that was consistent with the level of detail provided in this test specification.

1.2.2.2 Nonapplicable Inputs

In some instances, the specification will include input values that do not apply to the input structure of your program. For example, your program (1) may not allow you to specify variation of cooling system sensible capacity with entering dry bulb temperature, (2) may not require an evaporator coil geometry description, (3) may not use the listed combined convective/radiative film coefficients, or (4) may not apply other listed inputs. When nonapplicable input values are found, either use the approximation methods suggested in your user manual, or simply disregard the nonapplicable inputs and continue. Such inputs are in the specifications for those programs that may need them.

1.2.2.3 Time Convention

References to time in this specification are to local standard time. Assume that: *hour 1 = the interval from midnight to 1 A.M.* Do not use daylight savings time or holidays for scheduling. However, the TMY data are in hourly bins corresponding to solar time as described in Section 1.3.1. The equivalent TMY2 data are in hourly bins corresponding to local standard time.

1.2.2.4 Geometry Convention

If your program includes the thickness of walls in a three-dimensional definition of the building geometry, define wall, roof, and floor thicknesses so that the interior air volume of the building remains as specified (6 m x 8 m x 2.7 m = 129.6 m^3) Make the thicknesses extend exterior to the currently defined internal volume.

1.2.2.5 Simulation Initialization

If your software allows, begin the simulation initialization process with zone air conditions that equal the outdoor air conditions. Preliminary sensitivity tests indicate that differences in initialization starting points can affect the resulting zone air humidity ratio in the dry-coil cases.

1.2.2.6 Simulation Preconditioning

If your program allows for preconditioning (iterative simulation of an initial time period until temperatures or fluxes, or both, stabilize at initial values), use that capability.

1.2.2.7 Simulation Duration

Run the simulation for at least the first 2 months for which the weather data are provided. Give output for the second month of the simulation (February) per Section 1.2.3. The first month of the simulation period (January) serves as an initialization period. Weather data for the third month (March) are included because at least one of the simulation programs used in the field trials required some additional weather data to be able to run a 2-month simulation.

1.2.3 Output Requirements

Enter all your output data into the preformatted spreadsheet with the file name HVBTOUT.XLS on the enclosed diskette. Instructions for using the spreadsheet appear at the top of the spreadsheet and in Appendix E.

The outputs listed immediately below are to include loads or consumptions (as appropriate) for the entire month of February (the second month in the 3-month weather data sets). The terms “cooling energy consumption,” “evaporator coil loads,” “zone cooling loads,” and “coefficient of performance” are defined in Appendix H.

- Cooling energy consumptions (kWh)
    - Total consumption (compressor and fans)
    - Disaggregated compressor consumption
    - Disaggregated indoor air distribution fan consumption
    - Disaggregated outdoor condenser fan consumption
- Evaporator coil loads (kWh)
    - Total evaporator coil load (sensible + latent)
    - Disaggregated sensible evaporator coil load
    - Disaggregated latent evaporator coil load
- Zone cooling loads (kWh)
    - Total cooling load (sensible + latent)
    - Disaggregated sensible cooling load
    - Disaggregated latent cooling load.

The outputs listed immediately below are to include the mean value for the month of February, and the hourly-integrated maximum and minimum values for the month of February.

- Calculated coefficient of performance (COP) (dimensionless)
  ((Net refrigeration effect)/(total cooling energy consumption))
- Zone dry bulb temperature (°C)
- Zone humidity ratio (kg moisture/kg dry air).

1.2.4 Comparing Your Output to the Analytical Solution and Example Simulation Results

As a minimum, the user should compare output with the analytical solution results found in Part II. The user may also choose to compare output with the example simulation results in Part IV, or with other results that were generated using this test procedure. Information about how the analytical solutions and example simulation results were produced is included in Parts II and III, respectively. For convenience to users who wish to plot or tabulate their results along with the analytical solution or example simulation results, or both, an electronic version of the example results has been included with the file RESULTS.XLS on the accompanying electronic media.

1.2.4.1 Criteria for Determining Agreement between Results

No formal criteria exist for when results agree or disagree; determining the agreement or disagreement of results is left to the user. In making this determination, the user should consider that the analytical solution results represent a “mathematical truth standard” (i.e., a mathematically provable and deterministic set of results based on acceptance of the underlying physical assumptions represented by the case specifications). The authors recognize that although the underlying physical assumptions of the case definitions of the mechanical equipment are consistent with those of typical manufacturer equipment performance data, they are by definition a simplification of reality and may not fully represent real empirical behavior.

In making a determination about the agreement of results, the user should also consider:

- The magnitude of results for individual cases
- The magnitude of difference in results between certain cases (e.g., “Case E110–Case E100”)
- The same direction of sensitivity (positive or negative) for difference in results between certain cases (e.g., “Case E110–Case E100”)
- The degree of disagreement that occurred for other simulation results in Part IV versus the analytical solution results.

1.2.4.2 Diagnostic Logic for Determining Causes of Differences among Results

To help you identify which algorithm in the tested program is causing specific differences between programs, we have included a diagnostic logic flow chart in Appendix F.

1.3 Specific Input Information

Assembling an accurate base building and mechanical system represents the bulk of the work for implementing this test. We recommend that you double check all inputs. Weather data, building zone, and mechanical equipment details are described topically in the following subsections.

1.3.1 Weather Data

Use the TMY **or** TMY2 format weather data provided on the diskette. These data represent typical TMY and TMY2 weather data files, respectively, with modifications so that the initial fundamental series of mechanical equipment tests may be very tightly controlled. For the purposes of HVAC BESTEST, which uses a near-adiabatic building envelope, the TMY and TMY2 data sets are equivalent. (Note that there are small differences in solar radiation, wind speed, etc., that result in a sensible loads difference of 0.2%–0.3% in cases with low internal gains [i.e. E130, E140, E190, and E195]. This percentage load difference is less [0.01%–0.04%] for the other cases because they have higher internal gains. These TMY and TMY2 data are not equivalent for use with a non-near-adiabatic building envelope.)

Four 3-month-long weather data files are used in the test suite:

- HVBT294.TM?
- HVBT350.TM?
- HVBT406.TM?
- HVBT461.TM?

where ".TM?" corresponds to either TMY (.TMY) or TMY2 (.TM2) format, whichever is preferred.

Ambient dry bulb and dew point temperatures are constant in all the weather files; constant values of ambient dry bulb vary among the files. Site and weather characteristics are summarized in Tables 1-2a and 1-2b for the TMY and TMY2 data files, respectively. See Appendix A for details about the TMY and TMY2 weather data file formats.

The hourly time convention for TMY weather data is solar time, where

Solar time = standard time + 4 minutes/degree x (Lst - Lloc) + E,

and where:

Standard time = local standard time
Lst = standard meridian longitude (degrees)
Lloc = local site longitude (degrees)
$E = 9.87 \sin 2B - 7.53 \cos B - 1.5 \sin B$ (minutes),

where

$B = 360(n - 81)/364$ (degrees)
$n \equiv$ day of the year, $1 \leq n \leq 365$.

E varies roughly ± 15 min throughout the year because of cosmology. Additional information on the equation of time may be found in the references (Duffie and Beckman 1980).

The hourly time convention for TMY2 weather data is local standard time.

Table 1-2a. Site and Weather Summary—TMY Data

Weather Type		Artificial Conditions
Weather Format		TMY
Latitude		25.8° North
Longitude (local site)		80.3° West
Altitude		2 m (6.6 ft)
Time Zone (Standard Meridian Longitude)		5 (75° West)
Ground Reflectivity		0.2
Site		Flat, unobstructed, located exactly at weather station
Dew Point Temperature (constant)		14.0°C (57.2°F)
Humidity Ratio		0.010 kg moisture/kg dry air (0.010 lb moisture/lb dry air)
Mean 3-Month Wind Speed		4.4 m/s (9.8 miles/h)
Maximum 3-Month Wind Speed		12.4 m/s (27.7 miles/h)
Global Horizontal Solar Radiation 3-Month Total		1354 MJ/m² (119.2 kBtu/ft²)
Direct Normal Solar Radiation 3-Month Total		1350 MJ/m² (118.8 kBtu/ft²)
Direct Horizontal Solar Radiation 3-Month Total		817 MJ/m² (71.9 kBtu/ft²)
Diffuse Horizontal Solar Radiation 3-Month Total		536 MJ/m² (47.2 kBtu/ft²)
Quantities That Vary Between Data Sets	Ambient Dry Bulb Temperature (constant)	Ambient Relative Humidity
HVBT294.TMY	29.4°C (85.0°F)	39%
HVBT350.TMY	35.0°C (95.0°F)	28%
HVBT406.TMY	40.6°C (105.0°F)	21%
HVBT461.TMY	46.1°C (115.0°F)	16%

Table 1-2b. Site and Weather Summary—TMY2 Data

Weather Type		Artificial Conditions
Weather Format		TMY2
Latitude		25.8° North
Longitude (local site)		80.3° West
Altitude		2 m (6.6 ft)
Time Zone (Standard Meridian Longitude)		5 (75° West)
Ground Reflectivity		0.2
Site		Flat, unobstructed, located exactly at weather station
Dew Point Temperature (constant)		14.0°C (57.2°F)
Humidity Ratio		0.010 kg moisture/kg dry air (0.010 lb moisture/lb dry air)
Mean 3-Month Wind Speed		4.9 m/s (11.0 miles/h)
Maximum 3-Month Wind Speed		11.8 m/s (26.4 miles/h)
Global Horizontal Solar Radiation 3-Month Total		1412 MJ/m² (124.3 kBtu/ft²)
Direct Normal Solar Radiation 3-Month Total		1460 MJ/m² (124.3 kBtu/ft²)
Diffuse Horizontal Solar Radiation 3-Month Total		557 MJ/m² (49.1 kBtu/ft²)
Quantities That Vary Between Data Sets	Ambient Dry Bulb Temperature (constant)	Ambient Relative Humidity
HVBT294.TM2	29.4°C (85.0°F)	39%
HVBT350.TM2	35.0°C (95.0°F)	28%
HVBT406.TM2	40.6°C (105.0°F)	21%
HVBT461.TM2	46.1°C (115.0°F)	16%

1.3.2 Case E100 Base Case Building and Mechanical System

1.3.2.1 Building Zone Description

1.3.2.1.1 Building Geometry. The base building is a 48-m² floor area, single-story, low-mass building with rectangular-prism geometry as shown in Figure 1-1. Zone air volume is 129.6 m^3.

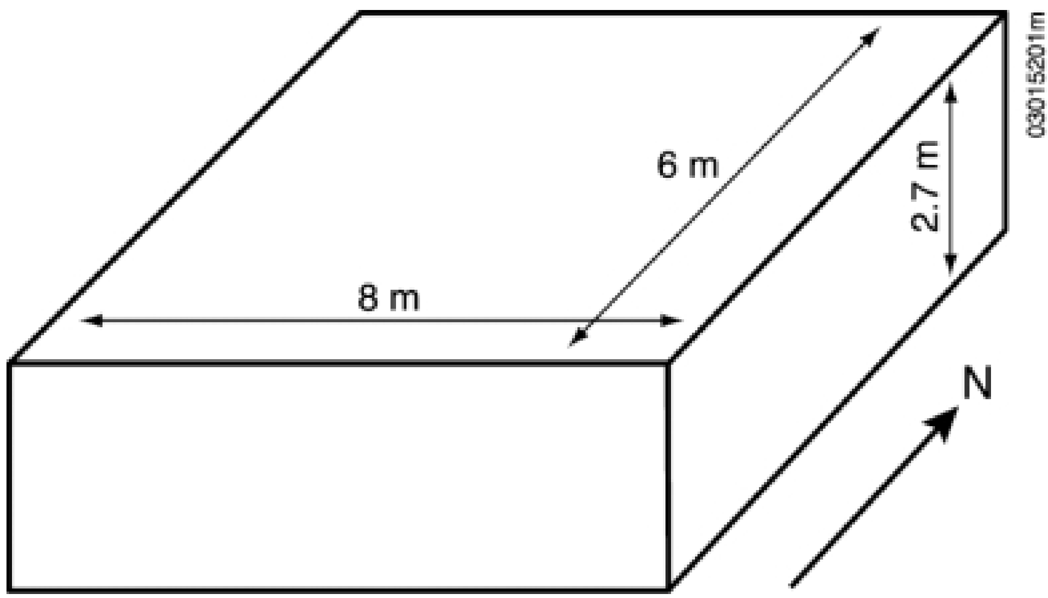


Figure 1-1. HVAC BESTEST: Near-adiabatic envelope geometry

1.3.2.1.2 Building Envelope Thermal Properties. The base building zone is intended as a near-adiabatic test cell with cooling load driven by user-specified internal gains. Tables 1-3a and 1-3b list material properties in Système Internationale (SI) and English units, respectively. The building insulation has been made very thick to effectively thermally decouple the zone from ambient conditions. Materials of the space have no thermal or moisture capacitance, and there is no moisture diffusion through them. If your software requires inputs for any of thermal capacitance, moisture capacitance, or moisture diffusion, use the minimum values your software allows.

If your software does not allow this much insulation, use the thickest insulation your program will permit and reduce the floor, roof, and wall areas to achieve the thermal conductance (UA) values listed in Tables 1-3a or 1-3b. The zone air volume must remain at 129.6 m^3.

Air density at sea level is 1.201 kg/m^3.

The floor has the same exterior film coefficient as the other walls, as if the entire zone were suspended above the ground.

Table 1-3a. Material Specifications Base Case (SI Units)

mspec4.xls; May 10, 2001

EXTERIOR WALL (inside to outside)				
	k	Thickness	U	R
ELEMENT	(W/m*K)	(m)	(W/m2*K)	(m2*K/W)
Int Surf Coef			8.290	0.121
Insulation (Note 1)	0.010	1.000	0.010	100.000
Ext Surf Coef			29.300	0.034
Total air - air			0.010	100.155
Total surf - surf			0.010	100.000

FLOOR (inside to outside)				
	k	Thickness	U	R
ELEMENT	(W/m*K)	(m)	(W/m2*K)	(m2*K/W)
Int Surf Coef (Note 2)			8.290	0.121
Insulation (Note 1)	0.010	1.000	0.010	100.000
Ext Surf Coef			29.300	0.034
Total air - air			0.010	100.155
Total surf - surf			0.010	100.000

ROOF (inside to outside)				
	k	Thickness	U	R
ELEMENT	(W/m*K)	(m)	(W/m2*K)	(m2*K/W)
Int Surf Coef (Note 2)			8.290	0.121
Insulation (Note 1)	0.010	1.000	0.010	100.000
Ext Surf Coef			29.300	0.034
Total air - air			0.010	100.155
Total surf - surf			0.010	100.000

SUMMARY		
	AREA	UA
COMPONENT	(m2)	(W/K)
Wall	75.600	0.755
Floor	48.000	0.479
Roof	48.000	0.479
Infiltration (Note 3)		0.000
Total UA		1.713

	ACH	VOLUME (m3)	ALTITUDE (m)
	0.00	129.6	2.0

Note 1: This level of insulation defines a near-adiabatic condition such that conduction gains are < 1% of the total cooling load. If your software does not allow this much insulation, then reduce the floor, roof and wall areas to achieve the listed UA values.

Note 2: The interior film coefficient for floors and ceilings is a compromise between upward and downward heat flow for summer and winter.

Note 3: Infiltration derived from:
ACH*Volume*(specific heat of air)*(density of air at specified altitude)

Table 1-3b. Material Specifications Base Case (English Units)

mspec4.xls; May 10, 2001

EXTERIOR WALL (inside to outside)				
	k	Thickness	U	R
ELEMENT	Btu/(h*ft*F)	(ft)	Btu/(h*ft2*F)	h*ft2*F/Btu
Int Surf Coef			1.461	0.684
Insulation (Note 1)	0.00578	3.281	0.002	567.447
Ext Surf Coef			5.163	0.194
Total air - air			0.00176	568.325
Total surf - surf			0.00176	567.447

FLOOR (inside to outside)				
	k	Thickness	U	R
ELEMENT	Btu/(h*ft*F)	(ft)	Btu/(h*ft2*F)	h*ft2*F/Btu
Int Surf Coef (Note 2)			1.461	0.684
Insulation (Note 1)	0.00578	3.281	0.002	567.447
Ext Surf Coef			5.163	0.194
Total air - air			0.00176	568.325
Total surf - surf			0.00176	567.447

ROOF (inside to outside)				
	k	Thickness	U	R
ELEMENT	Btu/(h*ft*F)	(ft)	Btu/(h*ft2*F)	h*ft2*F/Btu
Int Surf Coef (Note 2)			1.461	0.684
Insulation (Note 1)	0.00578	3.281	0.002	567.447
Ext Surf Coef			5.163	0.194
Total air - air			0.00176	568.325
Total surf - surf			0.00176	567.447

SUMMARY		
	AREA	UA
COMPONENT	(ft2)	(Btu/h*F)
Wall	813.752	1.432
Floor	516.668	0.909
Roof	516.668	0.909
Infiltration		0.000
Total UA		3.250

	ACH	VOLUME (ft3)	ALTITUDE (ft)	UAinf (Note 3) (Btu/h*F)
	0.000	4577	6.56	0.000

Note 1: This level of insulation defines a near-adiabatic condition such that conduction gains are < 1% of the total cooling load. If your software does not allow this much insulation, then reduce the floor, roof and wall areas to achieve the listed UA values.

Note 2: The interior film coefficient for floors and ceilings is a compromise between upward and downward heat flow for summer and winter.

Note 3: Infiltration derived from:
ACH*Volume*(specific heat of air)*(density of air at specified altitude)

1.3.2.1.3 Weather Data.

HVBT461.TMY or HVBT461.TM2.

1.3.2.1.4 Infiltration.

Infiltration rate = 0.0 ACH (air changes per hour), for the entire simulation period.

1.3.2.1.5 Internal Heat Gains.

Sensible internal gains = 5400 W (18430 Btu/h), continuously.

Latent internal gains = 0 W (0 Btu/h), continuously.

Sensible gains are 100% convective.

Zone sensible and latent internal gains are assumed to be distributed evenly throughout the zone air. These are internally generated sources of heat that are not related to the operation of the mechanical cooling system or its air distribution fan.

1.3.2.1.6 Opaque Surface Radiative Properties. Interior and exterior opaque surface solar (visible and ultraviolet wavelengths) absorptances and infrared emittances are included in Table 1-4.

Table 1-4. Opaque Surface Radiative Properties

	Interior Surface	**Exterior Surface**
Solar Absorptance	0.6	0.1
Infrared Emittance	0.9	0.9

1.3.2.1.7 Exterior Combined Radiative and Convective Surface Coefficients. If your program calculates exterior surface radiation and convection automatically, you may disregard this section. If your program does not calculate this effect, use 29.3 W/m^2K for all exterior surfaces. This value is based on a mean annual wind speed of 4.02 m/s for a surface with roughness equivalent to rough plaster or brick.

1.3.2.1.8 Interior Combined Radiative and Convective Surface Coefficients. If your program calculates interior surface radiation and convection automatically, you may disregard this section. If your program does not calculate these effects, use the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) constant combined radiative and convective coefficients given in Table 1-5. (Note that the ASHRAE values are not exactly the same as the Chartered Institution of Building Services Engineers [CIBSE] values.)

Table 1-5. Interior Combined Surface Coefficient versus Surface Orientation

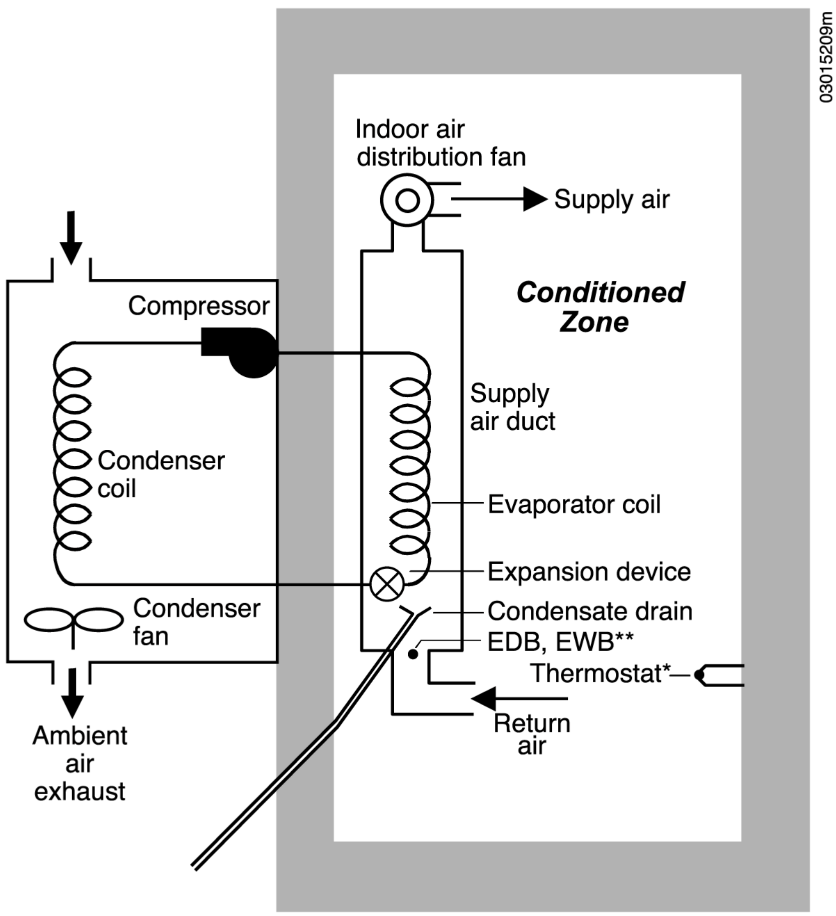
Orientation of Surface and Heat Flow	**Interior Combined Surface Coefficient**
Horizontal heat transfer on vertical surfaces	8.29 W/m^2K (1.46 $Btu/(hft^2F)$)
Upward heat transfer on horizontal surfaces	9.26 W/m^2K (1.63 $Btu/(hft^2F)$)
Downward heat transfer on horizontal surfaces	6.13 W/m^2K (1.08 $Btu/(hft^2F)$)

The radiative portion of these combined coefficients may be taken as: 5.13 W/m^2K (0.90 $Btu/(hft^2F)$) for an interior infrared emissivity of 0.9.

If your program does not allow you to schedule these coefficients, use 8.29 W/m^2K (1.46 $Btu/(hft^2F)$) for all horizontal surfaces when interior infrared emissivity is 0.9. If you can justify using different values, go ahead and use them.

1.3.2.2 Mechanical System

The mechanical system represents a simple vapor compression cooling system, or more precisely a unitary split air-conditioning system consisting of an air-cooled condensing unit and indoor evaporator coil. Figure 1-2 is a schematic diagram of this system. See the Glossary (Appendix H) for definitions of some terminology used in this section.



* The thermostat senses only the zone air temperature (IDB)

** Location of entering dry bulb and wet bulb temperatures

Figure 1-2. Unitary split air-conditioning system consisting of an air-cooled condensing unit and indoor evaporator coil

1.3.2.2.1 General Information.

- 100% convective air system
- Zone air is perfectly mixed
- No outside air; no exhaust air
- Single-speed, draw-through air distribution fan
- Indoor and outdoor fans cycle on and off together with compressor
- Air-cooled condenser
- Single-speed reciprocating compressor, R-22 refrigerant, no cylinder unloading
- No system hot gas bypass
- The compressor, condenser, and condenser fan are located outside the conditioned zone
- All zone air moisture that condenses on the evaporator coil (latent load) leaves the system through a condensate drain
- Crankcase heater and other auxiliary energy = 0.

Note that we found in NREL's DOE-2 simulations that simultaneous use of "0" outside air and "0" infiltration caused an error in the simulations. We worked around this by specifying minimum outside air = 0.000001 ft^3/min. We recommend that you run a sensitivity test to check that using 0 for both these inputs does not cause a problem.

1.3.2.2.2 Thermostat Control Strategy.

Heat = off
Cool = on if temperature > 22.2°C (72.0°F); otherwise Cool = off.

There is no zone humidity control. This means that the zone humidity level will float in accordance with zone latent loads and moisture removal by the mechanical system.

The thermostat senses only the zone air temperature; the thermostat itself does not sense any radiative heat transfer exchange with the interior surfaces.

The controls for this system are ideal in that the equipment is assumed to maintain the set point exactly, when it is operating and not overloaded. 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°C or OFF at set point - y°C). If your software requires input for these, use the minimum values your software allows.

The thermostat is nonproportional in the sense 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 model can be made to approximate a nonproportional thermostat model by setting a very small throttling range (the minimum allowed by your program). A COP=f(PLR) curve is given in Section 1.3.2.2.4 to account for equipment cycling.

1.3.2.2.3 Full-Load Cooling System Performance Data. Equipment full-load capacity and full-load performance data are given in six formats in Tables 1-6a through 1-6f. Before using these tables, read all of the discussion in this section (1.3.2.2.3) and its subsections (1.3.2.2.3.1 through 1.3.2.2.3.7). Use the table that most closely matches your software's input requirements. The tables contain similar information with the following differences:

- Table 1-6a lists net capacities (metric units)
- Table 1-6b lists net capacities (English units)
- Table 1-6c lists gross capacities (metric units)
- Table 1-6d lists gross capacities (English units)
- Table 1-6e lists adjusted net capacities (metric units)
- Table 1-6f lists adjusted net capacities (English units).

For your convenience, an electronic file (PERFMAP.WK3) that contains these tables is included on the accompanying compact disc (CD).

The meaning of the various ways to represent system capacity is discussed below; specific terms are also defined in the Glossary (Appendix H). These tables use outdoor dry-bulb temperature (ODB), entering dry-bulb temperature (EDB), and entering wet-bulb temperature (EWB) as independent variables for performance data; the location of EDB and EWB are shown in Figure 1-2.

Listed capacities of Tables 1-6a and 1-6b are net values after subtracting manufacturer default fan heat based on 365 W per 1,000 cubic feet per minute (CFM), so that the default fan heat for the 900-CFM fan is 329 W. For example, in Table 1-6a the listed net total capacity at Air-Conditioning and Refrigeration Institute (ARI) rating conditions (EDB = 26.7°C, outdoor dry-bulb temperature [ODB] = 35.0°C, EWB = 19.4°C) is 7852 W, and the assumed fan heat is 329 W. Therefore, the gross total capacity (see Table 1-6c) of the system at ARI rating conditions—including both the net total capacity and the distribution system fan heat—is 7,852 + 329 = 8,181 W. Similarly, the gross sensible capacity—including both the net sensible capacity and air distribution system fan heat—is 6,040 + 329 = 6,369 W.

The unit as described actually uses a 230-W fan. Therefore, the "real" net capacity is actually an adjusted net capacity, $(\text{net cap})_{adj}$, which is determined by:

$$(\text{net cap})_{adj} = (\text{net cap})_{listed} + (\text{default fan heat}) - (\text{actual fan power}),$$

so for the adjusted net total (sensible + latent) capacity at ARI conditions and 900 CFM:

$$(\text{net cap})_{adj} = 7852 \text{ W} + 329 \text{ W} - 230 \text{ W} = 7951 \text{ W}.$$

The technique for determining adjusted net sensible capacities (see Table 1-6e) is similar.

Table 1-6a: Equipment full-load performance[1] (Metric)

ODB (°C)	EWB (°C)	Net Total Capacity[2,3,4] (kW)	Net Sensible Capacity[2,3] (kW) at entering drybulb temperature (EDB, °C) 22.2	23.3	24.4	25.6	26.7	Compressor Power[4] (kW)	Apparatus Dew Point[4] (°C)
29.4	15.0	7.09	6.21	6.77	7.18	7.38	7.56	1.62	8.9
	17.2	7.68	5.16	5.71	6.27	6.80	7.35	1.66	11.1
	19.4	8.32	4.01	4.57	5.13	5.65	6.21	1.71	13.4
	21.7	8.97	2.87	3.40	3.96	4.48	5.04	1.76	15.8
32.2	15.0	6.91	6.12	6.68	7.03	7.21	7.41	1.69	9.1
	17.2	7.47	5.10	5.63	6.18	6.71	7.27	1.74	11.3
	19.4	8.09	3.93	4.48	5.04	5.57	6.12	1.79	13.6
	21.7	8.70	2.75	3.31	3.87	4.39	4.95	1.84	15.9
35.0	15.0	6.71	6.07	6.59	6.89	7.06	7.24	1.77	9.3
	17.2	7.27	5.01	5.54	6.09	6.62	7.18	1.81	11.4
	19.4	7.85	3.84	4.39	4.95	5.48	6.04	1.86	13.8
	21.7	8.47	2.67	3.22	3.75	4.31	4.86	1.91	16.2
37.8	15.0	6.53	5.98	6.53	6.71	6.89	7.06	1.85	9.4
	17.2	7.06	4.92	5.45	6.01	6.53	7.06	1.89	11.6
	19.4	7.62	3.75	4.31	4.83	5.39	5.95	1.94	13.9
	21.7	8.17	2.58	3.14	3.66	4.22	4.75	1.98	16.3
40.6	15.0	6.33	5.89	6.39	6.53	6.71	6.89	1.94	9.6
	17.2	6.83	4.83	5.36	5.92	6.45	6.89	1.98	11.8
	19.4	7.35	3.66	4.22	4.75	5.30	5.83	2.02	14.2
	21.7	7.91	2.49	3.02	3.57	4.13	4.66	2.06	16.6
46.1	15.0	5.92	5.71	6.04	6.21	6.36	6.50	2.11	10.0
	17.2	6.39	4.66	5.19	5.74	6.27	6.50	2.14	12.2
	19.4	6.89	3.49	4.04	4.57	5.13	5.65	2.18	14.6
	21.7	7.38	2.31	2.84	3.40	3.93	4.48	2.21	16.9

Values at ARI Rating Conditions (EDB = 26.7°C, EWB = 19.4°C, ODB = 35.0°C)

Net Total Capacity	7852	W
Airflow	0.425	m^3/s
Apparatus Dew Point	13.8	°C
Compressor Power	1858	W
Indoor Fan Power	230	W
Outdoor Fan Power	108	W
COP	3.62	

Seasonal Efficiency Rating

COP_{SEER}	3.78	

Abbreviations: ODB = outdoor drybulb temperature; EWB = entering wetbulb temperature; EDB = entering drybulb temperature; ARI = Air-Conditioning and Refrigeration Institute; COP = coefficient of performance; COP_{SEER} = dimensionless Seasonal Energy Efficiency Ratio.

Notes:

[1] Full-load performance data, courtesy Trane Co., Tyler, Texas, USA. Data is for "TTP024C with TWH039P15-C" at 900 CFM, published April 1993. Performance rated with 25 feet of 3/4" suction and 5/16" liquid lines.

[2] Listed net total and net sensible capacities are gross total and gross sensible capacities respectively, with manufacturer default fan heat (329 W) deducted.

[3] Where (Sensible Capacity) > (Total Capacity) indicates dry coil condition; in such case (Total Capacity) = (Sensible Capacity).

[4] Compressor kW, Apparatus Dew Point, and Net Total Capacity valid only for wet coil.

perfmap.xls, b:a3..j54; Apr 06, 2001

Table 1-6b: Equipment full-load performance[1] (English)

ODB (°F)	EWB (°F)	Net Total Capacity[2,3,4] (kBtu/h)	Net Sensible Capacity[2,3] (kBtu/h) at entering drybulb temperature (EDB, °F) 72	74	76	78	80	Compressor Power[4] (kW)	Apparatus Dew Point[4] (°F)
	59	24.2	21.2	23.1	24.5	25.2	25.8	1.62	48.1
	63	26.2	17.6	19.5	21.4	23.2	25.1	1.66	52.0
85	67	28.4	13.7	15.6	17.5	19.3	21.2	1.71	56.1
	71	30.6	9.8	11.6	13.5	15.3	17.2	1.76	60.4
	59	23.6	20.9	22.8	24.0	24.6	25.3	1.69	48.4
	63	25.5	17.4	19.2	21.1	22.9	24.8	1.74	52.3
90	67	27.6	13.4	15.3	17.2	19.0	20.9	1.79	56.5
	71	29.7	9.4	11.3	13.2	15.0	16.9	1.84	60.7
	59	22.9	20.7	22.5	23.5	24.1	24.7	1.77	48.7
	63	24.8	17.1	18.9	20.8	22.6	24.5	1.81	52.6
95	67	26.8	13.1	15.0	16.9	18.7	20.6	1.86	56.8
	71	28.9	9.1	11.0	12.8	14.7	16.6	1.91	61.1
	59	22.3	20.4	22.3	22.9	23.5	24.1	1.85	49.0
	63	24.1	16.8	18.6	20.5	22.3	24.1	1.89	52.9
100	67	26.0	12.8	14.7	16.5	18.4	20.3	1.94	57.1
	71	27.9	8.8	10.7	12.5	14.4	16.2	1.98	61.4
	59	21.6	20.1	21.8	22.3	22.9	23.5	1.94	49.3
	63	23.3	16.5	18.3	20.2	22.0	23.5	1.98	53.2
105	67	25.1	12.5	14.4	16.2	18.1	19.9	2.02	57.5
	71	27.0	8.5	10.3	12.2	14.1	15.9	2.06	61.8
	59	20.2	19.5	20.6	21.2	21.7	22.2	2.11	50.0
	63	21.8	15.9	17.7	19.6	21.4	22.2	2.14	53.9
115	67	23.5	11.9	13.8	15.6	17.5	19.3	2.18	58.2
	71	25.2	7.9	9.7	11.6	13.4	15.3	2.21	62.5

Values at ARI Rating Conditions (EDB = 80°F, EWB = 67°F, ODB = 95°F)

Net Total Capacity	26800	Btu/h
Airflow	900	CFM
Apparatus Dew Pt	56.8	°F
Compressor Power	1858	W
Indoor Fan Power	230	W
Outdoor Fan Power	108	W
EER	12.36	(Btu/h)/W

Seasonal Efficiency Rating

SEER	12.90	(Btu/h)/W

Abbreviations: ODB = outdoor drybulb temperature; EWB = entering wetbulb temperature; EDB = entering drybulb temperature; ARI = Air-Conditioning and Refrigeration Institute; EER = energy efficiency ration; SEER = seasonal energy efficiency ratio.

Notes:

[1] Full-load performance data, courtesy Trane Co., Tyler, Texas, USA. Data is for "TTP024C with TWH039P15-C" at 900 CFM, published April 1993. Performance rated with 25 feet of 3/4" suction and 5/16" liquid lines.

[2] Listed net total and net sensible capacities are gross total and gross sensible capacities respectively, with manufacturer default fan heat (1.12 kBtu/h) deducted.

[3] Where (Sensible Capacity) > (Total Capacity) indicates dry coil condition; in such case (Total Capacity) = (Sensible Capacity).

[4] Compressor kW, Apparatus Dew Point, and Net Total Capacity valid only for wet coil.

perfmap.xls, a:a12..j62; apr 06, 2001

Table 1-6c: Equipment full-load performance with gross capacities[1] (Metric)

ODB (°C)	EWB (°C)	Gross Total Capacity[2,3,4] (kW)	Gross Sensible Capacity[2,3] (kW) at entering drybulb temperature (EDB, °C) 22.2	23.3	24.4	25.6	26.7	Compressor Power[4] (kW)	Apparatus Dew Point[4] (°C)
29.4	15.0	7.42	6.54	7.10	7.51	7.71	7.89	1.62	8.9
	17.2	8.01	5.49	6.04	6.60	7.13	7.68	1.66	11.1
	19.4	8.65	4.34	4.90	5.46	5.98	6.54	1.71	13.4
	21.7	9.29	3.20	3.73	4.28	4.81	5.37	1.76	15.8
32.2	15.0	7.24	6.45	7.01	7.36	7.54	7.74	1.69	9.1
	17.2	7.80	5.43	5.95	6.51	7.04	7.60	1.74	11.3
	19.4	8.42	4.26	4.81	5.37	5.90	6.45	1.79	13.6
	21.7	9.03	3.08	3.64	4.20	4.72	5.28	1.84	15.9
35.0	15.0	7.04	6.39	6.92	7.21	7.39	7.57	1.77	9.3
	17.2	7.60	5.34	5.87	6.42	6.95	7.51	1.81	11.4
	19.4	8.18	4.17	4.72	5.28	5.81	6.36	1.86	13.8
	21.7	8.80	3.00	3.55	4.08	4.64	5.19	1.91	16.2
37.8	15.0	6.86	6.31	6.86	7.04	7.21	7.39	1.85	9.4
	17.2	7.39	5.25	5.78	6.34	6.86	7.39	1.89	11.6
	19.4	7.95	4.08	4.64	5.16	5.72	6.28	1.94	13.9
	21.7	8.50	2.91	3.46	3.99	4.55	5.08	1.98	16.3
40.6	15.0	6.66	6.22	6.72	6.86	7.04	7.21	1.94	9.6
	17.2	7.16	5.16	5.69	6.25	6.77	7.21	1.98	11.8
	19.4	7.68	3.99	4.55	5.08	5.63	6.16	2.02	14.2
	21.7	8.24	2.82	3.35	3.90	4.46	4.99	2.06	16.6
46.1	15.0	6.25	6.04	6.36	6.54	6.69	6.83	2.11	10.0
	17.2	6.72	4.99	5.52	6.07	6.60	6.83	2.14	12.2
	19.4	7.21	3.82	4.37	4.90	5.46	5.98	2.18	14.6
	21.7	7.71	2.64	3.17	3.73	4.26	4.81	2.21	16.9

Values at ARI Rating Conditions (EDB = 26.7°C, EWB = 19.4°C, ODB = 35.0°C)

Gross Total Capacity	8181	W
Airflow	0.425	m^3/s
Apparatus Dew Point	13.8	°C
Compressor Power	1858	W
Indoor Fan Power	230	W
Outdoor Fan Power	108	W
COP	3.62	

Seasonal Efficiency Rating

COP_{SEER}	3.78	

Abbreviations: ODB = outdoor drybulb temperature; EWB = entering wetbulb temperature; EDB = entering drybulb temperature; ARI = Air-Conditioning and Refrigeration Institute; COP = coefficient of performance; COP_{SEER} = dimensionless Seasonal Energy Efficiency Ratio.

Notes:

1 Based on full-load performance data, courtesy Trane Co., Tyler, Texas, USA. Data is for "TTP024C with TWH039P15-C" at 900 CFM, published April 1993. Performance rated with 25 feet of 3/4" suction and 5/16" liquid lines.

2 Listed gross total and gross sensible capacities include manufacturer default fan heat of 329 W.

3 Where (Sensible Capacity) > (Total Capacity) indicates dry coil condition; in such case (Total Capacity) = (Sensible Capacity).

4 Compressor kW, Apparatus Dew Point, and Gross Total Capacity valid only for wet coil.

perfmap.xls, d:a3..j54; Mar 30, 2001

Table 1-6d: Equipment full-load performance with gross capacities[1] (English)

ODB (°F)	EWB (°F)	Gross Total Capacity[2,3,4] (kBtu/h)	Gross Sensible Capacity[2,3] (kBtu/h) at entering drybulb temperature (EDB, °F) 72	74	76	78	80	Compressor Power[4] (kW)	Apparatus Dew Point[4] (°F)
	59	25.3	22.3	24.2	25.6	26.3	26.9	1.62	48.1
	63	27.3	18.7	20.6	22.5	24.3	26.2	1.66	52.0
85	67	29.5	14.8	16.7	18.6	20.4	22.3	1.71	56.1
	71	31.7	10.9	12.7	14.6	16.4	18.3	1.76	60.4
	59	24.7	22.0	23.9	25.1	25.7	26.4	1.69	48.4
	63	26.6	18.5	20.3	22.2	24.0	25.9	1.74	52.3
90	67	28.7	14.5	16.4	18.3	20.1	22.0	1.79	56.5
	71	30.8	10.5	12.4	14.3	16.1	18.0	1.84	60.7
	59	24.0	21.8	23.6	24.6	25.2	25.8	1.77	48.7
	63	25.9	18.2	20.0	21.9	23.7	25.6	1.81	52.6
95	67	27.9	14.2	16.1	18.0	19.8	21.7	1.86	56.8
	71	30.0	10.2	12.1	13.9	15.8	17.7	1.91	61.1
	59	23.4	21.5	23.4	24.0	24.6	25.2	1.85	49.0
	63	25.2	17.9	19.7	21.6	23.4	25.2	1.89	52.9
100	67	27.1	13.9	15.8	17.6	19.5	21.4	1.94	57.1
	71	29.0	9.9	11.8	13.6	15.5	17.3	1.98	61.4
	59	22.7	21.2	22.9	23.4	24.0	24.6	1.94	49.3
	63	24.4	17.6	19.4	21.3	23.1	24.6	1.98	53.2
105	67	26.2	13.6	15.5	17.3	19.2	21.0	2.02	57.5
	71	28.1	9.6	11.4	13.3	15.2	17.0	2.06	61.8
	59	21.3	20.6	21.7	22.3	22.8	23.3	2.11	50.0
	63	22.9	17.0	18.8	20.7	22.5	23.3	2.14	53.9
115	67	24.6	13.0	14.9	16.7	18.6	20.4	2.18	58.2
	71	26.3	9.0	10.8	12.7	14.5	16.4	2.21	62.5

Values at ARI Rating Conditions (EDB = 80°F, EWB = 67°F, ODB = 95°F)

Gross Total Capacity	27920	Btu/h
Airflow	900	CFM
Apparatus Dew Pt	56.8	°F
Compressor Power	1858	W
Indoor Fan Power	230	W
Outdoor Fan Power	108	W
EER	12.36	(Btu/h)/W

Seasonal Efficiency Rating

SEER	12.90	(Btu/h)/W

Abbreviations: ODB = outdoor drybulb temperature; EWB = entering wetbulb temperature; EDB = entering drybulb temperature; ARI = Air-Conditioning and Refrigeration Institute; EER = energy efficiency ration; SEER = seasonal energy efficiency ratio.

Notes:

[1] Based on full-load performance data, courtesy Trane Co., Tyler, Texas, USA. Data is for "TTP024C with TWH039P15-C" at 900 CFM, published April 1993. Performance rated with 25 feet of 3/4" suction and 5/16" liquid lines.

[2] Listed gross total and gross sensible capacities include manufacturer default fan heat of 1.12 kBtu/h.

[3] Where (Sensible Capacity) > (Total Capacity) indicates dry coil condition; in such case (Total Capacity) = (Sensible Capacity).

[4] Compressor kW, Apparatus Dew Point, and Gross Total Capacity valid only for wet coil.

perfmap.xls, c:a3..j53; apr 06, 2001

Table 1-6e: Equipment full-load performance with adjusted net capacities[1] (Metric)

ODB (°C)	EWB (°C)	Adj Net Total Capacity[2,3,4] (kW)	Adjusted Net Sensible Capacity[2,3] (kW) at entering drybulb temperature (EDB, °C) 22.2	23.3	24.4	25.6	26.7	Compressor Power[4] (kW)	Apparatus Dew Point[4] (°C)
29.4	15.0	7.19	6.31	6.87	7.28	7.48	7.66	1.62	8.9
	17.2	7.78	5.26	5.81	6.37	6.90	7.45	1.66	11.1
	19.4	8.42	4.11	4.67	5.23	5.75	6.31	1.71	13.4
	21.7	9.06	2.97	3.50	4.05	4.58	5.14	1.76	15.8
32.2	15.0	7.01	6.22	6.78	7.13	7.31	7.51	1.69	9.1
	17.2	7.57	5.20	5.72	6.28	6.81	7.37	1.74	11.3
	19.4	8.19	4.03	4.58	5.14	5.67	6.22	1.79	13.6
	21.7	8.80	2.85	3.41	3.97	4.49	5.05	1.84	15.9
35.0	15.0	6.81	6.16	6.69	6.98	7.16	7.34	1.77	9.3
	17.2	7.37	5.11	5.64	6.19	6.72	7.28	1.81	11.4
	19.4	7.95	3.94	4.49	5.05	5.58	6.13	1.86	13.8
	21.7	8.57	2.77	3.32	3.85	4.41	4.96	1.91	16.2
37.8	15.0	6.63	6.08	6.63	6.81	6.98	7.16	1.85	9.4
	17.2	7.16	5.02	5.55	6.11	6.63	7.16	1.89	11.6
	19.4	7.72	3.85	4.41	4.93	5.49	6.05	1.94	13.9
	21.7	8.27	2.68	3.23	3.76	4.32	4.85	1.98	16.3
40.6	15.0	6.43	5.99	6.49	6.63	6.81	6.98	1.94	9.6
	17.2	6.93	4.93	5.46	6.02	6.54	6.98	1.98	11.8
	19.4	7.45	3.76	4.32	4.85	5.40	5.93	2.02	14.2
	21.7	8.01	2.59	3.12	3.67	4.23	4.76	2.06	16.6
46.1	15.0	6.02	5.81	6.13	6.31	6.46	6.60	2.11	10.0
	17.2	6.49	4.76	5.29	5.84	6.37	6.60	2.14	12.2
	19.4	6.98	3.59	4.14	4.67	5.23	5.75	2.18	14.6
	21.7	7.48	2.41	2.94	3.50	4.03	4.58	2.21	16.9

Values at ARI Rating Conditions (EDB = 26.7°C, EWB = 19.4°C, ODB = 35.0°C)

Adj Net Total Capacity	7951	W
Airflow	0.4248	m^3/s
Apparatus Dew Point	13.8	°C
Compressor Power	1858	W
Indoor Fan Power	230	W
Outdoor Fan Power	108	W
COP	3.62	

Seasonal Efficiency Rating

COP_{SEER} 3.78

Abbreviations: ODB = outdoor drybulb temperature; EWB = entering wetbulb temperature; EDB = entering drybulb temperature; ARI = Air-Conditioning and Refrigeration Institute; COP = coefficient of performance; COP_{SEER} = dimensionless Seasonal Energy Efficiency Ratio.

Notes:

[1] Based on full-load performance data, courtesy Trane Co., Tyler, Texas, USA. Data is for "TTP024C with TWH039P15-C" at 900 CFM, published April 1993. Performance rated with 25 feet of 3/4" suction and 5/16" liquid lines.

[2] Listed adjusted net total and adjusted net sensible capacities are gross capacities with actual fan heat (230 W) subtracted.

[3] Where (Sensible Capacity) > (Total Capacity) indicates dry coil condition; in such case (Total Capacity) = (Sensible Capacity).

[4] Compressor kW, Apparatus Dew Point, and Adjusted Net Total Capacity valid only for wet coil.

perfmap.xls, f:a3..j54; Apr 06, 2001

Table 1-6f: Equipment full-load performance with adjusted net capacities[1] (English)

ODB (°F)	EWB (°F)	Adj Net Total Capacity[2,3,4] (kBtu/h)	Adjusted Net Sensible Capacity[2,3] (kBtu/h) at entering drybulb temperature (EDB, °F) 72	74	76	78	80	Compressor Power[4] (kW)	Apparatus Dew Point[4] (°F)
	59	24.5	21.5	23.4	24.8	25.5	26.1	1.62	48.1
	63	26.5	17.9	19.8	21.7	23.5	25.4	1.66	52.0
85	67	28.7	14.0	15.9	17.8	19.6	21.5	1.71	56.1
	71	30.9	10.1	11.9	13.8	15.6	17.5	1.76	60.4
	59	23.9	21.2	23.1	24.3	24.9	25.6	1.69	48.4
	63	25.8	17.7	19.5	21.4	23.2	25.1	1.74	52.3
90	67	27.9	13.7	15.6	17.5	19.3	21.2	1.79	56.5
	71	30.0	9.7	11.6	13.5	15.3	17.2	1.84	60.7
	59	23.2	21.0	22.8	23.8	24.4	25.0	1.77	48.7
	63	25.1	17.4	19.2	21.1	22.9	24.8	1.81	52.6
95	67	27.1	13.4	15.3	17.2	19.0	20.9	1.86	56.8
	71	29.2	9.4	11.3	13.1	15.0	16.9	1.91	61.1
	59	22.6	20.7	22.6	23.2	23.8	24.4	1.85	49.0
	63	24.4	17.1	18.9	20.8	22.6	24.4	1.89	52.9
100	67	26.3	13.1	15.0	16.8	18.7	20.6	1.94	57.1
	71	28.2	9.1	11.0	12.8	14.7	16.5	1.98	61.4
	59	21.9	20.4	22.1	22.6	23.2	23.8	1.94	49.3
	63	23.6	16.8	18.6	20.5	22.3	23.8	1.98	53.2
105	67	25.4	12.8	14.7	16.5	18.4	20.2	2.02	57.5
	71	27.3	8.8	10.6	12.5	14.4	16.2	2.06	61.8
	59	20.5	19.8	20.9	21.5	22.0	22.5	2.11	50.0
	63	22.1	16.2	18.0	19.9	21.7	22.5	2.14	53.9
115	67	23.8	12.2	14.1	15.9	17.8	19.6	2.18	58.2
	71	25.5	8.2	10.0	11.9	13.7	15.6	2.21	62.5

Values at ARI Rating Conditions (EDB = 80°F, EWB = 67°F, ODB = 95°F)

Adj Net Total Capacity	27140	Btu/h
Airflow	900	CFM
Apparatus Dew Pt	56.8	°F
Compressor Power	1858	W
Indoor Fan Power	230	W
Outdoor Fan Power	108	W
EER	12.36	(Btu/h)/W

Seasonal Efficiency Rating

SEER	12.90	(Btu/h)/W

Abbreviations: ODB = outdoor drybulb temperature; EWB = entering wetbulb temperature; EDB = entering drybulb temperature; ARI = Air-Conditioning and Refrigeration Institute; EER = energy efficiency ration; SEER = seasonal energy efficiency ratio.

Notes:

1 Based on full-load performance data, courtesy Trane Co., Tyler, Texas, USA. Data is for "TTP024C with TWH039P15-C" at 900 CFM, published April 1993. Performance rated with 25 feet of 3/4" suction and 5/16" liquid lines.

2 Listed adjusted net total and adjusted net sensible capacities are gross capacities with actual fan heat (785 Btu/h) subtracted.

3 Where (Sensible Capacity) > (Total Capacity) indicates dry coil condition; in such case (Total Capacity) = (Sensible Capacity).

4 Compressor kW, Apparatus Dew Point, and Adjusted Net Total Capacity valid only for wet coil.

perfmap.xls, e:a3..j53; apr 06, 2001

1.3.2.2.3.1 Validity of Listed Data <u>(VERY IMPORTANT).</u> Compressor kW (kilowatts) and apparatus dew point, along with net total, gross total, and adjusted net total capacities given in Tables 1-6a through 1-6f are valid only for "wet" coils (when dehumidification is occurring). A dry-coil condition—no dehumidification —occurs when the entering air humidity ratio is decreased to the point where the entering air dew point temperature is less than the effective coil surface temperature (apparatus dew point). In Tables 1-6a through 1-6f, the dry-coil condition is evident from a given table for conditions where the listed sensible capacity is greater than the corresponding total capacity. For such a dry-coil condition, set total capacity equal to sensible capacity.

For a given EDB and ODB, the compressor power, total capacity, sensible capacity, and apparatus dew point for wet-coils change only with varying EWB. Once the coil becomes dry—which is apparent for a given EDB and ODB from the maximum EWB where total and sensible capacities are equal— for a given EDB compressor power and capacities remain constant with decreasing EWB. (Brandemuehl 1983; pp. 4-82–83)

To evaluate equipment performance for a dry-coil condition, establish the performance at the maximum EWB where total and sensible capacities are equal. Make this determination by interpolating or extrapolating with EWB for a given EDB and ODB. For example, to determine the dry-coil compressor power for ODB/EDB = 29.4°C/26.7°C, find the "maximum EWB" dry-coil condition (net sensible capacity = net total capacity) using the data shown in Table 1-7 (extracted from Table 1-6e):

Table 1-7. Determination of Maximum Dry-Coil EWB Using Interpolation

EWB (°C)	Adjusted Net Total Capacity (kW)	Adjusted Net Sensible Capacity (kW)	Compressor Power (kW)
15.0	**7.19**	**7.66**	**1.62**
*Maximum dry EWB 16.75**	*7.66**	*7.66**	*1.652**
17.2	**7.78**	**7.45**	**1.66**

* Italicized values are not specifically listed with Table 1-6e; they are determined based on the accompanying discussion. Data in bold font are from Table 1-6e.

At the dry-coil condition:

Adjusted net total capacity = adjusted net sensible capacity = 7.66 kW.

Linear interpolation based on adjusted net total capacity gives:

Maximum EWB for the dry-coil condition = 16.75°C

Compressor power = 1.652 kW.

1.3.2.2.3.2 Extrapolation of Performance Data. For Cases E100–E200, allow your software to perform the necessary extrapolations of the performance data as may be required by these cases, if it has that capability. Cases E100, E110, E130, and E140 require some extrapolation of data for EWB <15.0°C (<59°F). Additionally, Case E180 may require (depending on the model) a small amount of extrapolation of data for EWB >21.7°C (>71°F). Case E200 may require (depending on the model) some extrapolation of data for EDB >26.7°C (>80°F).

In cases where the maximum-EWB dry-coil condition occurs at EWB <15.0°C, extrapolate the total capacity and sensible capacity to the intersection point where they are both equal. For example, use the data shown in Table 1-8 (extracted from Table 1-6e) to find the maximum EWB dry-coil condition for ODB/EDB = 29.4°C/22.2°C:

Linear extrapolation of the total and sensible capacities to the point where they are equal gives:

Adjusted net total capacity = adjusted net sensible capacity = 6.87 kW
Maximum dry-coil EWB = 13.8°C
Resulting compressor power = 1.598 kW.

This technique is also illustrated in the analytical solutions presented in Part II.

Table 1-8. Determination of Maximum Dry-Coil EWB Using Extrapolation

EWB (°C)	Adjusted Net Total Capacity (kW)	Adjusted Net Sensible Capacity (kW)	Compressor Power (kW)
*Maximum dry EWB 13.8**	*6.87**	*6.87**	*1.598**
15.0	**7.19**	**6.31**	**1.62**
17.2	**7.78**	**5.26**	**1.66**

* Italicized values are not specifically listed with Table 1-6e; they are determined based on the accompanying discussion. Data in bold font are from Table 1-6e.

1.3.2.2.3.3 Apparatus Dew Point. Apparatus dew point (ADP) is defined in Appendix H. Listed values of ADP may vary somewhat from those calculated using the other listed performance parameters. For more discussion of this, see Appendix C (Cooling Coil Bypass Factor).

1.3.2.2.3.4 Values at ARI Rating Conditions. In Tables 1-6a through 1-6f, nominal values at ARI rating conditions are useful to system designers for comparing the capabilities of one system to those of another. Some detailed simulation programs utilize inputs for ARI rating conditions in conjunction with the full performance maps of Tables 1-6a through 1-6f. For simplified simulation programs and other programs that do not allow performance maps of certain parameters, appropriate values at ARI conditions may be used and assumed constant.

1.3.2.2.3.5 SEER. In Tables 1-6a through 1-6f, seasonal energy efficiency ratio (SEER), which is a generalized seasonal efficiency rating, is not generally a useful input for detailed simulation of mechanical systems. SEER (or "COP_{SEER}" in the metric versions) is useful to system designers for comparing one system to another. SEER is further discussed in the Glossary (Appendix H) and in Appendix B.

1.3.2.2.3.6 Cooling Coil Bypass Factor. If your software does not require an input for bypass factor (BF), or automatically calculates it based on other inputs, ignore this information.

BF at ARI rating conditions is *approximately*:

$$0.049 \leq BF \leq 0.080.$$

Calculation techniques and uncertainty about this range of values are discussed in Appendix C. We supply the information in Appendix C for illustrative purposes; some models may perform the calculation with minor differences in technique or assumptions, or both. If your software requires this input, calculate the BF in a manner consistent with the assumptions of your specific model. If the assumptions of your model are not apparent from its documentation, use a value consistent with the above range and Appendix C.

Calculations based on the listed performance data indicate that BF varies as a function of EDB, EWB, and ODB. Incorporate this aspect of equipment performance into your model if your software allows it, using a consistent method for developing all points of the BF variation map.

1.3.2.2.3.7 Minimum Supply Air Temperature. This system is a variable temperature system, meaning that the supply air temperature varies with the operating conditions. If your software requires an input for minimum allowable supply air temperature, use:

Minimum supply air temperature ≤ 7.7°C (45.9°F).

This value is the lowest value of ADP that occurs in the E100–E200 series cases based on the analytical solutions for E110 presented in Part II.

If your software does not require this input, ignore this information.

1.3.2.2.4 Part Load Operation. The system COP degradation that results from part load operation is described in Figure 1-3. In this figure the COP degradation factor (CDF) is a multiplier to be applied to the full-load system COP (as defined in Appendix H) at a given part load ratio (PLR), where:

COP(PLR) = (full load COP(ODB,EWB,EDB)) * CDF(PLR).

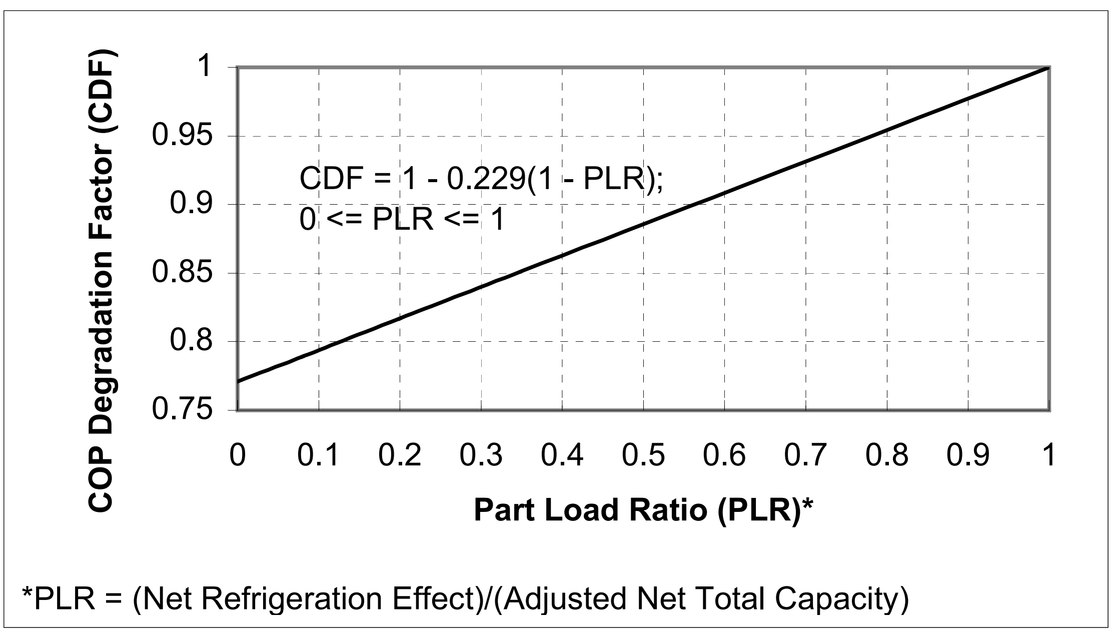


Figure 1-3. Cooling equipment part load performance (COP degradation factor versus PLR)

This representation is based on information provided by the equipment manufacturer (Cawley 1997). It might be helpful to think of the efficiency degradation as being caused by additional start-up run time required to bring the evaporator coil temperature down to its equilibrium temperature for the time(s) when the compressor is required to operate during an hour with part load.

Then, because the controller is nonproportional (see Section 1.3.2.2.2),

Hourly fractional run time = PLR/CDF.

In Figure 1-3, the PLR is calculated by:

$$\frac{\text{(Net refrigeration effect)}}{\text{(Adjusted net total capacity)}},$$

where the net refrigeration effect and the adjusted net total capacity are as defined in the Glossary (Appendix H). The adjusted net total capacity is a function of EWB, ODB, and EDB.

Simplifying assumptions in Figure 1-3 are:

- There is no minimum on/off time for the compressor and related fans; they may cycle on/off as often as necessary to maintain the set point.
- The decrease in efficiency with increased on/off cycling at very low PLR remains linear.

Appendix B includes additional details about how Figure 1-3 was derived.

If your software utilizes cooling coil bypass factor, model the BF as independent of (not varying with) the PLR (Cawley 1997).

1.3.2.2.5 Evaporator Coil. Geometry of the evaporator coil is included in Figures 1-4 and 1-5 (Houser 1997). Evaporator coil fins are actually contoured to enhance heat transfer. More details about fin geometry are proprietary, and therefore unavailable.

1.3.2.2.5.1 Frontal Dimensions (also see Figure 1-4).

- Height = 68.6 cm (27 in.)
- Width = 61.0 cm (24 in.)
- Frontal area = 0.418 m^2 (4.50 ft^2)
- Depth = 9.53 cm (3.75 in.)

1.3.2.2.5.2 Tubes.

- 130 tubes total
    - 5 tubes per row
    - 26 rows
- Tube outside diameter = 9.53 mm (0.375 in.)
- Tube inside diameter = 8.81 mm (0.347 in.)
- Exposed tube surface area = 2.229 m^2 (23.99 ft^2).

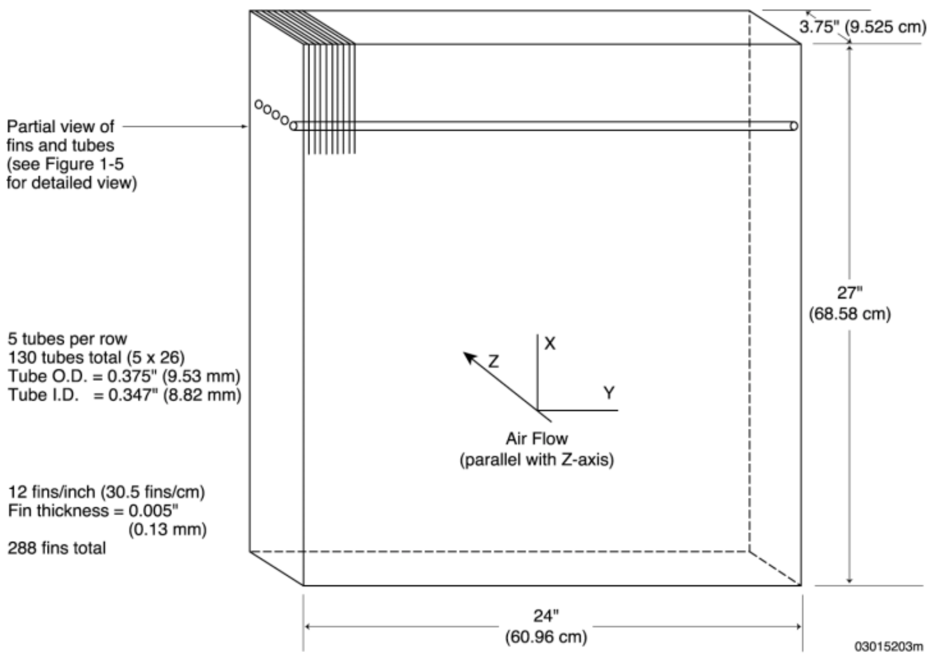


Figure 1-4. Evaporator coil overall dimensions

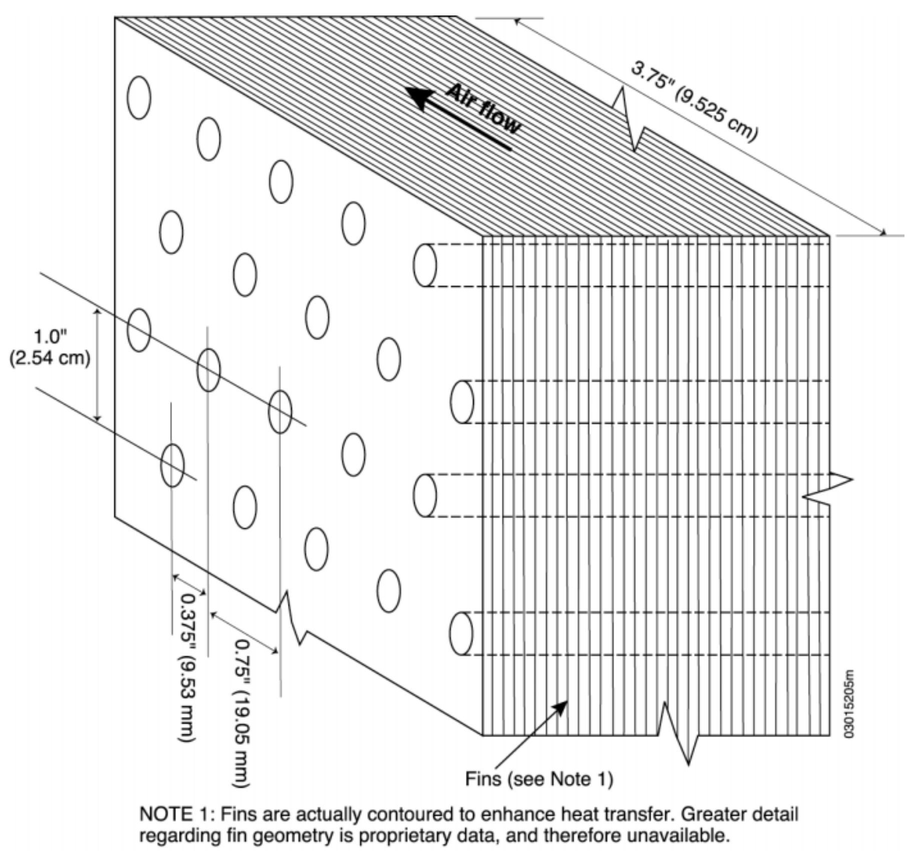


Figure 1-5. Evaporator coil detail, tube and fin geometry

1.3.2.2.5.3 Fins.

- 12 fins per inch
- Fin thickness = 0.13 mm (0.005 in.)
- 288 fins total
- Exposed fin surface area = 32.085 m^2 (345.36 ft^2).

1.3.2.2.6 Fans.

1.3.2.2.6.1 Indoor Air Distribution Fan.

- Indoor fan power = 230 W
- Airflow rate = 0.425 m^3/s = 1529 m^3/h = 900 CFM
- Total combined fan and motor efficiency = 0.5
- Total fan pressure = 271 Pa = 1.09 in. wg (water gauge)
- Supply air temperature rise from fan heat = 0.44°C = 0.8°F
- Air distribution efficiency = 100% (adiabatic ducts).

For further discussion of these inputs, see Appendix D.

The draw-through indoor air distribution fan (LeRoy 1998) cycles on and off with the compressor. For calculating additional heating of the distribution air related to waste heat from the indoor distribution fan, assume that the distribution fan motor is mounted in the distribution air stream so that **100% of the heat from fan energy use goes to the distribution (supply) air**.

1.3.2.2.6.2 Outdoor Condenser Fan.

- Outdoor fan power = 108 W.

The draw-through outdoor condenser fan cycles on and off with the compressor.

1.3.3 Additional Dry Coil Test Cases

This section describes sequential revisions to the base case required to model additional dry-coil cases. The dry-coil cases have no latent load in the zone. In many instances the base case for a given case is not Case E100; appropriate base cases for a given dry-coil case are:

Case	Basis for That Case
E110	E100
E120	E110
E130	E100
E140	E130

1.3.3.1 Case E110: Reduced Outdoor Dry Bulb Temperature

Case E110 is **exactly as Case E100 except** the applicable weather data file is:

HVBT294.TMY or HVBT294.TM2.

1.3.3.2 Case E120: Increased Thermostat Set Point

Case E120 is **exactly as Case E110 except** the thermostat control strategy is:

Heat = off.
Cool = on if zone air temperature >26.7°C (80.0°F); otherwise cool = off.

All other features of the thermostat remain as before.

1.3.3.3 Case E130: Low Part Load Ratio

Case E130 is **exactly as Case E100 except** the internal heat gains are:

Sensible internal gains = 270 W (922 Btu/h), continuously
Latent internal gains = 0 W (0 Btu/h), continuously.

Sensible gains remain as 100% convective.

These internally generated sources of heat are not related to the operation of the mechanical cooling system or its air distribution fan.

1.3.3.4 Case E140: Reduced Outdoor Dry Bulb Temperature at Low Part Load Ratio

Case E140 is **exactly as Case E130 except** the weather applicable weather data file is:

HVBT294.TMY or HVBT294.TM2.

1.3.4 Humid Zone Test Cases

In this section, we describe the sequential revisions required to model humid zone cases. The humid zone cases have latent load in the zone, and therefore have moisture removed by the evaporator coil. All condensed moisture is assumed to leave the system through a condensate drain. The appropriate base cases for a given case are:

Case	Basis for That Case
E150	E110
E160	E150
E165	E160
E170	E150
E180	E170
E185	E180
E190	E180
E195	E190
E200	E150

1.3.4.1 Case E150: Latent Load at High Sensible Heat Ratio

Case E150 is **exactly as Case E110 except** the internal heat gains are:

Sensible internal gains = 5400 W (18430 Btu/h), continuously
Latent internal gains = 1100 W (3754 Btu/h), continuously.

Sensible gains remain as 100% convective.

Zone sensible and latent internal gains are assumed to be distributed evenly throughout the zone air. These internally generated sources of heat are not related to operation of the mechanical cooling system or its air distribution fan.

1.3.4.2 Case E160: Increased Thermostat Set Point at High Sensible Heat Ratio

Case E160 is **exactly as Case E150 except** the thermostat control strategy is:

Heat = off.
Cool = on if zone air temperature >26.7°C (80.0°F); otherwise cool = off.

All other features of the thermostat remain as before.

1.3.4.3 Case E165: Variation of Thermostat Set Point and Outdoor Dry Bulb Temperature at High Sensible Heat Ratio

Case E165 is **exactly as Case E160 except** the thermostat control strategy and weather data are changed as noted below.

- Weather data
  - HVBT406.TMY or HVBT406.TM2
- Thermostat control strategy
  - Heat = off.
  - Cool = on if zone air temperature >23.3°C (74.0°F); otherwise cool = off.

All other features of the thermostat remain as before.

1.3.4.4 Case E170: Reduced Sensible Load

Case E170 is **exactly as Case E150 except** the internal heat gains are:

Sensible internal gains = 2100 W (7166 Btu/h), continuously
Latent internal gains = 1100 W (3754 Btu/h), continuously.

Sensible gains remain as 100% convective.

Zone sensible and latent internal gains are assumed to be distributed evenly throughout the zone air. These internally generated sources of heat are not related to operation of the mechanical cooling system or its air distribution fan.

1.3.4.5 Case E180: Increased Latent Load

Case E180 is **exactly as Case E170 except** the internal heat gains are:

Sensible internal gains = 2100 W (7166 Btu/h), continuously
Latent internal gains = 4400 W (15018 Btu/h), continuously.

Sensible gains remain as 100% convective.

Zone sensible and latent internal gains are assumed to be distributed evenly throughout the zone air. These internally generated sources of heat are not related to operation of the mechanical cooling system or its air distribution fan.

1.3.4.6 Case E185: Increased Outdoor Dry Bulb Temperature at Low Sensible Heat Ratio

Case E185 is **exactly as Case E180 except** the weather applicable weather data file is:

HVBT461.TMY or HVBT461.TM2.

1.3.4.7 Case E190: Low Part Load Ratio at Low Sensible Heat Ratio

Case E190 is **exactly as Case E180 except** the internal heat gains are:

Sensible internal gains = 270 W (922 Btu/h), continuously
Latent internal gains = 550 W (1877 Btu/h), continuously.

Sensible gains remain as 100% convective.

Zone sensible and latent internal gains are assumed to be distributed evenly throughout the zone air. These internally generated sources of heat are not related to operation of the mechanical cooling system or its air distribution fan.

1.3.4.8 Case E195: Increased Outdoor Dry Bulb Temperature at Low Sensible Heat Ratio and Low Part Load Ratio

Case E195 is **exactly as Case E190 except** the weather applicable weather data file is:

HVBT461.TMY or HVBT461.TM2.

1.3.4.9 Case E200: Full Load Test at ARI Conditions

This case compares simulated performance of mechanical equipment to the manufacturer's listed performance at full load and at ARI-specified operating conditions. Case E200 is **exactly as Case E150 except** for the changes noted below.

- Weather data
  - HVBT350.TMY or HVBT350.TM2.
- Internal heat gains
  - Sensible internal gains = 6120 W (20890 Btu/h), continuously
  - Latent internal gains = 1817 W (6200 Btu/h), continuously.

  Sensible gains remain as 100% convective.

  Zone sensible and latent internal gains are assumed to be distributed evenly throughout the zone air. These internally generated sources of heat are not related to operation of the mechanical cooling system or its air distribution fan.

- Thermostat control strategy
  - Heat = off.
  - Cool = on if zone air temperature > 26.7°C (80.0°F); otherwise cool = off.

  All other features of the thermostat remain as before.

Appendix A
Weather Data Format Description

A.1 Typical Meteorological Year (TMY) Format

For convenience we have reprinted the following discussion from the documentation for DOE2.1A *Reference Manual*, (p. VIII-31), and tables (see Table A-1) from *Typical Meteorological Year User's Manual* (National Climatic Center 1981). The reprint of tables from this manual includes some additional notes from our experience with TMY data. If this summary is insufficient for your weather processing needs, the complete documentation on TMY weather data can be obtained from the National Climatic Center (NCC; Federal Bldg., Asheville, NC 28801-2733, telephone 704-271-4800).

Solar radiation and surface meteorological data recorded on an hourly[1] basis are maintained at the NCC. These data cover recording periods from January 1953 through December 1975 for 26 data rehabilitation stations, although the recording periods for some stations may differ. The data are available in blocked (compressed) form on magnetic tape (SOLMET) for the entire recording period for the station of interest.

Contractors who wish to use a database for simulation or system studies for a particular geographic area require a database that is more tractable than these, and also one that is representative of the area. Sandia National Laboratories has used statistical techniques to develop a method for producing a typical meteorological year for each of the 26 rehabilitation stations. This section describes the use of these magnetic tapes.

The TMY tapes comprise specific calendar months selected from the entire recorded span for a given station as the most representative, or typical, for that station and month. For example, a single January is chosen from the 23 Januarys for which data were recorded from 1953 through 1975 because it is most nearly like the composite of all 23 Januarys. Thus, for a given station, January of 1967 might be selected as the typical meteorological month (TMM) after a statistical comparison with all of the other 22 Januarys. This process is pursued for each of the other calendar months, and the 12 months chosen then constitute the TMY.

Although NCC has rehabilitated the data, some recording gaps do occur in the SOLMET tapes. Moreover, there are data gaps because of the change from 1-hour to 3-hour meteorological data recording in 1965. Consequently, as TMY tapes were being constituted from the SOLMET data, the variables data for barometric pressure, temperature, and wind velocity and direction were scanned on a month-by-month basis, and missing data were replaced by linear interpolation. Missing data in the leading and trailing positions of each monthly segment are replaced with the earliest or latest legitimate observation.

Also, because the TMMs were selected from different calendar years, discontinuities occurred at the month interfaces for the above continuous variables. Hence, after the monthly segments were rearranged in calendar order, the discontinuities at the month interfaces were ameliorated by cubic spline smoothing covering the 6-hourly points on either side of the interface.

[1]Hourly readings for meteorological data are available through 1964; subsequent readings are on a 3-hour basis.

TAPE DECK	Table A-1. Typical Meteorological Year Data Format			
9734				
Tape Field Number[a]	Tape Positions[a]	Element	Tape Configuration	Code Definitions and Remarks
002	001-005	WBAN Station number	01001–98999	Unique number used to identify each station
003	006–015 006–007 008–009 010–011 012–015	Solar time Year Month Day Hour	 00–99 01–12 01–31 0001–2400	 Year of observation, 00–99 = 1900–1999 Month of observation, 01–12 = Jan.–Dec. Day of month End of the hour of observation in solar time (hours and *minutes*)
004	016–019	Local Standard Time	0000–2359	Local Standard Time in hours and minutes corresponding to end of solar hour indicated in field 003. For some weather data stations add 30 minutes to the local standard time on the tape.[b]
101	020–023	Extraterrestrial radiation	0000–4957	Amount of solar energy in kJ/m^2 received at top of atmosphere during solar hour ending at time indicated in field 003, based on solar constant = 1377 J/($m^2 \cdot s$). **0000 = nighttime values for extraterrestrial radiation, and 80000 = corresponding nighttime value in field 108. 99999 = nighttime values defined as zero kJ/m^2, for stations noted as “rehabilitated” in the station list**.[c]
102 Use for direct normal solar radiation	024–028 024 025–028	Direct radiation Data code indicator[d] Data[e]	 0–9 0000–4957	Portion of radiant energy in kJ/m^2 received at the pyrheliometer directly from the sun during solar hour ending at time indicated in field 003. **99999 = nighttime values defined as zero kJ/m^2**.
103	 029 030–033	Diffuse radiation Data code indicator[d] Data[e]	 0–9 0000–4957	Amount of radiant energy in kJ/m^2 received at the instrument indirectly from reflection, scattering, etc., during the solar hour ending at the time indicated in field 003. Note: *Diffuse data not available*.
104	034–038 034 035–038	Net radiation Data code indicator[d] Data[e]	 0–9 2000–8000	Difference between the incoming and outgoing radiant energy in kJ/m^2 during the solar hour ending at the time indicated in field 003. A constant of 5000 has been added to all net radiation data. Note: *Net radiation data not available*.
105	039–043 039 040–043	Global radiation on a tilted surface Data code indicator[d] Data[e]	 0–9 0000–4957	Total of direct and diffuse radiant energy in kJ/m^2 received on a tilted surface (tilt angle indicated in station - period of record list) during solar hour ending at the time indicated in field 003. Note: *Data not available*.
	044–058	Global radiation on a horizontal surface		Total of direct and diffuse radiant energy in kJ/m^2 received on a horizontal surface by a pyranometer during solar hour ending at the time indicated in field 003.

TAPE DECK 9734	Table A-1. Typical Meteorological Year Data Format			
Tape Field Number[a]	Tape Positions[a]	Element	Tape Configuration	Code Definitions and Remarks
106	044–048 044 045–048	Observed data Data code indicator[d] Data[e]	 0–9 0000–4957	Observed value. Note: *These data are not corrected. Recommend use of data in field 108*.
107	049–053 049 050–053	Engineering corrected data Data code indicator[d] Data[e]	 0–9 0000–4957	*Note: Recommend use of data in field 108*. Observed value corrected for known scale changes, station moves, recorder and sensor calibration changes, etc.
108 Use for total horizontal solar radiation	054–058 054 055–058	Standard year Corrected data Data code indicator[d] Data[e]	 0–9 000–4957	Observed value adjusted to Standard Year Model. This model yields expected sky irradiance received on a horizontal surface at the elevation of the station. The value includes the effects of clouds. Note: *All nighttime values coded as 80000 except stations noted as rehabilitated in the station list; for those stations, nighttime values are coded 99999.*[c]
109, 110	059–068 059–064 060–063 065–068	Additional radiation measurements Data code indicators[d] Data[e] Data[e]	 0–9	Supplemental fields A and B for additional radiation measurements: type of measurement specified in station-period of record list.
111	069–070	Minutes of sunshine	00–60	For Local Standard Hour most closely matching solar hour. Note: *Data available only for when observations were made.*
201	071–072	Time of TD 1440 Observations	00–23	Local Standard Hour of TD 1440 Meteorological Observation that comes closest to midpoint of the solar hour for which solar data are recorded.
202	073–076	Ceiling height	0000–3000 7777 8888	Ceiling height in dekameters (dam = m $\times$ 10^1); ceiling is defined as opaque sky cover of 0.6 or greater. 0000–3000 = 0 to 30,000 meters 7777 = unlimited; clear 8888 = unknown height of cirroform ceiling

TAPE DECK 9734	Table A-1. Typical Meteorological Year Data Format			
Tape Field Number[a]	Tape Positions[a]	Element	Tape Configuration	Code Definitions and Remarks
203	077–081	Sky condition		
	077	Indicator	0	Identifies observation after June 1, 1951.
	078–081	Sky condition	0000–8888	Coded by layer in ascending order; four layers are described; if fewer than four layers are present the remaining positions are coded 0. The code for each layer is: 0 = Clear or less than 0.1 cover 1 = Thin scattered (0.1–0.5 cover) 2 = Opaque scattered (0.1–0.5 cover) 3 = Thin broken (0.6–0.9 cover) 4 = Opaque broken (0.6–0.9 cover) 5 = Thin overcast (1.0 cover) 6 = Opaque overcast (1.0 cover) 7 = Obscuration 8 = Partial obscuration
204	082–085	Visibility	0000–1600 8888	Prevailing horizontal visibility in hectometers (hm = m $\times$ 10^2). 0000–1600 = 0 to 160 kilometers 8888 = unlimited
205	086–093	Weather		
	086	Occurrence of thunderstorm, tornado, or squall	0–4	0 = None 1 = Thunderstorm—lightning and thunder. Wind gusts less than 50 knots, and hail, if any, less than ¾ inch diameter. 2 = Heavy or severe thunderstorm—frequent intense lightning and thunder. Wind gusts 50 knots or greater and hail, if any, ¾ inch or greater diameter. 3 = Report of tornado or waterspout. 4 = Squall (sudden increase of wind speed by at least 16 knots, reaching 22 knots or more and lasting for at least one minute).
	087	Occurrence of rain, rain showers, or freezing rain	0–8	0 = None 1 = Light rain 2 = Moderate rain 3 = Heavy rain 4 = Light rain showers 5 = Moderate rain showers 6 = Heavy rain showers 7 = Light freezing rain 8 = Moderate or heavy freezing rain

TAPE DECK	Table A-1. Typical Meteorological Year Data Format			
9734				
Tape Field Number[a]	Tape Positions[a]	Element	Tape Configuration	Code Definitions and Remarks
205 (cont'd)	088	Occurrence of drizzle, freezing drizzle	0–6	0 = None 1 = Light drizzle 2 = Moderate drizzle 3 = Heavy drizzle 4 = Light freezing drizzle 5 = Moderate freezing drizzle 6 = Heavy freezing drizzle
	089	Occurrence of snow, snow pellets, or ice crystals	0–8	0 = None 1 = Light snow 2 = Moderate snow 3 = Heavy snow 4 = Light snow pellets 5 = Moderate snow pellets 6 = Heavy snow pellets 7 = Light ice crystals 8 = Moderate ice crystals Beginning April 1963, intensities of ice crystals were discontinued. All occurrences since this date are recorded as an 8.
	090	Occurrence of snow showers or snow grains	0–6	0 = None 1 = Light snow showers 2 = Moderate snow showers 3 = Heavy snow showers 4 = Light snow grains 5 = Moderate snow grains 6 = Heavy snow grains Beginning April 1963, intensities of snow grains were discontinued. All occurrences since this date are recorded as a 5.

TAPE DECK 9734	Table A-1. Typical Meteorological Year Data Format			
Tape Field Number[a]	Tape Positions[a]	Element	Tape Configuration	Code Definitions and Remarks
205 (Cont'd)	091	Occurrence of sleet (ice pellets), sleet showers, or hail	0–8	0 = None 1 = Light sleet or sleet showers (ice pellets) 2 = Moderate sleet or sleet showers (ice pellets) 3 = Heavy sleet or sleet showers (ice pellets) 4 = Light hail 5 = Moderate hail 6 = Heavy hail 7 = Light small hail 8 = Moderate or heavy small hail Prior to April 1970, ice pellets were coded as sleet. Beginning April 1970, sleet and small hail were redefined as ice pellets and are coded as a 1, 2, or 3 in this position. Beginning September 1956, intensities of hail were no longer reported and all occurrences were recorded as a 5.
	092	Occurrence of fog, blowing dust, or blowing sand	0–5	0 = None 1 = Fog 2 = Ice fog 3 = Ground fog 4 = Blowing dust 5 = Blowing sand These values recorded only when visibility less than 7 miles.
	093	Occurrence of smoke, haze, dust, blowing snow, or blowing spray	0–6	0 = None 1 = Smoke 2 = Haze 3 = Smoke and haze 4 = Dust 5 = Blowing snow 6 = Blowing spray These values recorded only when visibility less than 7 miles.

TAPE DECK	Table A-1. Typical Meteorological Year Data Format			
9734				
Tape Field Number[a]	Tape Positions[a]	Element	Tape Configuration	Code Definitions and Remarks
206	094–103	Pressure		
	094–098	Sea level pressure	08000–10999	Pressure, reduced to sea level, in kilopascals (kPa) and hundredths.
	099–103	Station pressure	08000–10999	Pressure at station level in kilopascals (kPa) and hundredths. 08000–10999 = 80 to 109.99 kPa
207	104–111	Temperature		
	104–107	Dry bulb	-700 to 0600	°C and tenths
	108–111	Dew point	-700 to 0600	-700 to 0600 = -70.0 to +60.0°C
208	112–118	Wind		
	112–114	Wind direction	000–360	Degrees
	115–118	Wind speed	0000–1500	m/s and tenths; 0000 with 000 direction indicates calm. 000–1500 = 0 to 150.0 m/s
209	119–122	Clouds		
	119–120	Total sky cover	00–10	Amount of celestial dome in tenths covered by clouds or obscuring phenomena. Opaque means clouds or obscuration through which the sky or higher cloud layers cannot be seen.
	121–122	Total opaque sky cover	00–10	
210	123	Snow cover Indicator	0–1	0 indicates no snow or trace of snow. 1 indicates more than a trace of snow on the ground.
211	124–132	Blank		

[a]Tape positions are the precise column locations of data. Tape Field Numbers are ranges representing topical groups of tape positions.

[b]This remark does NOT apply to the weather data provided with this test procedure.

[c]Weather data used in HVAC BESTEST is based on that from a "rehabilitated" station.

[d]Note for Fields 102-110: Data code indicators are:0=Observed data, 1=Estimated from model using sunshine and cloud data, 2=Estimated from model using cloud data, 3=Estimated from model using sunshine data, 4=Estimated from model using sky condition data, 5=Estimated from linear interpolation, 6=Reserved for future use, 7=Estimated from other model (see individual station notes in SOLMET: Volume 1), 8=Estimated without use of a model, 9=Missing data follows (See model description in SOLMET: Volume 2).

[e]"9s" may represent zeros or missing data or the quantity nine depending on the positions in which they occur. Except for tape positions 001-023 in fields 002-101, elements with a tape configuration of 9's indicate missing or unknown data.

A.2 Typical Meteorological Year 2 (TMY2) Data and Format

The following TMY2 format description is extracted from Section 3 of the TMY2 user manual (Marion and Urban 1995).

For each station, a TMY2 file contains 1 year of hourly solar radiation, illuminance, and meteorological data. The files consist of data for the typical calendar months during 1961–1990 that are concatenated to form the typical meteorological year for each station.

Each hourly record in the file contains values for solar radiation, illuminance, and meteorological elements. A two-character source and uncertainty flag is attached to each data value to indicate whether the data value was measured, modeled, or missing, and to provide an estimate of the uncertainty of the data value.

Users should be aware that the format of the TMY2 data files is different from the format used for the NSRDB and the original TMY data files.

File Convention

File naming convention uses the Weather Bureau Army Navy (WBAN) number as the file prefix, with the characters TM2 as the file extension. For example, 13876.TM2 is the TMY2 file name for Birmingham, Alabama. The TMY2 files contain computer readable ASCII characters and have a file size of 1.26 MB.

File Header

The first record of each file is the file header that describes the station. The file header contains the WBAN number, city, state, time zone, latitude, longitude, and elevation. The field positions and definitions of these header elements are given in Table A-2, along with sample FORTRAN and C formats for reading the header. A sample of a file header and data for January 1 is shown in Figure A-1.

Hourly Records

Following the file header, 8,760 hourly data records provide 1 year of solar radiation, illuminance, and meteorological data, along with their source and uncertainty flags. Table A-3 provides field positions, element definitions, and sample FORTRAN and C formats for reading the hourly records.

Each hourly record begins with the year (field positions 2-3) from which the typical month was chosen, followed by the month, day, and hour information in field positions 4-9. *The times are in local standard time (previous TMYs based on SOLMET/ERSATZ data are in solar time).*

```
 14944 SIOUX_FALLS            SD  -6 N 43 34 W  96 44   435
 85010101000000000000?00000?00000?00000?00000?00000?00000?010A710A7-150A7-211A7060A70975A7360A7052A70161A700945A70999099999004E7050F8000A700E7
 85010102000000000000?00000?00000?00000?00000?00000?00000?010A710A7-144A7-206A7060A70975A7350A7077A70161A700914A70999099999004E7050F8000A700E7
 85010103000000000000?00000?00000?00000?00000?00000?00000?010A710A7-144A7-200A7063A70975A7340A7062A70161A700732A70999099999004E7050F8000A700E7
 85010104000000000000?00000?00000?00000?00000?00000?00000?010A710A7-150A7-206A7063A70976A7330A7072A70161A700640A70999099999004E7050F8000A700E7
 85010105000000000000?00000?00000?00000?00000?00000?00000?010A710A7-156A7-217A7060A70976A7330A7067A70161A700640A70999099999003E7050F8000A700E7
 85010106000000000000?00000?00000?00000?00000?00000?00000?010A710A7-167A7-222A7062A70976A7340A7067A70161A700640A70999099999003E7050F8000A700E7
 85010107000000000000?00000?00000?00000?00000?00000?00000?004A704A7-183A7-233A7065A70977A7300A7052A70193A777777A70999999999003E7050F8000A700E7
 85010108000000000000?00000?00000?00000?00000?00000?00000?002A702A7-194A7-244A7065A70978A7310A7036A70193A777777A70999999999003E7050F8000A700E7
 85010109010212970037G50173G40024G50038I50071I40033I50043I604A700A7-200A7-256A7062A70978A7330A7046A70193A777777A70999999999003E7050F8000A700E7
 85010110028714150157G50560G40043G50159I50444I40069I50079I600A700A7-189A7-256A7056A70979A7310A7067A70193A777777A70999999999003E7050F8000A700E7
 85010111043614150276G40714G40056G50286I40642I40088I50111I500A700A7-172A7-250A7051A70979A7310A7062A70161A777777A70999999999003E7050F8000A700E7
 85010112053014150357G40782G40064G50374I40735I40098I50131I500A700A7-167A7-244A7051A70978A7300A7062A70161A777777A70999999999003E7050F8000A700E7
 85010113056214150387G40806G40067G50407I40767I40101I50139I500A700A7-156A7-244A7047A70978A7320A7067A70193A777777A70999999999003E7050F8000A700E7
 85010114053014150359G40788G40064G50377I40742I40098I50131I500A700A7-144A7-239A7045A70978A7310A7062A70193A777777A70999999999003E7050F8000A700E7
 85010115043614150277G40716G40056G50289I40645I40088I50111I500A700A7-139A7-239A7043A70978A7330A7052A70193A777777A70999999999003E7050F8000A700E7
 85010116028614150157G50564G40043G50162I50450I40069I50080I600A700A7-139A7-233A7045A70978A7300A7052A70161A777777A70999999999003E7050F8000A700E7
 85010117010412730038G50209G40021G50038I50104I40030I50038I600A700A7-150A7-233A7049A70978A7290A7041A70241A777777A70999999999003E7050F8000A700E7
 85010118000000000000?00000?00000?00000?00000?00000?00000?000A700A7-167A7-233A7057A70978A7000A7000A70241A777777A70999999999003E7050F8000A700E7
 85010119000000000000?00000?00000?00000?00000?00000?00000?000A700A7-172A7-233A7059A70978A7000A7000A70241A777777A70999999999003E7050F8000A700E7
 85010120000000000000?00000?00000?00000?00000?00000?00000?000A700A7-178A7-233A7062A70978A7000A7000A70241A777777A70999999999003E7050F8000A700E7
 85010121000000000000?00000?00000?00000?00000?00000?00000?000A700A7-183A7-239A7062A70978A7260A7015A70241A777777A70999999999003E7050F8000A700E7
 85010122000000000000?00000?00000?00000?00000?00000?00000?000A700A7-183A7-239A7062A70977A7220A7021A70241A777777A70999999999003E7050F8000A700E7
 85010123000000000000?00000?00000?00000?00000?00000?00000?000A700A7-178A7-239A7059A70977A7220A7015A70241A777777A70999999999003E7050F8000A700E7
 85010124000000000000?00000?00000?00000?00000?00000?00000?000A700A7-178A7-239A7059A70977A7240A7010A70241A777777A70999999999003E7050F8000A700E7

                                                                                                   1         1         1         1         1
         1         2         3         4         5         6         7         8         9         0         1         2         3         4
1234567890123456789012345678901234567890123456789012345678901234567890123456789012345678901234567890123456789012345678901234567890123456789012
                                                   (for field position identification only)
```

Figure A-1. Sample file header and data in the TMY2 format for January 1

Table A-2. Header Elements in the TMY2 Format (For First Record of Each File)

Field Position	Element	Definition
002 - 006	WBAN Number	Station's Weather Bureau Army Navy number (see Table 2-1 of Marion and Urban [1995])
008 - 029	City	City where the station is located (maximum of 22 characters)
031 - 032	State	State where the station is located (abbreviated to two letters)
034 - 036	Time Zone	Time zone is the number of hours by which the local standard time is ahead of or behind Universal Time. For example, Mountain Standard Time is designated -7 because it is 7 hours behind Universal Time.
038 - 044 038 040 - 041 043 - 044	Latitude	Latitude of the station N = North of equator Degrees Minutes
046 - 053 046 048 - 050 052 - 053	Longitude	Longitude of the station W = West, E = East Degrees Minutes
056 - 059	Elevation	Elevation of station in meters above sea level

FORTRAN Sample Format:
`( 1X,A5,1X,A22,1X,A2,1X,I3,1X,A1,1X,I2,1X,I2,1X,A1,1X,I3,1X,I2,2X,I4 )`
C Sample Format:
`( %s %s %s %d %s %d %d %s %d %d %d )`		

Table A-3. Data Elements in the TMY2 Format (For All Except the First Record)

Field Position	Element	Values	Definition
002 - 009 002 - 003 004 - 005 006 - 007 008 - 009	Local Standard Time Year Month Day Hour	 61 - 90 1 - 12 1 - 31 1 - 24	 Year, 1961-1990 Month Day of month Hour of day in local standard time
010 - 013	Extraterrestrial Horizontal Radiation	0 - 1415	Amount of solar radiation in Wh/m^2 received on a horizontal surface at the top of the atmosphere during the 60 minutes preceding the hour indicated
014 - 017	Extraterrestrial Direct Normal Radiation	0 - 1415	Amount of solar radiation in Wh/m^2 received on a surface normal to the sun at the top of the atmosphere during the 60 minutes preceding the hour indicated
018 - 023 018 - 021 022 023	Global Horizontal Radiation Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 1200 A - H, ? 0 - 9	Total amount of direct and diffuse solar radiation in Wh/m^2 received on a horizontal surface during the 60 minutes preceding the hour indicated
024 - 029 024 - 027 028 029	Direct Normal Radiation Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 1100 A - H, ? 0 - 9	Amount of solar radiation in Wh/m^2 received within a 5.7° field of view centered on the sun, during the 60 minutes preceding the hour indicated

Table A-3. Data Elements in the TMY2 Format (Continued)

Field Position	Element	Values	Definition
030 - 035 030 - 033 034 035	Diffuse Horizontal Radiation Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 700 A - H, ? 0 - 9	Amount of solar radiation in Wh/m^2 received from the sky (excluding the solar disk) on a horizontal surface during the 60 minutes preceding the hour indicated
036 - 041 036 - 039 040 041	Global Horiz. Illuminance Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 1,300 I, ? 0 - 9	Average total amount of direct and diffuse illuminance in hundreds of lux received on a horizontal surface during the 60 minutes preceding the hour indicated. 0 to 1,300 = 0 to 130,000 lux
042 - 047 042 - 045 046 047	Direct Normal Illuminance Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 1,100 I, ? 0 - 9	Average amount of direct normal illuminance in hundreds of lux received within a 5.7 degree field of view centered on the sun during the 60 minutes preceding the hour indicated. 0 to 1,100 = 0 to 110,000 lux
048 - 053 048 - 051 052 053	Diffuse Horiz. Illuminance Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 800 I, ? 0 - 9	Average amount of illuminance in hundreds of lux received from the sky (excluding the solar disk) on a horizontal surface during the 60 minutes preceding the hour indicated. 0 to 800 = 0 to 80,000 lux
054 - 059 054 - 057 058 059	Zenith Luminance Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 7,000 I, ? 0 - 9	Average amount of luminance at the sky's zenith in tens of Cd/m^2 during the 60 minutes preceding the hour indicated. 0 to 7,000 = 0 to 70,000 Cd/m^2
060 - 063 060 - 061 062 063	Total Sky Cover Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 10 A - F, ? 0 - 9	Amount of sky dome in tenths covered by clouds or obscuring phenomena at the hour indicated
064 - 067 064 - 065 066 067	Opaque Sky Cover Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 10 A - F 0 - 9	Amount of sky dome in tenths covered by clouds or obscuring phenomena that prevent observing the sky or higher cloud layers at the hour indicated
068 - 073 068 - 071 072 073	Dry Bulb Temperature Data Value Flag for Data Source Flag for Data Uncertainty	 -500 to 500 A - F 0 - 9	Dry bulb temperature in tenths of °C at the hour indicated. -500 to 500 = -50.0 to 50.0 degrees C
074 - 079 074 - 077 078 079	Dew Point Temperature Data Value Flag for Data Source Flag for Data Uncertainty	 -600 to 300 A - F 0 - 9	Dew point temperature in tenths of °C at the hour indicated. -600 to 300 = -60.0 to 30.0 °C
080 - 084 080 - 082 083 084	Relative Humidity Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 100 A - F 0 - 9	Relative humidity in percent at the hour indicated

Table A-3. Data Elements in the TMY2 Format (Continued)

Field Position	Element	Values	Definition
085 - 090 085 - 088 089 090	Atmospheric Pressure Data Value Flag for Data Source Flag for Data Uncertainty	 700 - 1100 A - F 0 - 9	Atmospheric pressure at station in millibars at the hour indicated
091 - 095 091 - 093 094 095	Wind Direction Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 360 A - F 0 - 9	Wind direction in degrees at the hour indicated. (N = 0 or 360, E = 90, S = 180, W = 270). For calm winds, wind direction equals zero.
096 - 100 096 - 98 99 100	Wind Speed Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 400 A - F 0 - 9	Wind speed in tenths of meters per second at the hour indicated. 0 to 400 = 0 to 40.0 m/s
101 - 106 101 - 104 105 106	Visibility Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 1609 A - F, ? 0 - 9	Horizontal visibility in tenths of kilometers at the hour indicated. 7777 = unlimited visibility 0 to 1609 = 0.0 to 160.9 km 9999 = missing data
107 - 113 107 - 111 112 113	Ceiling Height Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 30450 A - F, ? 0 - 9	Ceiling height in meters at the hour indicated. 77777 = unlimited ceiling height 88888 = cirroform 99999 = missing data
114 - 123	Present Weather	See Appendix B of Marion and Urban (1995)	Present weather conditions denoted by a 10-digit number. See Appendix B of Marion and Urban (1995) for key to present weather elements.
124 - 128 124 - 126 127 128	Precipitable Water Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 100 A - F 0 - 9	Precipitable water in millimeters at the hour indicated
129 - 133 129 - 131 132 133	Aerosol Optical Depth Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 240 A - F 0 - 9	Broadband aerosol optical depth (broad-band turbidity) in thousandths on the day indicated. 0 to 240 = 0.0 to 0.240
134 - 138 134 - 136 137 138	Snow Depth Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 150 A - F, ? 0 - 9	Snow depth in centimeters on the day indicated. 999 = missing data
139 - 142 139 - 140 141 142	Days Since Last Snowfall Data Value Flag for Data Source Flag for Data Uncertainty	 0 - 88 A - F, ? 0 - 9	Number of days since last snowfall 88 = 88 or greater days 99 = missing data

FORTRAN Sample Format:

```
(1X,4I2,2I4,7(I4,A1,I1),2(I2,A1,I1),2(I4,A1,I1),1(I3,A1,I1),
 1(I4,A1,I1),2(I3,A1,I1),1(I4,A1,I1),1(I5,A1,I1),10I1,3(I3,A1,I1),
 1(I2,A1,I1))
```

C Sample Format:

```
(%2d%2d%2d%2d%4d%4d%4d%1s%1d%4d%1s%1d%4d%1s%1d%4d%1s%1d%4d%1s%1d%4d%1s
 %1d%4d%1s%1d%2d%1s%1d%2d%1s%1d%4d%1s%1d%4d%1s%1d%3d%1s%1d%4d%1s%1d%3d
 %1s%1d%3d%1s%1d%4d%1s%1d%5ld%1s%1d%1d%1d%1d%1d%1d%1d%1d%1d%1d%3d%1s
 %1d%3d%1s%1d%3d%1s%1d%2d%1s%1d)
```

Note: For ceiling height data, integer variable should accept data values as large as 99999.

For solar radiation and illuminance elements, the data values represent the energy received during the 60 minutes *preceding the hour indicated*. For meteorological elements (with a few exceptions), observations or measurements were made *at the hour indicated*. A few of the meteorological elements had observations, measurements, or estimates made at daily, instead of hourly, intervals. Consequently, the data values for broadband aerosol optical depth, snow depth, and days since last snowfall represent the values available for the day indicated.

Missing Data

Data for some stations, times, and elements are missing. The causes for missing data include such things as equipment problems, some stations not operating at night, and a National Oceanic and Atmospheric Administration (NOAA) cost-saving effort from 1965 to 1981 that digitized data for only every third hour.

Although both the National Solar Radiation Database (NSRDB) and the TMY2 data sets used methods to fill data where possible, some elements, because of their discontinuous nature, did not lend themselves to interpolation or other data-filling methods. Consequently, data in the TMY2 data files may be missing for horizontal visibility, ceiling height, and present weather for up to 2 consecutive hours for Class A stations and for up to 47 hours for Class B stations. For Colorado Springs, Colorado, snow depth and days since last snowfall may also be missing. No data are missing for more than 47 hours, except for snow depth and days since last snowfall for Colorado Springs, Colorado. As indicated in Table A-3, missing data values are represented by 9's and the appropriate source and uncertainty flags.

Source and Uncertainty Flags

With the exception of extraterrestrial horizontal and extraterrestrial direct radiation, the two field positions immediately following the data value provide source and uncertainty flags both to indicate whether the data were measured, modeled, or missing, and to provide an estimate of the uncertainty of the data. Source and uncertainty flags for extraterrestrial horizontal and extraterrestrial direct radiation are not provided because these elements were calculated using equations considered to give exact values.

For the most part, the source and uncertainty flags in the TMY2 data files are the same as the ones in NSRDB, from which the TMY2 files were derived. However, differences do exist for data that were missing in the NSRDB, but then filled while developing the TMY2 data sets. Uncertainty values apply to the data with respect to when the data were measured, and not as to how "typical" a particular hour is for a future month and day. More information on data filling and the assignment of source and uncertainty flags is found in Appendix A of Marion and Urban (1995).

Tables A-4 through A-7 define the source and uncertainty flags for the solar radiation, illuminance, and meteorological elements.

Table A-4. Solar Radiation and Illuminance Source Flags

Flag	Definition
A	Post-1976 measured solar radiation data as received from NCDC or other sources
B	Same as "A" except the global horizontal data underwent a calibration correction
C	Pre-1976 measured global horizontal data (direct and diffuse were not measured before 1976), adjusted from solar to local time, usually with a calibration correction
D	Data derived from the other two elements of solar radiation using the relationship, global = diffuse + direct $\times$ cosine (zenith)
E	Modeled solar radiation data using inputs of *observed* sky cover (cloud amount) and aerosol optical depths derived from direct normal data collected at the same location
F	Modeled solar radiation data using *interpolated* sky cover and aerosol optical depths derived from direct normal data collected at the same location
G	Modeled solar radiation data using *observed* sky cover and aerosol optical depths estimated from geographical relationships
H	Modeled solar radiation data using *interpolated* sky cover and estimated aerosol optical depths
I	Modeled illuminance or luminance data derived from measured or modeled solar radiation data
?	Source does not fit any of the above categories. Used for nighttime values, calculated extraterrestrial values, and missing data

Table A-5. Solar Radiation and Illuminance Uncertainty Flags

Flag	Uncertainty Range (%)
1	Not used
2	2 - 4
3	4 - 6
4	6 - 9
5	9 - 13
6	13 - 18
7	18 - 25
8	25 - 35
9	35 - 50
0	Not applicable

Table A-6. Meteorological Source Flags

Flag	Definition
A	Data as received from NCDC, converted to SI units
B	Linearly interpolated
C	Non-linearly interpolated to fill data gaps from 6 to 47 hours in length
D	Not used
E	Modeled or estimated, except: precipitable water, calculated from radiosonde data; dew point temperature calculated from dry bulb temperature and relative humidity; and relative humidity calculated from dry bulb temperature and dew point temperature
F	Precipitable water, calculated from surface vapor pressure; aerosol optical depth, estimated from geographic correlation
?	Source does not fit any of the above. Used mostly for missing data

Table A-7. Meteorological Uncertainty Flags

Flag	Definition
1 - 6	Not used
7	Uncertainty consistent with NWS practices and the instrument or observation used to obtain the data
8	Greater uncertainty than 7 because values were interpolated or estimated
9	Greater uncertainty than 8 or unknown
0	Not definable

APPENDIX B
COP Degradation Factor (CDF) as a Function of Part Load Ratio (PLR)

Per the equipment manufacturer (D Cawley),

CDF = 1 - Cd(1-PLR), 0 <= PLR <= 1
CDF = 1, PLR => 1 (1)

where Cd is assumed constant for a given unit.

Cd can be determined from listed performance data using:

SEER = EERb (1 - 0.5Cd) which solves for Cd as,

Cd = (1 - SEER/EERb)/0.5 (2)

where EERb = (Adjusted Net Total Capacity)/(Cooling Energy Consumption) at EDB = 80 F, EWB = 67 F, ODB =82 F; listed SEER = 12.90
Adjusted Net Total Capacity (Qnetcap,adj) is used to account for fan heat assumed for gross capacity calculation being different from actual fan heat.

Note: The procedures for independently calculating Cd and EERb from measured data are given in ANSI/ARI 210/240-89.

Calculations to obtain EERb

Extrapolate performance based on manufacturer data (Qnetcap,list), for EDB = 80, EWB = 67.

	Total Capacities				
ODB (°F)	Qnetcap,list (kBtu/h)	Qnetcap,adj (kBtu/h)	Qcomp (kW)	Qfans (kW)	EER
90	27.6	27.9	1.79	0.34	13.13
85	28.4	28.7	1.71	0.34	14.03
					extrap
82					14.57

Eqn (2) implies Cd =	0.229

So for some points along the CDF f(PLR) linear curve using Equation (1)

PLR	CDF
1.0	1.000
0.9	0.977
0.8	0.954
0.7	0.931
0.6	0.908
0.5	0.885
0.4	0.862
0.3	0.840
0.2	0.817
0.1	0.794
0.0	0.771

cdfvplr.xls; May 14, 2001

Appendix C
Cooling Coil Bypass Factor

We have provided the calculation techniques in this appendix for illustrative purposes. Some models may have slight variations in the calculation including the use of enthalpy ratios rather than dry bulb temperature ratios in Equation 1 (below), or different specific heat assumptions for leaving air conditions in Equation 3 (below), among others.

Cooling coil BF can be thought of as the fraction 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 surface temperature (ADP). BF at ARI rating conditions is *approximately*:

$0.049 \le \text{BF} \le 0.080.$

The uncertainty surrounding this value is illustrated in the two examples for calculating BF from given manufacturer data that are included in the rest of this appendix, as well as from separate calculation results by Technische Universität Dresden (TUD). The uncertainty can be traced to the calculated ADP (56.2°F) being different from the ADP listed by the manufacturer (56.8°F). Because we have been unable to acquire the manufacturer's specific method for determining ADP, we have not been able to determine which ADP number is better. However, the manufacturer has indicated that performance data are only good to within 5% of real equipment performance (Houser 1994). So we can hypothesize that the listed versus calculated ADP disagreements could be a consequence of the development of separate correlation equations for each performance parameter within the range of experimental uncertainty. Based on simulation sensitivity tests with DOE-2.1E, the above range of BF inputs causes total electricity consumption to vary by ±1%.

Calculations based on the listed performance data indicate that BF varies as a function of EDB, EWB, and ODB. Incorporate this aspect of equipment performance into your model if your software allows it, using a consistent method for developing all points of the BF variation map. (Note that sensitivity tests for cases E100–E200 using DOE-2.1E indicate that assuming a constant value of BF—versus allowing BF to vary as a function of EWB and ODB—adds an additional ±1% uncertainty to the total energy consumption results for Case E185, and less for the other cases.)

The equipment manufacturer recommends modeling the BF as independent of (<u>not</u> varying with) the PLR (Cawley 1997). This is because the airflow rate over the cooling coil is assumed constant when the compressor is operating (fan cycles on/off with compressor).

Calculation of Coil Bypass Factor

Nomenclature

ADP	Apparatus dew point (°F)
BF	Bypass factor (dimensionless)
cp_a	Specific heat of dry air (Btu/lb°F)
cp_w	Specific heat of water vapor (Btu/lb°F)
h_1	Enthalpy of air entering cooling coil (Btu/lb dry air)
h_2	Enthalpy of air leaving cooling coil (Btu/lb dry air)
q_s	Gross sensible capacity (Btu/h)
q_T	Gross total capacity (Btu/h)
Q	Indoor fan airflow rate (ft^3/min)
T_{db1}	Entering dry bulb temperature (°F)
T_{db2}	Leaving dry bulb temperature (°F)
T_{wb1}	Entering wet bulb temperature (°F)
w	Humidity ratio (lb water vapor/lb dry air)
ρ_r	Density of standard dry air at fan rating conditions (0.075 lb/ft^3)

Known Information

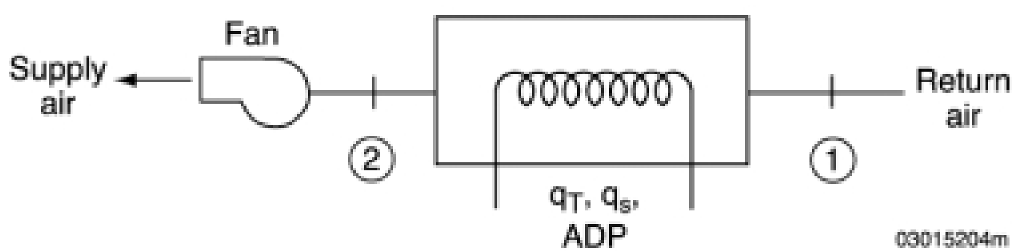


Figure C-1. System schematic

Air-Conditioning and Refrigeration Institute Conditions:

- T_{db1} = 80°F
- T_{wb1} = 67°F.

From Table 1-6d at ARI conditions:

- Q = 900 ft^3/min
- q_s = 21700 Btu/h (gross sensible capacity)
- q_T = 27900 Btu/h (gross total capacity)
- ADP = 56.8°F.

Governing Equations

$$BF = (T_{db2} - ADP)/(T_{db1} - ADP) \quad \text{(McQuiston and Parker 1994; pp. 80–81)} \qquad \text{(Eq. 1)}$$

The following equations and related properties are commonly used approximations for working with volumetric flow rates (Howell et al. pp. 3.4–3.5).

$$q_T = \rho_r\, Q\, (60\ \text{min/h})\, (h_1 - h_2) \qquad \text{(Eq. 2)}$$

$$q_s = \rho_r\, Q\, (60\ \text{min/h})\, (cp_a + cp_w(w))\, (T_{db1} - T_{db2}) \qquad \text{(Eq. 3)}$$

$\rho_r = 0.075\ \text{lb/ft}^3$

$cp_a = 0.24\ \text{Btu/lb°F}$

$cp_w = 0.45\ \text{Btu/lb°F}$

$w \approx 0.01$ lb water vapor/lb dry air.

So for these English units, Equations (2) and (3) become:

$$q_T = 4.5\, Q\, (h_1 - h_2) \qquad \text{(Eq. 2a)}$$

$$q_s = 1.10\, Q\, (T_{db1} - T_{db2}) \qquad \text{(Eq. 3a)}$$

Solution Technique Using ADP Calculated by Extending the Condition Line to the Saturation Curve.

To find ADP, extend the condition line of the system through the saturation curve on the psychrometric chart. The condition line is the line going through coil entering conditions with slope determined by sensible heat ratio for the given operating conditions (McQuiston and Parker 1994; pp. 77–81). This example is illustrated on the psychrometric chart in Figure C-2. To draw the condition line, State 2 must be determined; State 1 is ARI conditions (T_{db1} = 80.0°F, T_{wb1} = 67°F.). Defining State 2 requires two independent properties that can be identified from Equations (2) and (3).

Solve for h_2 using Equation (2) with q_T = 27,900 Btu/h and Q = 900 ft^3/min. From ideal gas equations commonly used for psychrometrics (ASHRAE 1997), at ARI conditions h_1 = 31.45 Btu/lb dry air. These values applied to Equation (2) give:

h_2 = 24.55 Btu/lb dry air.

Solving for T_{db2} using Equation (3) with T_{db1} = 80°F, q_s = 21700 Btu/h, and Q = 900 ft^3/min gives:

T_{db2} = 58.1°F.

On the psychrometric chart, drawing a line through these two states and extending it to the saturation curve gives:

ADP = 56.2°F.

Solving Equation (1) using T_{db1} = 80°F, T_{db2} = 58.1°F, and ADP = 56.2°F gives:

BF = 0.080

Solution Technique Using ADP Listed in Performance Data

Solving Equation (1) using T_{db1} = 80°F, T_{db2} = 58.1°F, and ADP = 56.8°F gives:

BF = 0.055

Solution by TUD

The TRNSYS-TUD modeler report indicates that:

BF = 0.049

This solution is based on manufacturer-listed values of ADP. See the TRNSYS-TUD Modeler Report in Part III for more discussion.

Conclusions

The BF for this system at ARI conditions is *approximately* in the range of:

$0.049 \leq BF \leq 0.080$

Some uncertainty is associated with the governing equations and related properties commonly used for calculating leaving air conditions; these equations are approximations. In addition, some uncertainty is associated with using the psychrometric chart to find the ADP (56.2°F) in the first solution. Finally, there may be additional uncertainty related to the methodology for developing ADP. For example, the results of Equation 1 can be slightly different if enthalpy ratios are used in place of dry bulb temperature ratios. Also, documentation of how the manufacturer calculated its listed ADP was unavailable, and the source code for manufacturer software used to develop catalog data is proprietary.

Based on sensitivity tests with DOE-2.1E:

- The above range of BF inputs causes total electricity consumption to vary by ±1%.
- Assuming a constant value of BF versus allowing BF to vary as a function of EWB and ODB adds an additional ±1% uncertainty to the total energy consumption results for Case E185, and less for the other cases.

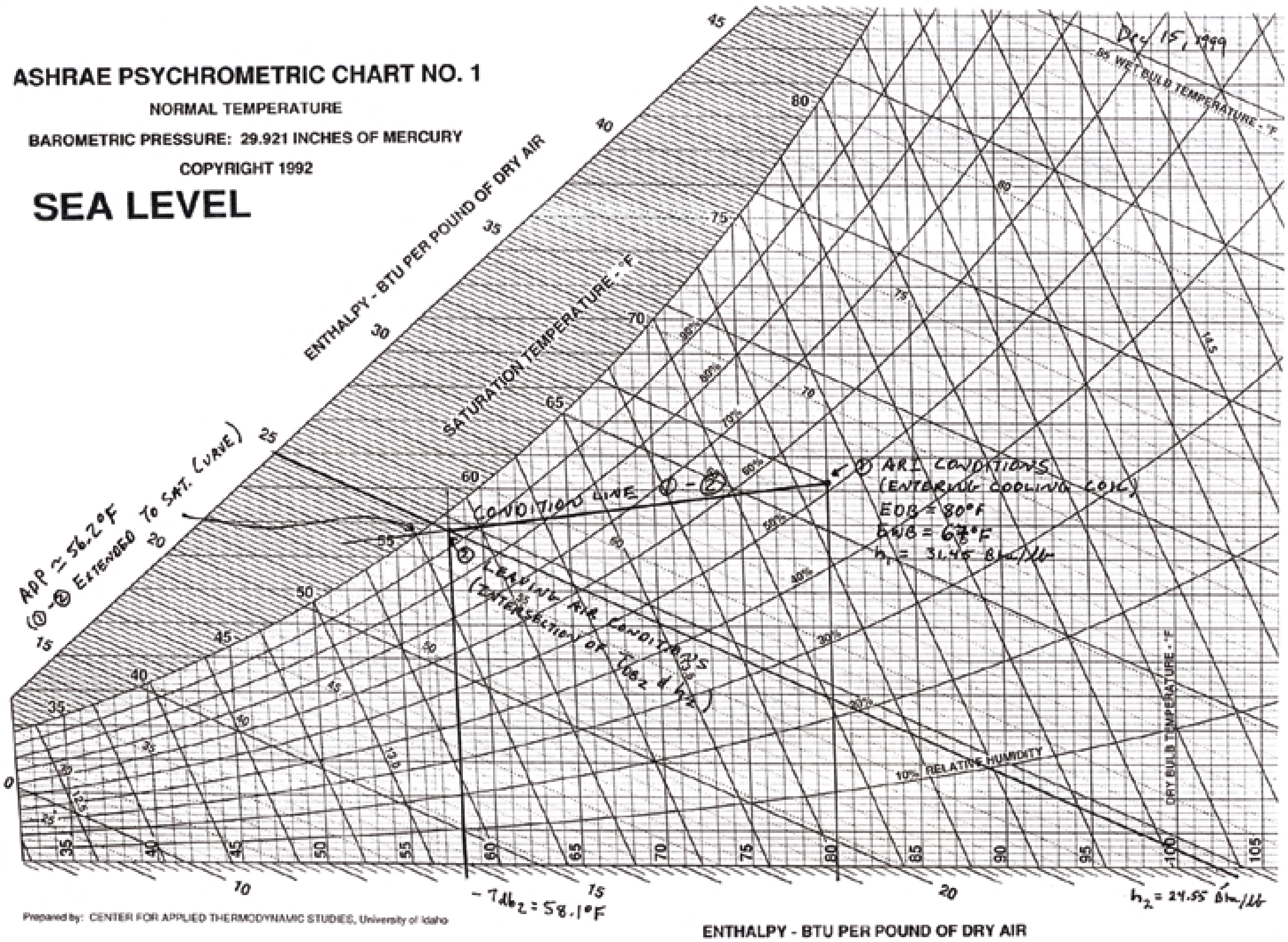


Figure C-2. ADP calculation

Appendix D
Indoor Fan Data Equivalence

Fan performance data for indoor fan power (230 W) and airflow rate (900 CFM = 0.425 m^3/s) are based on dry air at standard fan rating conditions. ASHRAE defines a standard condition as 1 atmosphere (101.325 kPa or 14.696 psi) and 68°F (20°C) with a density of 0.075 lb/ft^3 (1.204 kg/m^3) (Howell, Sauer, and Coad 1998; p. 3.4).

The fan efficiency of 0.5 is based on a discussion with the unitary system manufacturer (Houser 1994).

The **total fan pressure** is based on:

$Eff = Q * \Delta P / W$ (ANSI/AMCA 210-85, ANSI/ASHRAE 51, 1985; pp. 4, 46–48)

where:

- Q ≡ Indoor fan airflow rate (m^3/s)
- ΔP ≡ Total fan pressure (Pa)
- W ≡ Fan electric power input (W)
- Eff ≡ Total fan plus motor and drive efficiency (motor/drive in air stream).

Solving for ΔP:

$$\Delta P = W * Eff / Q$$

$$= 230 \text{ W} * 0.5 / 0.425 \text{ m}^3/\text{s} = \underline{271 \text{ Pa} = \Delta P}.$$

The **supply air temperature rise from fan heat** is based on:

$q_{fan} = \rho * c_p * Q * \Delta T * C$

where:

- q_{fan} ≡ Fan heat (Btu/h or W)
- ρ ≡ Standard air density = 0.075 lb/ft^3 (1.204 kg/m^3)
- c_p ≡ Specific heat of air (Btu/(lb°F) or kJ/(kgK))
- Q ≡ Indoor fan airflow rate (ft^3/min or m^3/s)
- ΔT ≡ Supply air temperature rise from fan heat (°F or °C)
- C ≡ Units conversion constant.

Solving for ΔT:

$\Delta T \quad = q_{fan} / (\rho * c_p * Q * C)$

where:

q_{fan} = 230 W = 785 Btu/h; Q = 900 CFM = 0.425 m^3/s

c_p = 0.24 Btu/lb F for dry air, or

c_p = 0.2445 Btu/lb F when humidity ratio = 0.01 (Howell, Sauer, and Coad 1998; p. 3.5).

Then, ΔT = 785 Btu/h / { 0.075 lb/ft^3 * 900 ft^3/min * 60 min/h * 0.2445 Btu/(lb°F) }

$\underline{\Delta T = 0.793°F\ (0.441\ °C)}$, or

for cp = 0.24 Btu/(lb°F), gives $\underline{\Delta T = 0.808°F\ (0.449°C)}$.

Appendix E
Output Spreadsheet Instructions

HVAC BESTEST Output Form, HVBTOUT.XLS

Instructions:

1. Use specified units
2. Data entry is restricted to columns B through T and rows 25 through 38. The protection option has been employed to help ensure that data are input in the correct cells.
3. February totals are consumptions and loads just for the month of February. Similarly, February means and maxima are those values just for the month of February.
4. Cooling energy consumption, evaporator coil load, zone load, and COP are defined in the Glossary (Appendix H).

We have included a printout of HVBTOUT.XLS on the following page.

HVBTOUT.XLS
Output spreadsheet for HVAC BESTEST

INSTRUCTIONS

1. Use specified units

2. Data entry is restricted to columns B through T and rows 25 through 38. The protection option has been employed to help assure that data is input in the correct cells.

3. February totals are consumption and or loads just for the month of February. Similarly, February means, maxima, and minima are those values just for the month of February.

4. Cooling Energy Consumption, Evaporator Coil Load, Zone Load, and COP (Coefficient of Performance) are defined in the Glossary appendix.

	February Totals										February Mean			February Maximum			February Minimum		
	Cooling Energy Consumption				Evaporator Coil Load			Zone Load											
Cases	Total (kWh)	Compressor (kWh)	Supply Fan (kWh)	Condenser Fan (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)
E100																			
E110																			
E120																			
E130																			
E140																			
E150																			
E160																			
E165																			
E170																			
E180																			
E185																			
E190																			
E195																			
E200																			

Appendix F
Diagnosing the Results Using the Flow Diagrams

F.1 General Description

The E100 series cases (E100 through E200) are steady-state cases that test basic performance map modeling capabilities and utilize comparisons with analytical solutions. Figure F-1 contains a flow diagram that serves as a guide for diagnosing the cause of disagreeing results for these cases. The flow diagram lists the feature(s) being tested, thus indicating potential sources of algorithmic differences.

F.2 Comparing Tested Software Results to Analytical Solution Results and Example Simulation Results

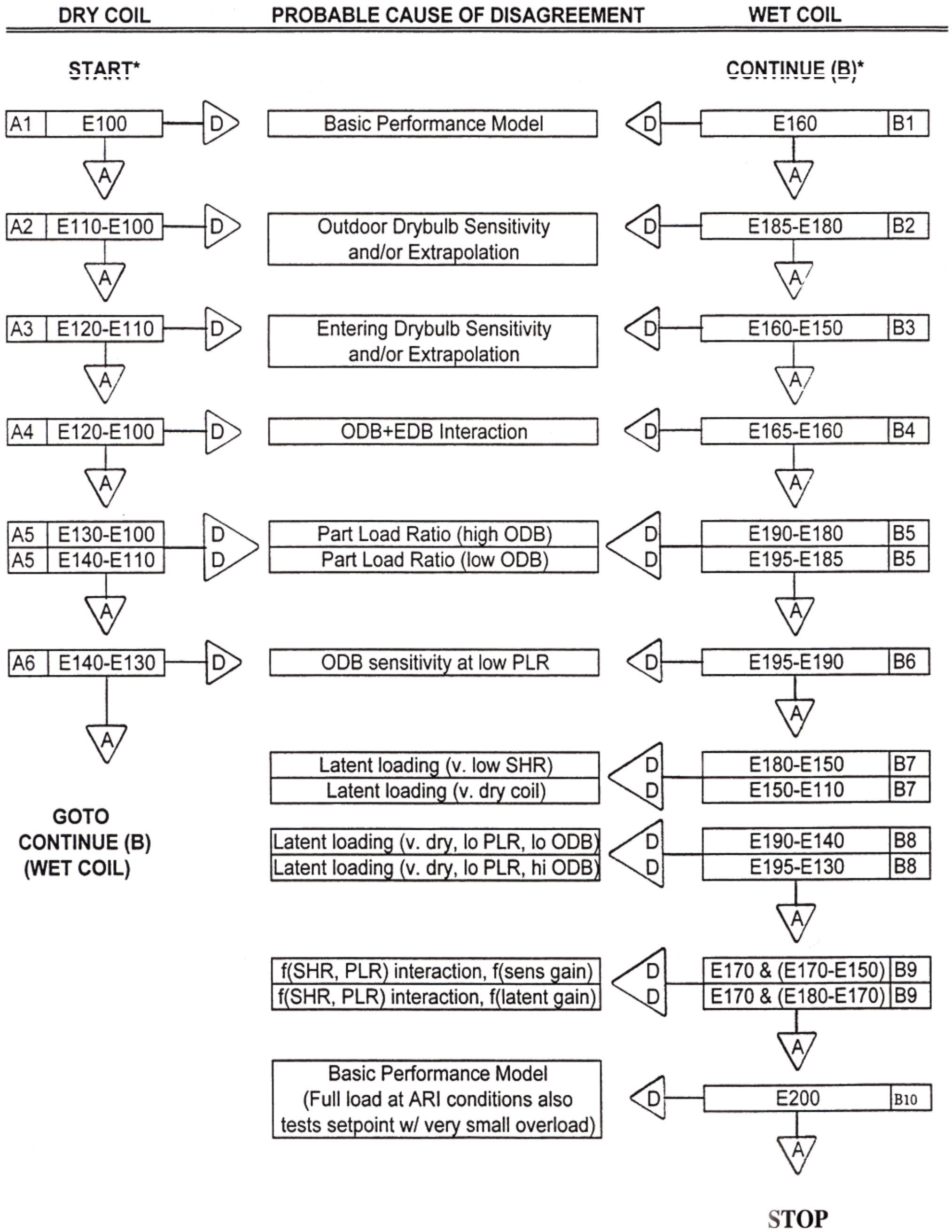
Analytical solution results are presented in Part II and also included in Part IV. Example simulation results are given in Part IV.

As a minimum, the user should compare output with the analytical solution results found in Part II. The user may also choose to compare output with the example simulation results in Part IV, or with other results that were generated using this test procedure. Information about how the analytical solutions and example simulation results were produced is included in Parts II and III, respectively. For convenience to users who wish to plot or tabulate their results along with the analytical solution or example simulation results, or both, an electronic version of the example results has been included with the file RESULTS.XLS on the accompanying electronic media.

No formal criteria exist for when results agree or disagree; determination of the agreement or disagreement of results is left to the user. In making this determination, the user should consider that the analytical solution results represent a “mathematical truth standard” (i.e., a mathematically provable and deterministic set of results based on acceptance of the underlying physical assumptions represented by the case specifications). The authors recognize that although the underlying physical assumptions of the case definitions of the mechanical equipment are consistent with those of typical manufacturer equipment performance data, they are by definition a simplification of reality and may not fully represent real empirical behavior.

In making a determination about the agreement of results, the user should also consider:

- The magnitude of results for individual cases
- The magnitude of difference in results between certain cases (e.g., “Case E110–Case E100”)
- The same direction of sensitivity (positive or negative) for difference in results between certain cases (e.g., “Case E110–Case E100”)
- The degree of disagreement that occurred for other simulation results in Part IV versus the analytical solution results.



ABBREVIATIONS
A = Agree, i.e., agree with analytical solution results for the case itself and the sensitivity case. E.g., to check for agreement regarding Case E130, compare example results for Case E130 and E130–E100 sensitivity.
D = Disagree, i.e., show disagreement with analytical solution results.
NOTES
* It is better to perform/analyze results of these tests in blocks such as E100–E140 and E150–E200.

Figure F-1. E100-E200 series (steady-state analytical verification) diagnostic logic flow diagram

Check the program being tested for agreement with analytical solution results for both the absolute outputs and the sensitivity (or “delta”) outputs. For example, when comparing to the analytical solution results for Case “E110–E100” in Figure F-1, the program results are compared with both the Case E110 results and the Case E110–E100 sensitivity results.

Compare all available output types specified for each case that can be produced by the program being tested. This includes appropriate energy consumption, coil load, zone load, zone temperature, and humidity ratio results if the software being tested is capable of producing that type of output. A disagreement with any one of the output types may be cause for concern.

The E100 series tests provide detailed diagnostic capabilities about basic modeling with performance maps. The E100 series flow diagram (Figure F-1) indicates similar diagnostics for dry-coil and wet-coil (without and with dehumidification) cases. This is really one continuous diagnostic path to be implemented for both dry-coil and wet-coil cases. Performing/analyzing results of the E100 series tests in blocks such as E100–E140 and E150–E200, or E100–E200 all at once is recommended. For the E100 series cases if a disagreement is uncovered for one of the cases, then fix it and rerun all the E100 series cases.

F.3 If Tested Software Results Disagree with Analytical Solution Results

If the tested program shows disagreement with the analytical solution results (using the criteria described above), recheck the inputs against the specified values. Use the diagnostic logic flow diagrams to help isolate the source of the difference. If no input error can be found, look for an error in the software. If an error is found, fix it and rerun the tests. If in the engineering judgment of the user the disagreement is due to a reasonable difference in algorithms between the tested software and the analytical solution results, continue with the next test case.

F.4 Example

A program shows agreement with Case E100, but shows large disagreement with the analytical solution results energy consumption predictions for Case E130. Figure F-1 suggests the potential algorithmic source of the difference is with the algorithm for incorporating part-load operating effects into the energy consumption for a dry coil.

Section 3.4 (Part III) gives examples of how the tests were used to trace and correct specific algorithmic and input errors in the programs used in the field trials.

Appendix G
Abbreviations and Acronyms

ACH	air changes per hour
ADP	apparatus dew point
ANSI	American National Standards Institute
ARI	Air Conditioning and Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
BF	bypass factor
CIBSE	Chartered Institution of Building Services Engineers
CDF	coefficient of performance degradation factor
CFM	cubic feet per minute
COP	coefficient of performance
EDB	entering dry bulb temperature
EER	energy efficiency ratio
EWB	entering wet bulb temperature
HVAC	heating, ventilating, and air-conditioning
I.D.	inside diameter
IDB	indoor dry bulb temperature
k	thermal conductivity (W/mK)
NOAA	National Oceanic & Atmospheric Administration
NSRDB	National Solar Radiation Database
O.D.	outside diameter
ODB	outdoor dry bulb temperature
PLR	part load ratio
R	unit thermal resistance (m^2K/W)
SEER	seasonal energy efficiency ratio
SHR	sensible heat ratio
SI	Système Internationale
U	unit thermal conductance or overall heat transfer coefficient ($W/(m^2K)$)
UA	thermal conductance (W/K)
TUD	Technische Universität Dresden
TMY	typical meteorological year
TMY2	typical meteorological year 2
WBAN	Weather Bureau Army Navy
wg	water gauge

Appendix H
Glossary

Glossary terms used in the definitions of other terms are highlighted with italics.

References for terms listed here that are not specific to this test procedure include: ANSI/ARI 210/240-89 (1989); ASHRAE Handbook of Fundamentals (1997); ASHRAE Psychrometric Chart No. 1 (1992); ASHRAE Terminology of Heating, Ventilation, Air-Conditioning, and Refrigeration (1991); Brandemuehl (1993); Cawley (1997); Lindeburg (1990); McQuiston and Parker (1994); and Van Wylen and Sonntag (1985).

Adjusted net sensible capacity is the *gross sensible capacity* less the actual fan power (230 W).

Adjusted net total capacity is the *gross total capacity* less the actual fan power (230 W).

Apparatus dew point (ADP) is the effective coil surface temperature when there is dehumidification; this is the temperature to which all the supply air would be cooled if 100% of the supply air contacted the coil. 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 ((*gross sensible capacity*)/(*gross total capacity*)).

Bypass factor (BF) can be thought of as 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*).

Coefficient of performance (COP) for a cooling (refrigeration) system is the ratio, using same units, of the *net refrigeration effect* to the *cooling energy consumption*.

Cooling energy consumption is the site electric energy consumption of the mechanical cooling equipment including the compressor, air distribution fan (regardless of whether the compressor is on or off), condenser fan, and related auxiliaries.

COP_{SEER} is a dimensionless seasonal energy efficiency ratio.

COP degradation factor (CDF) is a multiplier (≤ 1) applied to the full load system COP. CDF is a function of *part load ratio*.

Dew point temperature is the temperature of saturated air at a given *humidity ratio* and pressure. As moist air is cooled at constant pressure, the dew point is the temperature at which condensation begins.

Energy efficiency ratio (EER) is the ratio of *net refrigeration effect* (in units of Btu per hour) to *cooling energy consumption* (in units of watts) so that EER is stated in units of (Btu/h)/W.

Entering dry bulb temperature (EDB) is the temperature that a thermometer would measure for air entering the evaporator coil. For a draw-through fan configuration with no heat gains or losses in the ductwork, EDB equals the indoor dry bulb temperature.

Entering wet bulb temperature (EWB) is the temperature that the wet bulb portion of a psychrometer would measure if exposed to air entering the evaporator coil. For a draw-through fan with no heat gains or losses in the ductwork, this would also be the zone air wet bulb temperature. For mixtures of water vapor and dry air at atmospheric temperatures and pressures, the wet bulb temperature is approximately equal to the adiabatic saturation temperature (temperature of the air after undergoing a theoretical adiabatic saturation process). The wet bulb temperature given in psychrometric charts is really the adiabatic saturation temperature.

Evaporator coil loads are the actual *sensible heat* and *latent heat* removed from the distribution air by the evaporator coil. These loads include air distribution fan heat for times when the compressor is operating, and are limited by the system capacity (where system capacity is a function of operating conditions). If the fan operates while the compressor is off, the related fan heat is not a coil load, but rather an internal gain to the zone.

Gross sensible capacity is the rate of *sensible heat* removal by the cooling coil for a given set of operating conditions. This value varies as a function of performance parameters such as EWB, ODB, EDB, and airflow rate.

Gross total capacity is the total rate of both *sensible heat* and *latent heat* removal by the cooling coil for a given set of operating conditions. This value varies as a function of performance parameters such as EWB, ODB, EDB, and airflow rate.

Humidity ratio is the ratio of the mass of water vapor to the mass of dry air in a moist air sample.

Indoor dry bulb temperature (IDB) is the temperature that a thermometer would measure if exposed to indoor air.

Latent heat is the change in enthalpy associated with a change in *humidity ratio*, caused by the addition or removal of moisture.

Net refrigeration effect is the rate of heat removal (sensible + latent) by the evaporator coil, as regulated by the thermostat (i.e., not necessarily the full load capacity), after deducting internal and external heat transfers to air passing over the evaporator coil. For this series of tests, the net refrigeration effect is the *evaporator coil load* less the actual air distribution fan heat for the time when the compressor is operating; at full load this is also the *adjusted net total capacity*. Air distribution fan heat for times when the compressor is not operating (such as in forthcoming additional test cases with a continuously operating indoor fan, or with the economizer operating while the compressor is off) is not deducted for calculating this term. See also *sensible heat* and *latent heat*.

Net sensible capacity is the *gross sensible capacity* less the default rate of fan heat assumed by the manufacturer (329 W); this rate of fan heat is not necessarily the same as for the actual installed fan (see *adjusted net sensible capacity*).

Net total capacity is the *gross total capacity* less the default rate of fan heat assumed by the manufacturer (329 W); this rate of fan heat is not necessarily the same as for the actual installed fan (see *adjusted net total capacity*).

Outdoor dry bulb temperature (ODB) is the temperature that a thermometer would measure if exposed to outdoor air. This is the temperature of air entering the condenser coil.

Part load ratio (PLR) is the ratio of the *net refrigeration effect* to the *adjusted net total capacity* for the cooling coil.

Seasonal energy efficiency ratio (SEER) is the ratio of *net refrigeration effect* in Btu to the *cooling energy consumption* in watt-hours for a refrigerating device over its normal annual usage period, as determined using ANSI/ARI Standard 210/240-89. This parameter is commonly used for simplified estimates of energy consumption based on a given load, and is not generally useful for detailed simulations of mechanical systems.

Sensible heat is the change in enthalpy associated with a change in dry bulb temperature, caused by the addition or removal of heat.

Sensible heat ratio (SHR), also known as sensible heat factor (SHF), is the ratio of sensible heat transfer to total (sensible + latent) heat transfer for a process. See also *sensible heat* and *latent heat*.

Zone cooling loads are *sensible heat* and *latent heat* loads associated with heat and moisture exchange between the building envelope and its surroundings as well as internal heat and moisture gains within the building. These loads do not include internal gains associated with operating the mechanical system (e.g., air distribution fan heat).

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PART II:

Production of Analytical Solutions

2.0 Part II: Production of Analytical Solution Results

2.1 Introduction

In this section we describe how two of the working group participants developed independent analytical solutions, including a third party comparison and subsequent solution revisions. Section 2.4 tabulates the final analytical solution results.

At the March 1998 International Energy Agency (IEA) Solar Heating and Cooling (SHC) Programme Task 22 Meeting in Golden, Colorado, task participants were able to compare their modeling results for cases E100–E200. At that meeting, R. Judkoff and J. Neymark proposed that because of the highly constrained boundary conditions in cases E100–E200, solving those cases analytically should be possible. A set of analytical solutions would represent a "mathematical truth standard" for cases E100–E200; that is, given the underlying physical assumptions in the case definitions, a mathematically provable and deterministic solution exists for each case. In this context, the underlying physical assumptions about the mechanical equipment as defined in cases E100–E200 are representative of the typical manufacturer data normally used by building design practitioners. Many "whole-building" simulation programs are designed to work with this type of data.

It is important to understand the difference between a "mathematical truth standard" and an "absolute truth standard." In the former, we accept the given underlying physical assumptions while recognizing that these assumptions represent a simplification of physical reality. The ultimate or "absolute" validation standard would be a comparison of simulation results with a *perfectly performed* empirical experiment, the inputs for which are *perfectly specified* to the simulationists. In reality, an experiment is performed and the experimental object is specified within some acceptable band of uncertainty. Such experiments are possible but fairly expensive. We recommend the development of a set of empirical validation experiments for future work.

At the March 1998 meeting, two of the participating countries expressed interest in developing analytical solutions, which they subsequently did. The two sets of analytical solution results are mostly well within a < 1% range of disagreement. This remaining disagreement is small enough to identify bugs in software that would not otherwise be apparent from comparing software only to other software. For example, see the Part IV results and compare the range of disagreement among software versus the range of disagreement among analytical solutions. We can see, then, that having analytically solvable cases improves the diagnostic capabilities of the test procedure.

Two organizations developed the analytical solutions: Hochschule Technik + Architektur Luzern (HTAL), and Technische Universität Dresden (TUD). The organizations developed their initial solutions independently, and submitted them to a third party specializing in applied mathematics for review. Comparison of the results indicated some disagreements, which were then resolved by allowing the solvers to review the third party reviewers' comments, and to also review and critique each other's solution techniques. From this process, both solvers were able to resolve most differences in their solutions in a logical non-arbitrary manner. Remaining minor differences in the analytical solutions are due in part to the difficulty of completely describing boundary conditions. In this case, the boundary conditions are a compromise between full reality and some simplification of the real physical system that is analytically solvable. Therefore, the analytical solutions have some element of interpretation of the exact nature of the boundary conditions that causes minor differences in the results. For example, in the modeling of the controller, one group derived an analytical solution for an "ideal" controller, while another group developed a numerical solution for a "realistic" controller. Each solution yields slightly different results, but both are correct in the context of this exercise. This may be less than perfect from a mathematician's viewpoint, but quite acceptable from an engineering perspective. Section 2.2 supplies further details on the comparison process and documentation of the final solution techniques and their remaining differences. The analytical solution techniques are documented in Section 2.3. The final analytical solution results, including summaries of percent disagreement, are tabulated in Section 2.4. The Section 2.4 results are also included on the accompanying electronic media.

2.2 Comparison of Analytical Solution Results and Resolution of Disagreements

2.2.1 Procedure for Developing Analytical Solutions

The objective of developing analytical solutions is to arrive at a reliable set of theoretical results against which building energy simulation software results can be compared. This discussion is intended to document the process of development, comparison, and revision of the analytical solutions. The originally proposed procedure was:

- To initially develop two independent solutions
- To ask a third party to compare results and summarize areas of disagreements
- To direct solvers to modify their solutions or "agree to disagree," or both, about final details (under supervision of the third party, with hopefully only small disagreement remaining).

In this manner, after the initial independent solutions were developed, the solvers would then work together to reach agreement about what they both consider the most correct solution technique.

2.2.2 Development of Analytical Solutions by HTAL and TUD

TUD initially submitted analytical solution results to NREL in June 1998 (Le and Knabe 1998). In April 1999, NREL received documentation of the technique (Le and Knabe 1999a, 1999b). These results were based on the steady-state ideal-control solution technique that is documented in Section 2.3.1. The results were not shared with any of the other participants until they were sent to KST Berlin (M. Behne) and HTAL (G. Zweifel) in October 1999.

HTAL (M. Durig) submitted its results and solution technique documentation to NREL in March 1999. HTAL's initial solution technique is a hybrid problem-specific model analytical solution. The solution applies a steady-state solution technique, somewhat similar to TUD's, to a realistic control model (using 1-s time steps to model ideal control), as documented in HTAL's modeler report (see Section 2.3.2, Subsection 9). Results were submitted for both an adiabatic and near-adiabatic envelope.

NREL's preliminary review of the results (April 1999) indicated variation between the two sets of results by as much as 10%. Additionally, the HTAL results did not indicate any coefficient of performance (COP) sensitivity to variation in part load ratio, and their latent coil loads were greater than their latent zone loads. After this review, HTAL submitted revised results and more complete documentation in May 1999 (Zweifel and Durig 1999).

By the October 1999 experts meeting, a detailed third party review of both solution techniques had not yet begun. At the meeting, HTAL volunteered to provide an applied mathemetician (A. Glass) to identify disagreements between the TUD and HTAL solution techniques. This review, completed during February and March 2000, identified the following items as being handled differently (potentially causing disagreement) in the solutions (Durig 2000a; Glass 2000):

- Non-incorporation of COP f(PLR) performance degradation (CDF) by HTAL gives a 15%difference from TUD's results

- Use of 100,000 Pa by HTAL for Patm in psychrometric equations (TUD used 101,000 Pa)
- Use of c_p for liquid water instead of for water vapor in one of the HTAL equations
- Use of adiabatic envelope by HTAL; test spec gives a near-adiabatic envelope
- Unclear exactly how TUD determined entering wetbulb temperature (EWB), and resulting humidity ratio from EWB, and whether they neglect the difference between zone enthalpy and saturation enthalpy
- TUD humidity ratios and temperatures on the saturation curve seem "very slightly inconsistent"
- Both solvers should use Patm = 101,325 Pa.

In March 2000, HTAL submitted revised solutions including a new "100%-analytical" solution (similar technique to TUD's), as well as a revised version of the original solution technique incorporating changes as noted above.[1] Although the solution details are different, the TUD solution was "unblinded" to the HTAL team around this time. Additionally, all HTAL solutions were revised to use the near-adiabatic envelope except for the original solution technique version of Case E200, which uses the adiabatic envelope. Use of the near-adiabatic envelope caused inaccuracy of the HTAL2 result relative to the 100%-analytical solution (HTAL1) result for that case as explained in Section 2.3.2, Subsection 13. Other improvements to the HTAL1 and HTAL2 solutions included (Durig 2000b):

- Patm = 101,325
- Inclusion of COP degradation factor (CDF f(PLR))
- Calculation of saturation pressure with American Society for Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE) formula
- Correction of evaporation energy calculation
- Correction of enthalpy calculation to use c_p vapor.

These changes by HTAL gave much improved agreement between HTAL and TUD results. In March 2000, NREL distributed HTAL's solution to TUD, thereby unblinding the TUD team and allowing the team members to comment on HTAL's work.

Also during March 2000, NREL distributed the analytical solution results to all of the project participants. The results indicated good agreement (generally within < 1%) except for Case E120. Case E120 had a high disagreement of 8.0% for humidity ratio and 0.8% for consumption. Also at this time, TUD submitted its preliminary comments on results differences (Knabe and Le 2000). TUD's modeler report (see Section 2.3.1, Subsection 4) indicates further differences. A summary of the comments shows the following differences between TUD's and HTAL's methods in March–April 2000:

- TUD's calculation has an additional iteration to increase precision in terms of additional supply fan heat resulting from CDF adjustment at part load. Per the TUD calculation, this has the following effect on equipment run time: a 0.18% increase for E110 and a 0.54% increase for E170. For Case E170 electricity consumption, the level of disagreement between HTAL and TUD is similar to that caused by the E170 run time difference. A similar disagreement for the E170 coil load was also expected; however, considering the variation of performance parameters

[1] In the results of Section 2.4 and Part IV, the 100%-analytical solution results are indicated by "HTAL1," and the original solution technique (applying the realistic controller with a 1-s time step) as "HTAL2."

indicates that compensating differences in the coil load calculations are possible (so that TUD's and HTAL's electricity consumptions can have greater disagreement than the coil loads). For Case E110, the part load ratio is too high for the fan heat f(CDF) calculation difference to have a noticeable impact on the results. However, the greater total space-cooling electricity consumption percentage disagreement for HTAL versus TUD results as PLR decreases—indicated in Section 2.4 for cases E130 and E140, which have the lowest PLRs and also the greatest percentage disagreements—are likely caused by this difference in precision of the calculated run time.

- TUD used Patm = 101,000 Pa; HTAL uses 101,325 Pa
- For calculating humidity ratio (x_{zone}), TUD used the equation:

  $x_{zone} = (h_{sat} - c_{pa}*EDB) / (h_{ig} + c_{pv}*EDB)$

  where:

  c_{pa} and c_{pv} are specific heat of air and water vapor, respectively, and
  EDB ≡ entering drybulb temperature
  h_{ig} ≡ enthalpy of vaporization for water at 0°C and
  h_{sat} ≡ enthalpy of saturated air at zone EWB.

  This assumes the EWB lines and enthalpy lines on the psychrometric chart are parallel; i.e., that $h_{zone} = h_{sat}$, which ignores the secondary term:

  $\Delta h = h_{sat} - h_{zone} = c_{pw}*EWB*(x_{sat}-x_{zone})$

  where h_{zone} and x_{zone} are the enthalpy and humidity ratio of the zone air, respectively, and c_{pw} is the specific heat of liquid water.

  HTAL makes the more exact calculation as shown in Section 2.3.2, Subsection 4.5.

- TUD develops linear interpolation/extrapolation formulae over the full extremes of given "wet coil" (where moisture condenses on the cooling coil) data points, whereas HTAL selects more local intervals that can include "dry coil" (where no moisture condenses on the coil) data points for such calculations. For Case E120, this results in HTAL incorrectly using dry coil total capacity data for linear interpolations/extrapolations.
- The HTAL1 solution documentation does not give information about dependence of steady-state operating point on start values.

In May 2000, NREL submitted a comment to HTAL, based on its modeler report, that initial humidity ratios used in the HTAL2 model did not conform to the test specification's requirement that indoor humidity ratio equals outdoor humidity ratio at the start of the calculations. Later in May 2000, HTAL submitted revised results for the HTAL2 solution using corrected initial zone humidity ratios, except for cases E100 and E200. This had only a small effect on the results.

In July 2000, HTAL submitted corrected results for its HTAL1 and HTAL2 Case E120 solutions using extrapolation of wetcoil data rather than the previous interpolation between wetcoil and drycoil data. The new results showed improved agreement for Case E120 versus the TUD results. In August 2000, HTAL also submitted corrected documentation of its HTAL2 results, indicating that the initial zone humidity ratio of 0.01 kg/kg was applied in Case E200. For HTAL2, the researchers also submitted comparative results for Case E100 using initial zone humidity ratios of 0.009 versus 0.01. For this case, using the initial value of 0.009 was better for HTAL2 (Durig, Glass, and Zweifel 2000).

In August 2000, TUD submitted new analytical solution results incorporating Patm = 101,325 Pa into the calculations. TUD also revised its humidity ratio calculation to match HTAL's, so that the previously excluded difference between saturation enthalpy and zone enthalpy is now included. The change in Patm had a negligible effect on the results. The corrected humidity ratio calculation had a maximum effect on zone humidity ratio results of about 1.7%, but no effect on the other results.

After receiving HTAL's August 2000 modeler report, NREL commented on the possibility that the inability to run Case E200 with near-adiabatic conditions could be caused by insensitivity of sensible capacity to variation of EDB in the HTAL2 model. In September 2000, HTAL revised the HTAL2 model to include sensitivity of sensible capacity to EDB. The new results indicated that this improvement allows the HTAL2 model to run Case E200 with the near-adiabatic envelope, and allows an initial humidity ratio of 0.01 (kg/kg) to be used for Case E100 (see Section 2.3.2, Subsection 11).

Both analytical solutions reasonably neglect the effect of solar gains. Although neither solution report gives a justification for doing this, a heat balance calculation on an exterior surface indicates that the fraction of incident gains actually conducted inward through the near-adiabatic building shell and into the zone may be expressed by the exterior solar absorptance (0.1) multiplied by the ratio of the conductance of the exterior wall (0.01 W/m^2K) to the conductance of the exterior surface coefficient (29.3 W/m^2K). (Also see *ASHRAE Handbook of Fundamentals* [2001], pp. 30.36–30.37.) For average daily global horizontal solar flux of 174 W/m^2 incident on a horizontal surface area equivalent to the sum of the areas of the roof, south wall, and one east- or west-side wall (86 m^2), then the resulting inward-conducted portion of absorbed solar radiation is only 0.5 W on average throughout the simulation period. Relative to the given internal gains, this represents only 0.2% of sensible load for Case E130 (in the most significant case) and only 0.01% of sensible load for Case E200 (in the least significant case).

2.2.3 Conclusions about the Development of Analytical Solutions

The remaining differences in the analytical solution results are generally <1%, and are much smaller than the remaining differences among the simulations results (see Part IV). Therefore, further work to obtain more precise agreement between the analytical solutions was not pursued.

The remaining differences among the solution techniques are:

- More precise accounting of fan heat as a function of part load ratio by TUD than by HTAL1 and HTAL2; this is likely the most significant difference between the HTAL1 and TUD solution techniques, causing the consumption disagreement to approach 1% at low part-load ratios.
- More localized interpolation intervals used by HTAL1 and HTAL2 than used by TUD
- HTAL2 uses a realistic controller with a very short (1-s) timestep; TUD and HTAL1 use an ideal controller.

The resulting process conformed to the initial goal of obtaining high-quality solutions by beginning with independent blind solutions, then allowing a third party and eventually, the solvers themselves to comment on the work, fix errors, and move toward agreement in a logical and non-arbitrary process. The end result is two initially independent solutions (TUD and HTAL2) that each revealed differences (sometimes significant) in the other, so that both are improved and are now in high agreement. Additionally, a third solution (HTAL1) was developed semi-independently after the TUD solution techniques were received. The HTAL1 solution allows the effect on results (minimal) of a realistic controller with a 1-s time step versus a theoretical ideal controller to be isolated.

Based on the solution comparisons and revisions documented above, these analytical results constitute a set of reliable theoretical solutions—based on commonly accepted assumptions about mechanical equipment performance behavior used in the case descriptions—within their range of disagreement (generally < 1%) as documented in the tables of Section 2.4.

2.3 Documentation of the Solutions

Documentation of the final solution techniques for TUD and HTAL are presented in Sections 2.3.1 and 2.3.2, respectively.

NOTE: These sections are reproduced here largely as received from TUD and HTAL and have received minimal editing at NREL.

2.3.1 Analytical Solutions by Technische Universität Dresden (TUD)

Dresden University of Technology

Faculty for Mechanical Engineering

Department of Technical Building Services

HVAC BESTEST Modeler Report

Analytical Solution

By

H.-T. Le

G. Knabe

September 2000

Use of Terms

$P_{envelope,lat}$	latent heat flow through building envelope
$P_{envelope,sen}$	sensible heat flow through building envelope
$P_{gain,lat}$	internal latent gain
$P_{gain,sen}$	internal sensible gain
$P_{zone,lat}$	total latent load of the room
$P_{zone,sen}$	total sensible load of the room
$P_{zone,tot}$	total load of the room
$P_{Adj.\_Net\_Lat}$	Adj. Net Latent Capacity
$P_{Adj.\_Net\_Sen}$	Adj. Net Sensible Capacity
$P_{Adj.\_Net\_Tot}$	Adj. Net Total Capacity
$P_{ECL\_Capacity}$	Capacity of Evaporator Coil
P_{ODfan}	heat of condenser fan
P_{IDfan}	heat of indoor fan (supply fan)
$P_{Compressor}$	compressor power
$Q_{zone,lat}$	latent roomload (zone load) for a time period
$Q_{zone,sen}$	sensible roomload (zone load) for a time period
$Q_{zone,tot}$	total roomload (zone load) for a time period
$Q_{Adj.\_Net\_Lat}$	Adj. latent Coil Load for a time period
$Q_{Adj.\_Net\_Sen}$	Adj. sensible Coil Load for a time period
$Q_{Adj.\_Net\_Tot}$	Adj. total Coil Load for a time period
Q_{ECL_Lat}	latent Evaporator Coil Load for a time period
Q_{ECL_Sen}	sensible Evaporator Coil Load for a time period
Q_{ECL_Tot}	total Evaporator Coil Load for a time period
Q_{ODfan}	power of condenser fan for a time period
Q_{IDfan}	power of indoor fan for a time period
$Q_{Compressor}$	power of compressor for a time period
COP_{full}	Coefficient of Performance at full load operation
COP_{part}	Coefficient of Performance at part load operation
EWB	Entering WetBulb temperature
EDB	Entering DryBulb temperature
EHR	Entering Humidity Ratio
ZHR	Zone Humidity Ratio
DPT	Dew Point Temperature
PLR	Part Load Ratio
SHR	Sensible Heat Ratio
$x_{sat.}$	humidity ratio of saturated air at zone EWB
$h_{sat.}$	enthalpy of saturated air at zone EWB
x_{zone}	Zone Humidity Ratio (ZHR)
$p_{sat.}$	pressure of saturated air at zone EWB
p_{tot}	pressure of the zone air
h_{zone}	enthalpy of zone air
t_B	operating time for a time period

1.0 Introduction

The goal of HVAC BESTEST is testing mechanical system simulation models and diagnosing sources of predictive disagreements. As known, a typical simulation program contains hundreds of variables and parameters. The number of possible cases that can be simulated by varying each of these parameters cannot be fully tested. On the other hand, HVAC BESTEST consists of a series of stationary tests using a carefully specified mechanical system applied to a highly simplified near-adiabatic building envelope. So, it is possible to develop an analytical solution with the given underlying physical assumptions in the case definitions. The method is mathematically provable and is a deterministic solution for each case. Results of the analytical solution should be useful for improvement and debugging of the building simulation program.

This is a report on an analytical solution done by TUD. It shows how to solve the problem HVAC BESTEST without computer simulation with stationary behavior of building and the given underlying physical assumptions and applying an ideal controller

2.0 Development of an Algorithm for An Analytical Solution

2.1 Zone Load

Energy flux through building envelope:

The heat flow through the building envelope at stationary conditions that has to be considered with the analytical solution computes with equation (2-1) below.

$$P_{envelope,sen} = \sum_{i=1}^{n} \Delta\vartheta_i * A_i * U_i \quad (W) \qquad (2\text{-}1)$$

where:

$\Delta\vartheta_i$: temperature difference between zone and ambient air (K)

A_i : surface of the i^{th} wall (m^2)

U_i : heat transfer coefficient of the i^{th} wall $\left(\frac{W}{m^2K}\right)$

n : number of walls bounded the zone

Note: No latent energy through building envelope, therefore:

$P_{envelope,lat} = 0$ W

Total zone load:

$$P_{zone,tot} = P_{zone,sen} + P_{zone,lat} \qquad (2\text{-}2)$$

where:

$$P_{zone,lat} = P_{gain,lat} + P_{envelope,lat}$$
$$P_{zone,sen} = P_{gain,sen} + P_{envelope,sen}$$

2.2 Split System and Building

In this section, the system behavior is analyzed as well as the behavior of the system and the building in conjunction with the two-point controller of the compressor.

2.2.1 Behavior of Split System

In the test description data points of performance map at full load operation are given. In this map, wet coil conditions are indicated where the total capacities are greater than the sensible capacities. Otherwise dry coil conditions occur. These data points are only valid for wet coil conditions, so the data points of performance map for dry coil conditions cannot be used. Therefore, an analysis of system behavior for dry coil conditions is necessary. To analyze the system behavior, the adjusted net capacity given in Table 1-6e of HVAC BESTEST [22] is utilized because the supply fan is a part of the mechanical system. The following figures show the behavior of this split system for the wet coil conditions.

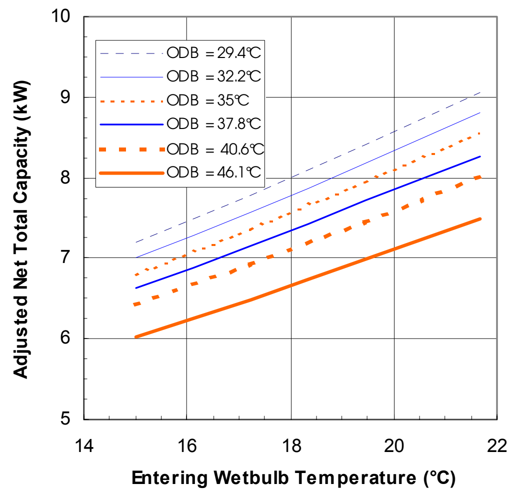

Figure 2-1. Adjusted net total capacity depending on EWB and ODB

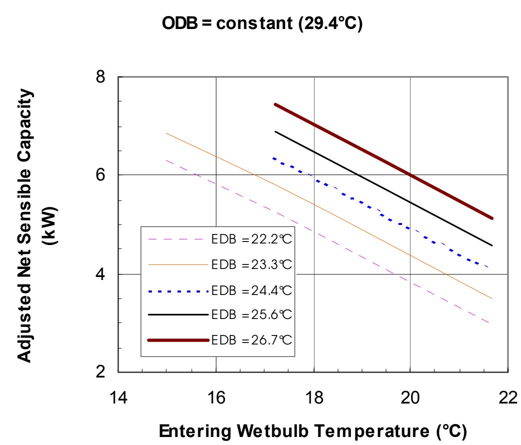

Figure 2-2. Adjusted net sensible capacity depending on EWB and EDB

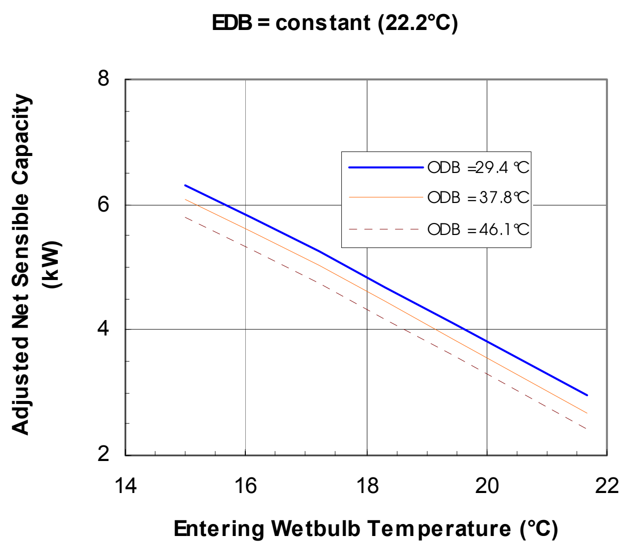

Figure 2-3. Adjusted net sensible capacity depending on EWB and ODB

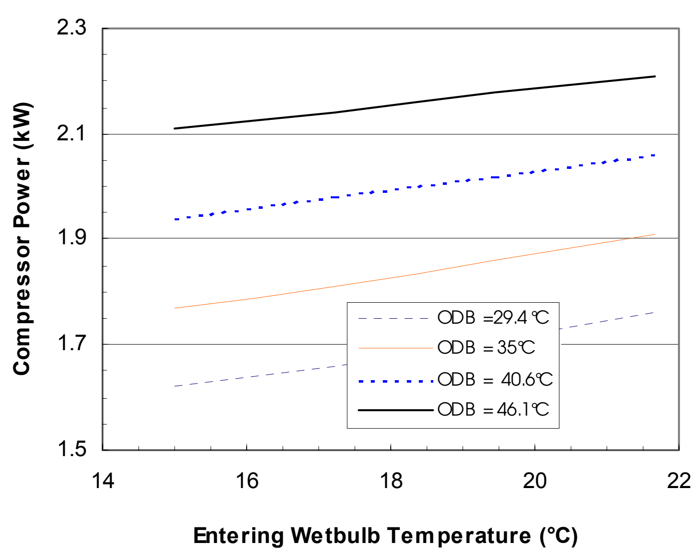

Figure 2-4. Compressor power depending on EWB and ODB

Figures 2-1 to 2-4 show that: In the field of wet coil conditions where the EWB is greater than the intersection point (EWB_1; see Figure 2-5) the adjusted net total capacity is proportional to the entering wet bulb temperature, whereas the adjusted net sensible capacity is inversely proportional. The coil capacities (sensible and total) do not change with varied EWB ($EWB < EWB_1$) by dry coil conditions. Figure 2-5 illustrates this behavior where the "intersection point" indicates the initial dry coil condition (boundary of the wet coil condition).

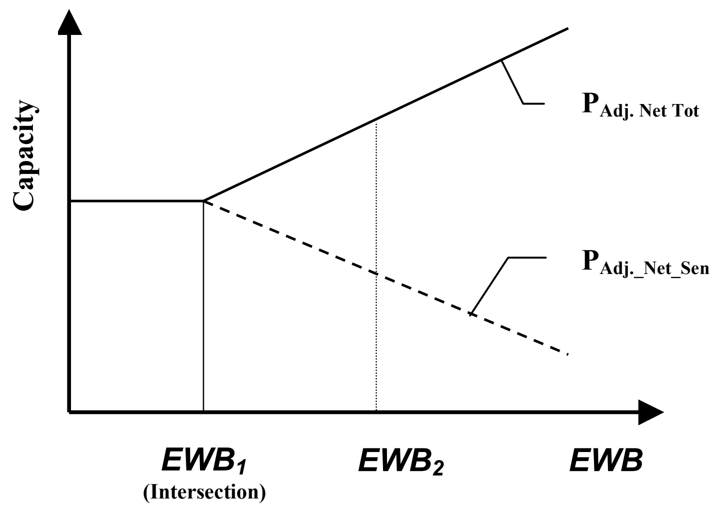

Figure 2-5: System behavior

These figures also show that the adjusted net total capacity and the compressor power for wet coil conditions behave linear to the EWB and the ODB, whereas the adjusted net sensible capacity is a linear function of EWB, EDB, and ODB. According to the manufacturer, the data points of the performance map contain some uncertainties in the experimental measurements [9]; therefore, it is recommended to apply over full extremes valid data points for the approximation/extrapolation. That could eliminate this uncertainty. So, a fitting of custom curve of the performance map using multi-linear approximations for the wet coil performance can be done. The equations of approximation have the following formulas:

$$P_{Adj.\_Net\_Tot} = (\vartheta_{ODB} * A_1 + A_2) * \vartheta_{EWB} + (\vartheta_{ODB} * A_3 + A_4) \tag{2-3}$$

$$P_{Adj.\_Net\_Sen} = (\vartheta_{ODB} * B_1 + \vartheta_{EDB} * B_2 + B_3) * \vartheta_{EWB} + (\vartheta_{ODB} * B_4 + \vartheta_{EDB} * B_5 + B_6) \tag{2-4}$$

$$P_{Compressor} = (\vartheta_{ODB} * C_1 + C_2) * \vartheta_{EWB} + (\vartheta_{ODB} * C_3 + C_4) \tag{2-5}$$

These equations (2-3), (2-4), and (2-5) are the characteristic curves of the evaporator coil identifiable from the performance map at full load operation.

The point between the dry and wet coil conditions is defined as the intersection point (EWB_1) that can be solved from equations (2-3) and (2-4):

$$\vartheta_{EWB,intersection} = \frac{(\vartheta_{ODB} \cdot B_4 + \vartheta_{EDB} \cdot B_5 + B_6) - (\vartheta_{ODB} \cdot A_3 + A_4)}{(\vartheta_{ODB} \cdot A_1 + A_2) - (\vartheta_{ODB} \cdot B_1 + \vartheta_{EDB} \cdot B_2 + B_3)} \tag{2-6}$$

where:

$A_1 =$ -0.00374128077
$A_2 =$ 0.390148024
$A_3 =$ -0.0135886344
$A_4 =$ 3.3894767

$B_1 =$ -0.000629934065
$B_2 =$ -0.00267022306
$B_3 =$ -0.424403961
$B_4 =$ -0.0199848128
$B_5 =$ 0.535732506
$B_6 =$ 2.57416166

$C_1 =$ -0.00040500166
$C_2 =$ 0.0345047542
$C_3 =$ 0.0357013738
$C_4 =$ 0.219759426

For determination of the coil capacities and the compressor power by the points where EWB is less than EWB_1, the EWB is replaced by EWB_1. That means if EWB < EWB_1 then the coil capacities and the compressor power are a function of EWB_1.

Note: The replacement is only for calculation of the coil capacities and the compressor power, because they are constant in the field of dry coil conditions. But for computation of zone humidity ratio from EWB and set point, EWB and EWB_1 must not be replaced.

2.2.2 Behavior of Building and Split System

The split system is controlled by the two-point controller of the compressor. For the controlling system there are valid following important physical underlying assumptions given in the case definitions. If the zone temperature is greater than the setpoint, then the compressor immediately starts. Otherwise, it switches off. That means there is no given hysteresis or the hysteresis is set to 0 K, respectively. Second, once the compressor starts running, the coil capacities immediately reach the values of the given performance map. If it turns off, the coil capacities are equal to zero. That means the dynamic coil behavior has been neglected. With these above assumptions Figure 2-6 represents the behavior of the split system and the building.

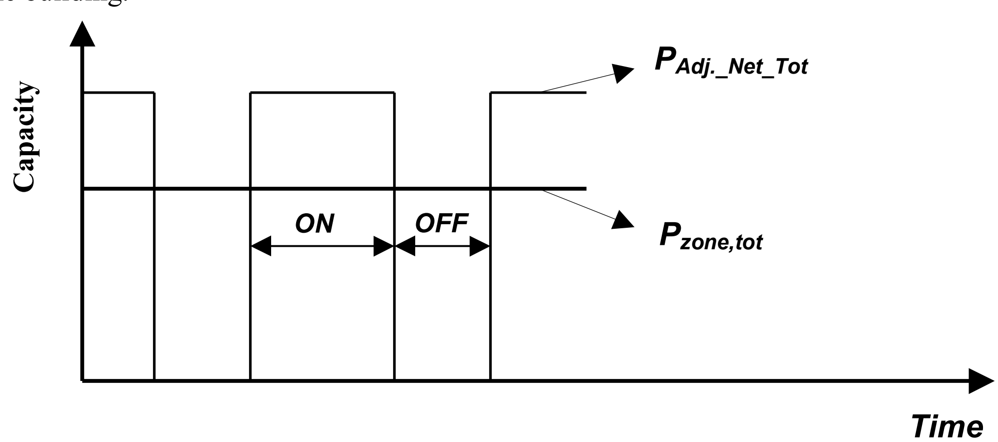

Figure 2-6. Behaviors of system in part load operation

According to the definition of the Part Load Ratio, one can derive equations (2-7) and (2-8) below:

$$\frac{ON}{ON + OFF} = \frac{P_{zone\_tot}}{P_{Adj.\_Net\_Tot}} = PLR \tag{2-7}$$

$$\Leftrightarrow \quad P_{zone\_tot} = PLR * P_{Adj.\_Net\_Tot} \tag{2-8}$$

where:

$$P_{zone\_tot} = P_{zone\_sen} + P_{zone\_lat}$$

$$P_{Adj.\_Net\_Tot} = P_{Adj.\_Net\_Sen} + P_{Adj.\_Net\_Lat}$$

$$\Rightarrow P_{zone\_sen} + P_{zone\_lat} = PLR * (P_{Adj.\_Net\_Sen} + P_{Adj.\_Net\_Lat}) \tag{2-9}$$

At the steady-state operating point, the sensible portion of the adjusted net capacity has to match the sensible portion of zone load. And the latent part of the adjusted net capacity has to be equal the part of latent zone load. From this consideration and equation (2-9), equations (2-10) and (2-11) are derived:

$$P_{zone\_sen} = PLR * P_{Adj.\_Net\_Sen} \tag{2-10}$$

$$P_{zone\_lat} = PLR * P_{Adj.\_Net\_Lat} \tag{2-11}$$

To divide equation (2-10) by (2-8):

$$\frac{P_{zone\_sen}}{P_{zone\_tot}} = \frac{PLR * P_{Adj.\_Net\_Sen}}{PLR * P_{Adj.\_Net\_Total}} \tag{2-12}$$

with: $$SHR = \frac{P_{sen}}{P_{tot}} \tag{2-13}$$

Criterion at steady-state operating point for analytical solution is derived from these equations (2-12) and (2-13):

$$\boxed{SHR_{zone} = SHR_{Adj.\_Net\_Capacity}} \tag{2-14}$$

2.2.2.1 Dry Coil Conditions

For this case, the latent zone load is zero. So,

$$P_{zone\_sen} = P_{zone\_tot} \tag{2-15}$$

$$SHR_{zone} = 1 \tag{2-16}$$

From equations (2-14) and (2-16) one derives:

$$P_{Adj.\_Net\_Sen} = P_{Adj.\_Net\_Tot} \qquad (2\text{-}17)$$

As known from above (Subsection 2.2.1), $P_{Adj.\_Net\_Tot}$ and $P_{Adj.\_Net\_Sen}$ are linear function of EWB, EDB, and ODB. But ODB and EDB are given, so solved EWB from equation (2-17) is the intersection point EWB_1 of both lines sensible and total capacities (see Figure 2-5).

Equation (2-17) means for dry coil conditions the evaporator coil only takes sensible energy away at the steady-state operating point. To determine the stationary operating point it is necessary to know the initial entering wet bulb temperature. Initialization of zone conditions at the beginning is equal to the outdoor conditions per the test specification amendment of December 16, 1999. So the initial entering wet bulb temperature can be calculated from the outdoor conditions.

If the initial value of entering wet bulb temperature is greater than the intersection point, then the coil takes away the sensible as well as the latent energy (operation under wetcoil conditions at the beginning). Because the latent zone load is equal to zero, the entering wet bulb temperature and the zone humidity ratio continuously decrease until the coil becomes dry. That means the entering wet bulb temperature and the intersection point are identical. In this case, the steady-state operating point is the intersection point.

If the initial entering wet bulb temperature is less than the intersection point, the coil operates under drycoil conditions at the beginning. So, the zone humidity remains constant. On the other hand, the zone temperature decreases to the set point because of cooling. The entering wet bulb temperature at steady-state operating point is determined from the set point temperature and the zone humidity ratio.

Note: For the E100 series cases, the initial EWB is always at or above the intersection point because zone humidity ratio is initially the ambient humidity ratio, and ambient humidity ratio is always 0.01 kg/kg.

2.2.2.2 Wet Coil Conditions

For this case, the sensible and the latent zone load are present. So,

$$P_{zone\_sen} < P_{zone\_tot}$$
$$0 < SHR_{zone} < 1$$

That means the coil takes away the sensible as well as the latent energy. The steady-state operating point EWB_2 (Figure 2-5) is solved from equation (2-14).

2.3 Determination of Supply Fan Heat

$P_{Adj.\_Net\_Capacity} = P_{ECL\_Capacity} - P_{IDfan}$; $P_{ECL\_Capacity} =$ constant; but P_{IDfan} depends on the PLR (Part Load Ratio) considering the CDF factor. This is because at part loads the system run time is extended. So, there is some additional fan heat that should be accounted for that is not included in Table 1-6e of the User‘s Manual (which gives adjusted net capacities for full load operation). Therefore, in order to determine the indoor fan heat a few iterations are required.

2.4 Results

If the steady-state operating point is known, then the operating time and all required outputs for a given time period are calculable, e.g.:

$$ZHR = f(EWB_{steady\text{-}state}, EDB)$$

$$t_B = f(PLR; \text{time period})$$

$$CDF = f(PLR)$$

$$Q_{Compressor} = f(CDF; P_{Compressor}; t_B)$$

$$Q_{IDfan} = f(CDF; P_{IDfan}; t_B)$$

$$Q_{ODfan} = f(CDF; P_{ODfan}; t_B)$$

$$Q_{Adj.\_Net\_Sen} = f(P_{Adj.\_Net\_Sen}; t_B)$$

$$Q_{Adj.\_Net\_Lat} = f(P_{Adj.\_Net\_Lat}; t_B)$$

$$Q_{Adj.\_Net\_Tot} = f(P_{Adj.\_Net\_Tot}; t_B)$$

$$Q_{zone,sen} = f(P_{zone,sen}; \text{time period})$$

$$Q_{zone,lat} = f(P_{zone,lat}; \text{time period})$$

$$Q_{zone,tot} = f(P_{zone,tot}; \text{time period})$$

$$COP = f(Q_{zone,tot}; Q_{Compressor}; Q_{IDfan}, Q_{ODfan})$$

Examples to illustrate the analytical solution for Wet and Drycoil Conditions are located in Subsection 3 below with the application of a constant pressure of 101,325 Pa.

2.5 Summary

The subsections above show, step by step, how to solve the problem of HVAC BESTEST with an analytical solution. Its algorithm is expressed in the flowcharts of Figures 2-7 and 2-8.

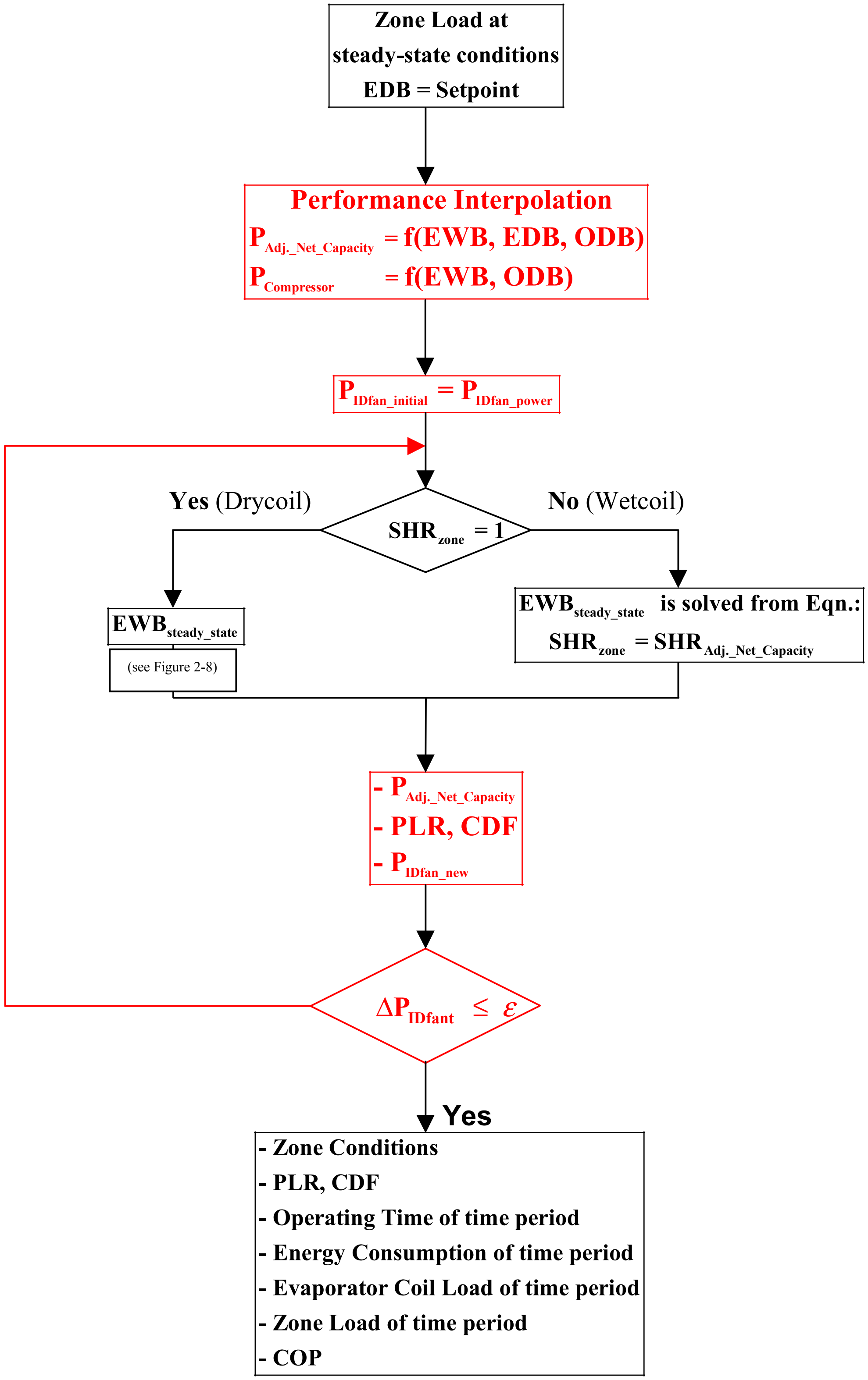

Figure 2-7. Flowchart for the analytical solution

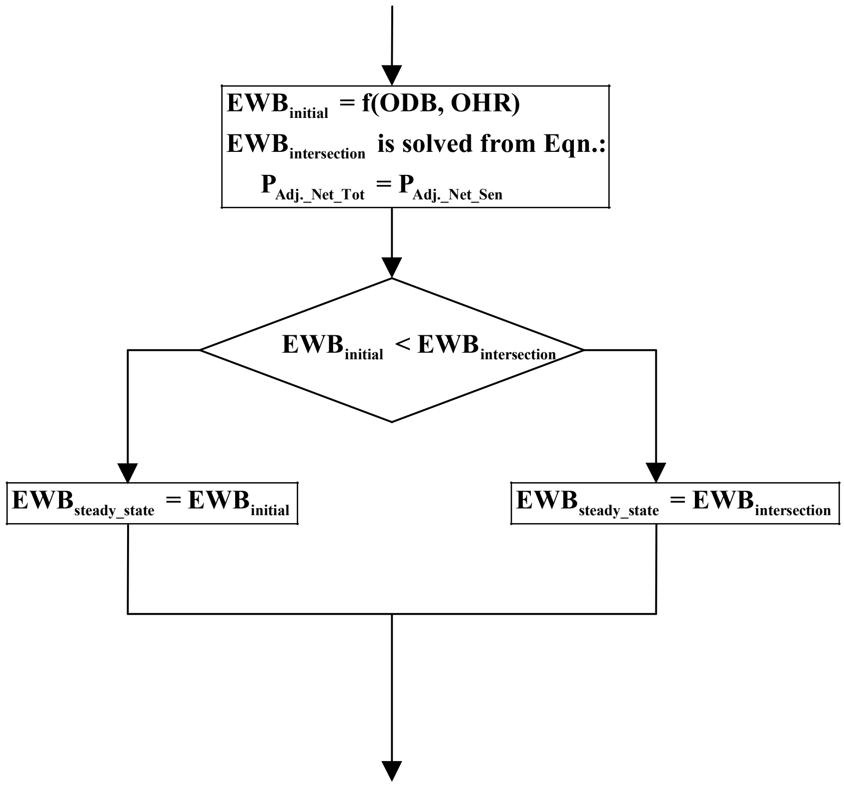

Figure 2-8. Calculation EWB_{steady_state} for drycoil condition by the analytical solution

3.0 Examples for Illustration of the Algorithm Analytical Solution of HVAC BESTEST

3.1 Dry Coil Conditions (Case E110)

3.1.1 Zone Load

<u>Energy flux through the building envelope:</u>

According to equation (2-1) it results:

$$P_{envelope,sen} = \Delta\vartheta * A * U = 7.22 \ K * 171.6 \ m^2 * 0.01 \ \frac{W}{m^2K} = 12.36 \ W$$

where:

$$\Delta\vartheta = ODB - EDB = 29.4 - 22.2 = 7.2 \ K \ (EDB = Setpoint)$$

$$A = 2*(8*6 + 8*2.7 + 6*2.7) = 171.6 \ m^2 \ \text{(Total surfaces of the Building)}$$

$$U = 0.01 \ \frac{W}{m^2K}$$

<u>Total zone load:</u>

The total zone load is calculable according to equation (2-2):

$$P_{zone,tot} = 5412.36 \ W$$

where:

$$P_{envelope,lat} = 0 \ W \ ; \ P_{envelope,sen} = 12.36 \ W$$

$$P_{gain,lat} = 0 \ W \ ; \ P_{gain,sen} = 5400 \ W$$

3.1.2 Steady-State Operating Point

<u>Intersection point:</u>

Intersection point EWB_1 is solved from equation (2-17) by given ODB = 29.4 and EDB = 22.2°C.

EWB_1 = 13.93°C

<u>Initial value for EWB:</u>

Zone condition at the beginning is equal to the outdoor condition (ODB = 29.4°C and outdoor relative humidity (OHR) = 39%). So, $EWB_{initial}$ is a function of ODB and OHR.

$EWB_{initial}$ = 19.43°C

Note: $EWB_{initial}$ is calculated based on the following formulas:

$$\varphi = \frac{x}{0.6222 + x} * \frac{p_{tot}}{p_{sat}(\vartheta)}$$

$$p_{sat}\ (\vartheta) = 288.68 * \left(1.098 + \frac{\vartheta}{100}\right)^{8.02} \quad (\text{for } 0°C < \vartheta < 100°C) \tag{3-1}$$

$$h = 1.006 * \vartheta + x*(2500 + 1.86 * \vartheta)$$

$$\Rightarrow\ h_{zone,initial} = 1.006 *ODB + \frac{0.6222 * \varphi * p_{sat}(ODB)}{p_{tot} - \varphi * p_{sat}(ODB)} * \left(2500 + 1.86 *ODB\right) \tag{3-2}$$

$$h_{sat,initial} = 1.006*EWB_{initial} + x_{sat,initial}*(2500 + 1.86*EWB_{initial}) \tag{3-3}$$

$$x_{sat,initial} = 0.6222 * \frac{p_{sat}(EWB_{initial})}{p_{tot} - p_{sat}(EWB_{initial})} \tag{3-4}$$

$$\left.\frac{\partial h}{\partial x}\right|_{initial} = 4.18 * EWB_{initial} \tag{3-5}$$

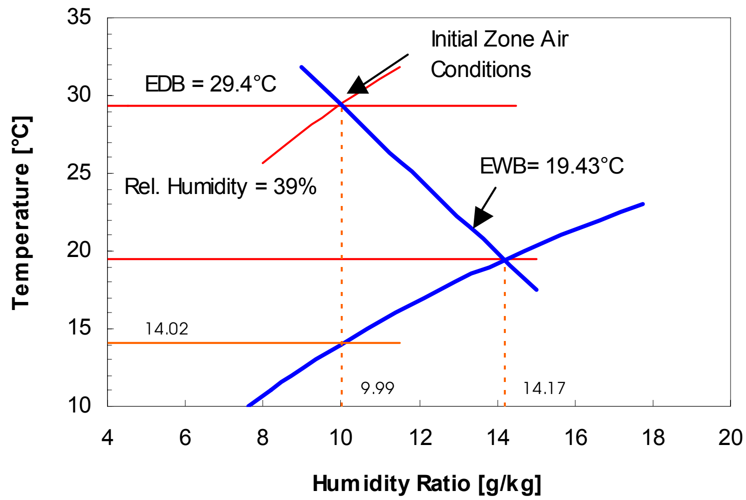

Figure 3-1. Initial zone condition identical to outside condition

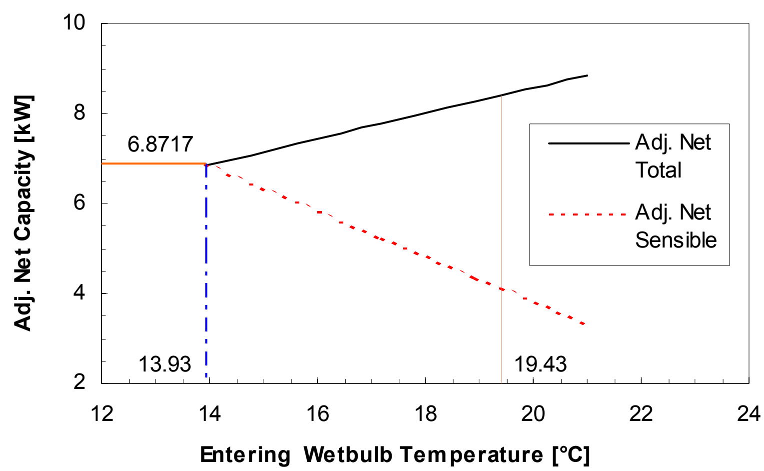

Figure 3-2. Determination of EWB_{steady_state} for Case E110

Figure 3-1 illustrates the initialization of zone conditions on the psychrometric chart and Figure 3-2 represents the intersection point EWB_1 and the initial EWB with a derived custom curve from using the performance map.

According to Subsection 2.2.2.1 above, the steady-state operating point results:

$$\mathbf{EWB_{steady\text{-}state} = 13.93°C}$$

3.1.3 Iteration Loop for Determination of Supply Fan Heat

From equations (2-3) and (2-4) and Figure 3-2, at $EWB_{steady\text{-}state}$ = 13.93°C, the following values are determined with $P_{IDfan\_initial}$ = 230 W:

$$P_{Adj.\_Net\_Lat} = 0 \text{ kW}$$
$$P_{Adj.\_Net\_Sen} = 6.8717 \text{ kW}$$
$$P_{Adj.\_Net\_Tot} = 6.8717 \text{ kW}$$

$$PLR = \left(\frac{P_{zone,tot}}{P_{Adj.\_Net\_Tot}}\right) = \frac{5.41236 \text{ kW}}{6.8717 \text{ kW}} = 0.78763$$

$$CDF = 1 - 0.229*(1 - PLR) = 0.95137$$

Now adjust for additional fan heat due to additional system runtime required for part load operation.

$$P_{IDfan,new1} = \frac{P_{IDfan}}{CDF} = \frac{230 \text{ W}}{0.95137} = 241.76 \text{ W}$$

With the $P_{IDfan,new1}$ all values have to be determined again.

$$P_{Adj.\_Net\_Lat} = 0 \text{ kW}$$
$$P_{Adj.\_Net\_Sen} = 6.860 \text{ kW}$$
$$P_{Adj.\_Net\_Tot} = 6.860 \text{ kW}$$

$$PLR = \left(\frac{P_{zone,tot}}{P_{Adj.\_Net\_Tot}}\right) = \frac{5.41236 \text{ kW}}{6.860 \text{ kW}} = 0.788978$$

$$CDF = 1 - 0.229 *(1 - PLR) = 0.951676$$

$$P_{IDfan,new2} = \frac{P_{IDfan}}{CDF} = 241.68 \text{ W}$$

This is exact enough so that additional iterations would not be necessary.

3.1.4 Results (Required Outputs)

Zone conditions:

From $EWB_{steady\text{-}state}$ and the EDB = Setpoint, the following values are determined on the principles of the psychrometric chart as Figure 3-1:

$$x_{sat.}(EWB_{steady\text{-}state}) = 9.936\ \frac{g}{kg} \qquad \text{(s. Eqn. 3-4)}$$

$$h_{Sat.}(EWB_{steady\text{-}state}) = 39.113\ \frac{kJ}{kg} \qquad \text{(s. Eqn. 3-3)}$$

$$\Rightarrow x_{Zone} = 6.517\ \frac{g}{kg}$$

with:

$$x_{zone} = \frac{h_{sat.} - 4.186 * EWB_{steady\text{-}state} * x_{sat.} - 1.006 * EDB)}{2500 + 1.86 * EDB - 4.186 * EWB_{steady\text{-}state}} \qquad \text{(derived from Eqn. 3-1 to 3-5)}$$

$$\text{and the belonging Dew Point Temperature}: DPT = 7.672\ °C$$

Operating hours for the month of February:

$$t_B = PLR \times 24 \times 28 = 530.193\ h$$

Energy consumption:

With the above-determined steady-state operating point, the compressor power at full load operation can be easily calculated (equation [2-5]). Figure 3-3 illustrates the characteristic curve of compressor power.

$$P_{Compressor} = 1.5939\ kW$$

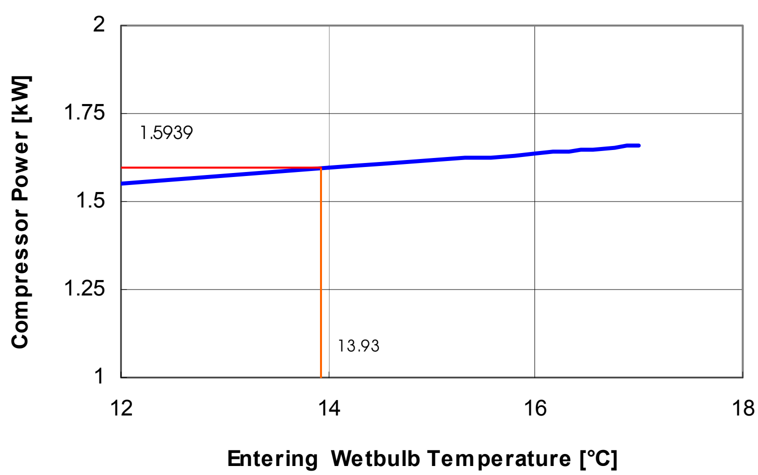

Figure 3-3. Characteristic curve of compressor power at full load operation

Compressor: $Q_{Compressor} = \dfrac{P_{Compressor} \times t_B}{CDF} = \dfrac{1.5939 \text{ kW} \times 530.193 \text{ h}}{0.951676} = 887.98 \text{ kWh}$

Indoorfan: $Q_{IDfan} = \dfrac{P_{IDfan} \times t_B}{CDF} = \dfrac{0.23 \text{ kW} \times 530.193 \text{ h}}{0.951676} = 128.14 \text{ kWh}$

Condenser Fan: $Q_{ODfan} = \dfrac{P_{ODfan} \times t_B}{CDF} = \dfrac{0.108 \text{ kW} \times 530.193 \text{ h}}{0.951676} = 60.17 \text{ kWh}$

Net refrigeration effect:

Latent: $Q_{Adj.\_Net\_Lat} = P_{Adj.\_Net\_Lat} \times t_B = 0 \times 530.193 \text{ h} = 0 \text{ kWh}$

Sensible: $Q_{Adj.\_Net\_Sen} = P_{Adj.\_Net\_Sen} \times t_B = 6.860 \times 530.193 \text{ h} = 3637.12 \text{ kWh}$

Total: $Q_{Adj.\_Net\_Tot} = Q_{Adj.\_Net\_Lat} + Q_{Adj.\_Net\_Sen} = 3637.12 \text{ kWh}$

Evaporator coil load:

Latent: $Q_{ECL\_Lat} = Q_{Adj.\_Net\_Lat} = 0 \text{ kWh}$

Sensible: $Q_{ECL\_Sen} = Q_{Adj.\_Net\_Sen} + Q_{IDfan} = 3765.26 \text{ kWh}$

Total: $Q_{ECL\_Tot} = Q_{ECL\_Lat} + Q_{ECL\_Sen} = 3765.26 \text{ kWh}$

Zone load:

Latent: $Q_{zone,lat} = P_{zone,lat} \times 24 \times 28 = 0 \text{ kW} \times 24 \times 28 = 0 \text{ kWh}$

Sensible: $Q_{zone,sen} = P_{zone,sen} \times 24 \times 28 = 5.41239 \text{ kW} \times 24 \times 28 = 3637.13 \text{ kWh}$

Total: $Q_{zone,tot} = Q_{zone,lat} + Q_{zone,sen} = 3637.13 \text{ kWh}$

COP:

Way 1:

COP at full load operation:

$$COP_{full} = \frac{P_{Adj.\_Net\_Tot}}{P_{Compressor} + P_{IDfan} + P_{ODfan}}$$

$$= \frac{6.86 \text{ kWh}}{1.5939 \text{ kWh} + 0.23 \text{ kWh} + 0.108 \text{ kWh}} = 3.5509$$

COP at part load operation:

$$COP_{part} = COP_{full} \times CDF = 3.5509 \times 0.951676 = 3.379$$

Way 2:

$$COP_{part} = \frac{Q_{zone,tot}}{Q_{Compressor} + Q_{IDfan} + Q_{ODfan}}$$

$$= \frac{3637.13 \text{ kWh}}{887.98 \text{ kWh} + 128.14 \text{ kWh} + 60.17 \text{ kWh}} = 3.379$$

3.2 Wet Coil Conditions (Case E170)

3.2.1 Zone Load

Energy flux through the building envelope:

According to equation (2-1) it results:

$$P_{envelope,sen} = \Delta\vartheta * A * U = 7.22\ K * 171.6\ m^2 * 0.01\ \frac{W}{m^2K} = 12.36\ W$$

where:

$$\Delta\vartheta = ODB - EDB = 29.4 - 22.2 = 7.2\ K\ \ (EDB = Setpoint)$$

$$A = 2*(\ 8*6 + 8*2.7 + 6*2.7\ ) = 171.6\ m^2\ \ (\text{Total surfaces of the Building})$$

$$U = 0.01\ \frac{W}{m^2K}$$

Total zone load:

The total zone load is calculable according to equation (2-2):

$$P_{zone,lat} = 1100\ W$$

$$P_{zone,sen} = 2112.36\ W$$

$$P_{zone,tot} = 3212.36\ W$$

where:

$$P_{envelope,lat} = 0\ W\ ; \quad P_{envelope,sen} = 12.36\ W$$

$$P_{gain,lat} = 1100\ W\ ; \quad P_{gain,sen} = 2100\ W$$

3.2.2 Steady-State Operating Point

For the case of wet-coil conditions, the evaporator coil takes away the sensible as well as the latent energy. The steady-state operating point is solved from equation (2-14) with

$$ODB = 29.4\ °C; \quad EDB = 22.2\ °C \quad \text{and}$$

$$SHR|_{zone} = \frac{P_{zone,sen}}{P_{zone,tot}} = \frac{2112.36}{3212.36} = 0.657576$$

$$\Rightarrow\ EWB_2 = 17.354\ °C$$

Figure 3-4 represents just solved EWB_2 with a derived custom curve from using the performance map and Figure 3-5 illustrates the depending zone conditions on the EWB_2 in the psychrometric chart.

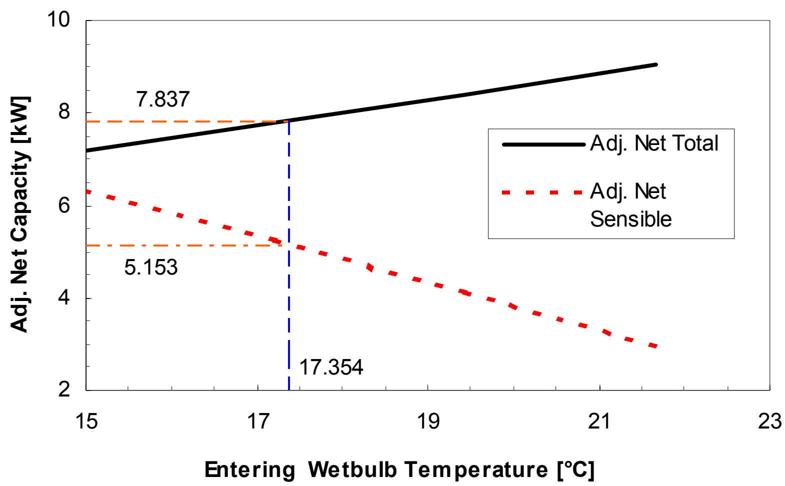

Figure 3-4. Solved EWB_2 from equation (2-14) for Case E170

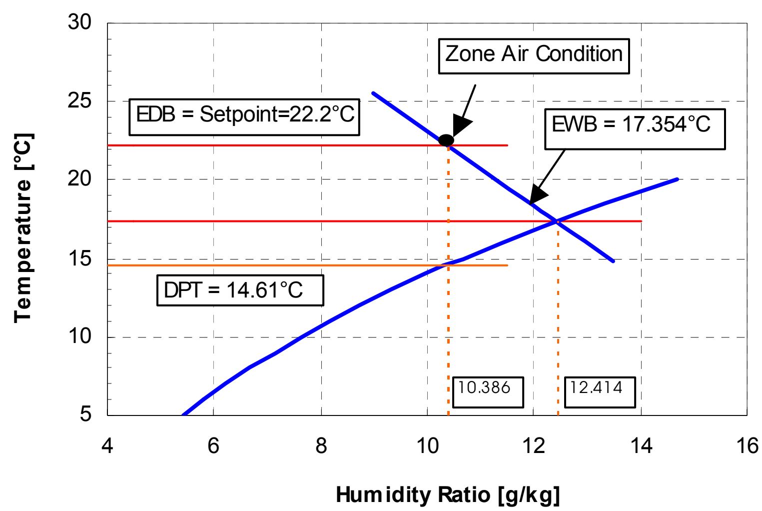

Figure 3-5. Depending zone air condition on EWB at the stationary condition

$EWB_{steady\text{-}state}$ is solved from equation (2-14), SHR_{zone} is constant, but $SHR_{Adj.\_Net\_Capacity}$ becomes a minor change considered depending supply fan heat of CDF factor at part load operation, because the system run time is extended. So, determination of $EWB_{steady\text{-}state}$ is involved within iteration loop of the supply fan heat.

3.2.3 Iteration Loop for Determination of Supply Fan Heat

From equations (2-3) and (2-4) and Figure 3-4, at EWB_2 = 17.354°C, the following values can be determined with $P_{IDfan\_initial}$ = 230 W:

$$P_{Adj.\_Net\_Lat} = 2.684 \text{ kW}$$
$$P_{Adj.\_Net\_Sen} = 5.153 \text{ kW}$$
$$P_{Adj.\_Net\_Tot} = 7.837 \text{ kW}$$

$$PLR = \left(\frac{P_{zone,tot}}{P_{Adj.\_Net\_Tot}} \right) = \frac{3.21236 \text{ kW}}{7.837 \text{ kW}} = 0.4099$$

$$CDF = 1- 0.229*(1- PLR) = 0.8648672$$

Now adjust for additional fan heat due to additional system runtime required for part load operation.

$$P_{IDfan,new1} = \frac{P_{IDfan}}{CDF} = \frac{230 \text{ W}}{0.8648672} = 265.94 \text{ W}$$

With the $P_{IDfan,new1}$ all values have to be determined again.

$$EWB2 = 17.336 \text{ °C}$$

$$\Rightarrow \begin{aligned} P_{Adj.\_Net\_Lat} &= 2.6695 \text{ kW} \\ P_{Adj.\_Net\_Sen} &= 5.1265 \text{ kW} \\ P_{Adj.\_Net\_Tot} &= 7.7960 \text{ kW} \end{aligned}$$

$$PLR = \left(\frac{P_{zone,tot}}{P_{Adj.\_Net\_Tot}} \right) = \frac{3.21236 \text{ kW}}{7.7960 \text{ kW}} = 0.412056$$

$$CDF = 1- 0.229 *(1- PLR) = 0.865361$$

$$P_{IDfan,new2} = \frac{P_{IDfan}}{CDF} = 265.78 \text{ W}$$

This is exact enough so that additional iterations would not be necessary. And it results in the steady-state operating point:

$$\mathbf{EWB_{steady\text{-}state} = 17.336 \text{ °C}}$$

3.2.4 Results (Required Outputs)

__Zone conditions:__

From $EWB_{steady\text{-}state}$ and the EDB = Setpoint, the following values are determined on the principles of the psychrometric chart in Figure 3-5.

$$x_{sat.}(EWB_{steady\text{-}state}) = 12.40\ \frac{g}{kg} \qquad \text{(s. Eqn. 3-4)}$$

$$h_{Sat.}(EWB_{steady\text{-}state}) = 48.84\ \frac{kJ}{kg} \qquad \text{(s. Eqn. 3-3)}$$

$$\Rightarrow x_{Zone} = 10.366\ \frac{g}{kg}$$

with:

$$x_{zone} = \frac{h_{sat.} - 4.186 * EWB_{steady\text{-}state} * x_{sat.} - 1.006 * EDB)}{2500 + 1.86 * EDB - 4.186 * EWB_{steady\text{-}state}} \qquad \text{(derived from Eqn. 3-1 to 3-5)}$$

and the belonging Dew Point Temperature: $DPT = 14.574°C$

Operating hours for the month of February:

$$t_B = PLR \times 24 \times 28 = 276.902\ \ h$$

Energy consumption:

With the above-determined steady-state operating point, the compressor power at full load operation can be easily calculated (equation [2-5]). Figure 3-6 illustrates the characteristic curve of compressor power.

$$P_{Compressor} = 1.6657\ kW$$

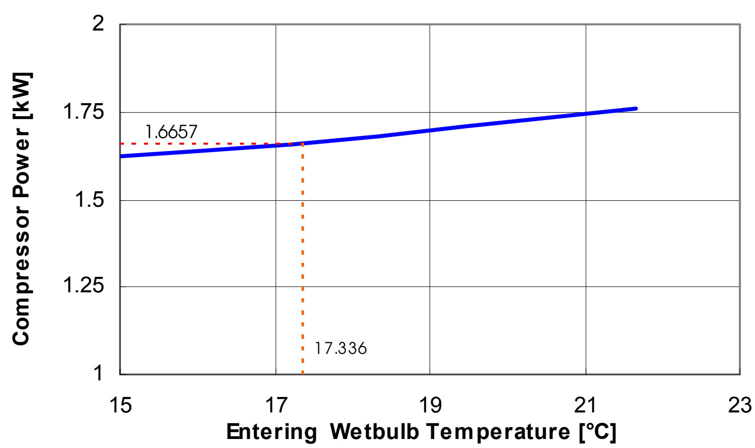

Figure 3-6. Characteristic curve of compressor power at full load operation

Compressor: $$Q_{Compressor} = \frac{P_{Compressor} \times t_B}{CDF} = \frac{1.6657\ \ kW \times 276.902\ \ h}{0.865361} = 532.999\ \ kWh$$

Indoorfan: $$Q_{IDfan} = \frac{P_{IDfan} \times t_B}{CDF} = \frac{0.23\ \ kW \times 276.902\ \ h}{0.865361} = 73.596\ \ kWh$$

Condenser Fan: $$Q_{ODfan} = \frac{P_{ODfan} \times t_B}{CDF} = \frac{0.108\ \ kW \times 276.902\ \ h}{0.865361} = 34.558\ \ kWh$$

Net refrigeration effect:

$$\text{Latent:}\quad Q_{Adj.\_Net\_Lat} = P_{Adj.\_Net\_Lat} \times t_B = 2.6695 \times 276.902 \text{ h} = 739.19 \text{ kWh}$$

$$\text{Sensible:}\quad Q_{Adj.\_Net\_Sen} = P_{Adj.\_Net\_Sen} \times t_B = 5.1265 \times 276.902 \text{ h} = 1419.54 \text{ kWh}$$

$$\text{Total:}\quad Q_{Adj.\_Net\_Tot} = Q_{Adj.\_Net\_Lat} + Q_{Adj.\_Net\_Sen} = 2158.73 \text{ kWh}$$

Evaporator coil load:

$$\text{Latent:}\quad Q_{ECL\_Lat} = Q_{Adj.\_Net\_Lat} = 739.19 \text{ kWh}$$

$$\text{Sensible:}\quad Q_{ECL\_Sen} = Q_{Adj.\_Net\_Sen} + Q_{IDfan} = 1493.14 \text{ kWh}$$

$$\text{Total:}\quad Q_{ECL\_Tot} = Q_{ECL\_Lat} + Q_{ECL\_Sen} = 2232.33 \text{ kWh}$$

Zone load:

$$\text{Latent:}\quad Q_{zone,lat} = P_{zone,lat} \times 24 \times 28 = 1.1 \text{ kW} \times 24 \times 28 = 739.20 \text{ kWh}$$

$$\text{Sensible:}\quad Q_{zone,sen} = P_{zone,sen} \times 24 \times 28 = 2.11236 \text{ kW} \times 24 \times 28 = 1419.53 \text{ kWh}$$

$$\text{Total:}\quad Q_{zone,tot} = Q_{zone,lat} + Q_{zone,sen} = 2158.73 \text{ kWh}$$

COP:

Way 1:

COP at full load operation:

$$COP_{full} = \frac{P_{Adj.\_Net\_Tot}}{P_{Compressor} + P_{IDfan} + P_{ODfan}}$$

$$= \frac{7.796 \text{ kWh}}{1.6657 \text{ kWh} + 0.23 \text{ kWh} + 0.108 \text{ kWh}} = 3.891$$

COP at part load operation:

$$COP_{part} = COP_{full} \times CDF = 3.891 \times 0.865361 = 3.367$$

Way 2:

$$COP_{part} = \frac{Q_{zone,tot}}{Q_{Compressor} + Q_{IDfan} + Q_{ODfan}}$$

$$= \frac{2158.73 \text{ kWh}}{532.999 \text{ kWh} + 73.596 \text{ kWh} + 34.558 \text{ kWh}} = 3.367$$

4.0 Comparison of Submitted Models for the Analytical Solution from HTAL and TUD

HTAL stands for Hochschule fuer **T**echnik+**A**chitektur **L**uzern, Switzerland
TUD stands for Technische **U**niversität **D**resden

The term "analytical solution" was previously used but without submitting the models described in [19] and [20].

4.1 Summary Comparison prior to March 2000 ([8], [15], [16] and [17])

The submitted algorithms for analytical solutions of HTAL [15] and of TUD [16, 17] build a basis for the comparison. These papers have appeared in [8].

- The HTAL model (HTAL2) is not an analytical solution, but it is a simulation with a:
  - realistic controller and
  - 1-s time step
- The TUD algorithm is a genuine analytical solution at steady-state room conditions
- HTAL did not incorporate the coefficient of performance degradation factor (CDF) with CDF=f(PLR). This causes a deviation over 15% of energy consumption and over 15% difference of COP factor
- Use of 100,000 Pa by HTAL for Patm in psychrometric equations (TUD used 101,000 Pa)
- HTAL used Cp for liquid water instead of for water vapor in one of the HTAL equations
- TUD calculates EWB by matching SHR of zone load to SHR of the performance. This is the key for solving the HVAC BESTEST with the analytical solution
- HTAL computes the EWB iteratively from the known room conditions
- TUD calculates more precise accounting of supply fan heat as a function of Part Load Ratio than HTAL
- TUD did not show what data points of performance map were applied for interpolation.
- TUD assumed the EWB lines and enthalpy line on the psychrometric chart are parallel, so the secondary term $\Delta h = h_{Sat.} - h_{Zone}$ was ignored
- Use of adiabatic envelope by HTAL; test spec gives a near-adiabatic envelope.

4.2 Summary Comparison during March–June 2000 ([3–7])

After the first round of comparison (see Subsection 4.1, immediately above), HTAL submitted revised solutions including a new "100%-analytical" solution (**very similar technique to TUD's**). HTAL also revised the version of their original solution technique incorporating the changes of CDF factor to match TUD results.

After development of the new 100%-analytical solution, the new analytical solution results are indicated by "HTAL1" and the original solution technique (applying the realistic controller with a 1-s time step) as "HTAL2."

Below is a summary comparison of model TUD and HTAL1, because the original solution technique (HTAL2) is not the genuine analytical solution.

At this time, TUD showed the data points they applied for interpolation.

Up to June 2000 TUD did not change their solution techniques, but they included the summary comparison of two models HTAL1 and TUD to show these similarities and differences between them (see modeler reports of TUD and of HTAL in [3] and [8]).

4.2.1 Similarities in the Models of HTAL1 and TUD

In general, both algorithms are similar but with different formulations.

4.2.1.1 Approximation of the Performance Map

For approximation of the performance map of this equipment,

TUD uses linear curve $Y = a*X + b$.

HTAL1 applies linear curve with $Y = \left(\frac{Y_2 - Y_1}{X_2 - X_1}\right) * \left(X - X_1\right) + Y_1$.

Note: This approximation is needed for analytical solution as well as for simulation.

4.2.1.2 Steady-State Operating Point

At the steady-state operating point,

TUD formulates $SHR_{zone} = SHR_{equipment}$; where $SHR = \frac{P_{sensible}}{P_{total}}$.

HTAL1 formulates $r_{zone} = r_{equipment}$; where $r = \frac{P_{sensible}}{P_{latent}}$.

$$SHR_{zone} = SHR_{equipment} \text{ ; where } SHR = \frac{P_{sensible}}{P_{total}} = \frac{P_{sensible}}{P_{sensible} + P_{latent}}$$

Note:

$$\Leftrightarrow \left(\frac{P_{sensible}}{P_{sensible} + P_{latent}}\right)_{zone} = \left(\frac{P_{sensible}}{P_{sensible} + P_{latent}}\right)_{equipment}$$

$$\Leftrightarrow \left(\frac{P_{sensible} + P_{latent}}{P_{sensible}}\right)_{zone} = \left(\frac{P_{sensible} + P_{latent}}{P_{sensible}}\right)_{equipment}$$

$$\Leftrightarrow 1 + \left(\frac{P_{latent}}{P_{sensible}}\right)_{zone} = 1 + \left(\frac{P_{latent}}{P_{sensible}}\right)_{equipment}$$

$$\Leftrightarrow \underline{\underline{\left(\frac{P_{sensible}}{P_{latent}}\right)_{zone} = \left(\frac{P_{sensible}}{P_{latent}}\right)_{equipment}}}$$

This formulation is the key for solving the HVAC BESTEST analytical solution. The formulation gives the results shown in Table 4-1.

Table 4-1. Results for drycoil and wetcoil conditions

	TUD	HTAL1
Dry Coil Conditions	SHR =1	$r = \infty$
Wet Coil Conditions	0< SHR < 1	$0 < r < \infty$

4.2.2 Differences in the Models of HTAL1 and TUD

4.2.2.1 Regarding the Approximation Method

TUD has analyzed the areas with dry coil and wet coil conditions. It makes clear the behavior of the equipment between these two areas. TUD never used the point where the $P_{total} < P_{sensible}$ for the approximation of the performance map. For determination of the steady-state operating point TUD has only applied the valid point at the performance map with $P_{total} \geq P_{sensible}$. This means that TUD develops linear interpolation/extrapolation formulae over the full extremes of given wetcoil data points. According to the manufacturer, the valid data points contain some uncertainty in the experimental measurements (e-mail from J. Neymark to participants January 10, 2000), so the use of over full extremes valid data points could eliminate this uncertainty.

HTAL1 selects more local intervals that can include invalid total capacities of *drycoil* data points for their calculations. However, according to tables 1-6a to 1-6f in Section 1.3.2.2 (see Part I), the given total capacities are valid only for the wetcoil data points. For example, for Case E120 this results in that HTAL used dry coil total capacity data for linear interpolations/extrapolations.

The Case E120 is an example to illustrate the difference of two models concerning the approximation method.

For Case E120: ODB = 29.4°C and EDB = 26.7°C;

and at this condition of ODB and EDB, the equipment operates with the capacities shown in Table 4-2.

Table 4-2. Operating capacities of the equipment

EWB (°C)	Adj. Net Total Capacity (W)	Adj. Net Sensible Capacity (W)
15.0	7190	7660
17.2	7780	7450
19.4	8420	6310
21.7	9060	5140

Because $P_{total} < P_{sensible}$ at EWB = 15°C TUD only used the capacities at EWB = 17.2; 19.4 and 21.7°C for the approximation, whereas HTAL1 applied the capacities at EWB = 15.0 and 17.2°C. Figures 4-1 and 4-2 make clear the difference.

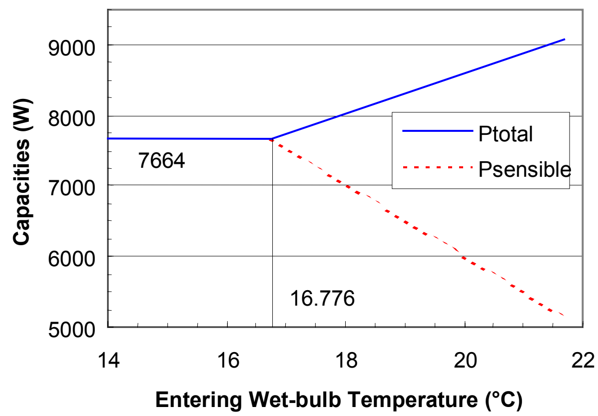

Figure 4-1. Approximation method of TUD

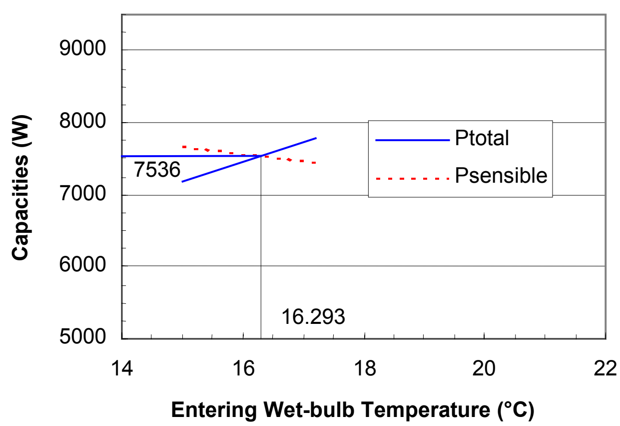

Figure 4-2. Approximation method of HTAL

TUD:

- IF EWB > 16.776°C THEN
  Wet Coil Conditions, where $P_{total} > P_{sensible}$

- IF EWB ≤ 16.776°C THEN
  Dry Coil Conditions, where $P_{total} = P_{sensible}$ = constant

HTAL1: It is unclear with HTAL1 about the areas for Wet and Dry Coil Conditions, e.g. how much are P_{total} and $P_{sensible}$ if EWB ≤ 16.293°C ?

Because of using an invalid total capacity of the dry-coil data point, the zone humidity ratio for Case E120 in the HTAL1 model is deviated from TUD model over 8%.

4.2.2.2 Regarding Using Pressure

TUD has utilized a pressure of 101,000 Pa.
HTAL1 has applied p = 101,325 Pa.

4.2.2.3 Regarding Zone Humidity Ratio (ZHR)

TUD used the following equation to calculate humidity ratio (x_{Zone}):

$$x_{Zone} = (h_{Sat.} - c_{pa} *EDB) / (r_0 + c_{pv}*EDB)$$

where: c_{pa} and c_{pv} are specific heat of air and water vapor respectively;
r_0 enthalpy of vaporization for water at 0°C
$h_{Sat.}$ enthalpy of saturated air at zone EWB.

This assumes the EWB lines and enthalpy lines on the psychrometric chart are parallel, i.e., that $h_{Zone} = h_{Sat.}$ which ignores the secondary term:

$$\Delta h = h_{Sat.} - h_{Zone} = c_{pw}*EWB*(x_{Sat.} - x_{Zone})$$

where: h_{Zone} and x_{Zone} are respectively the enthalpy and humidity ratio of the zone
c_{pw} is the specific heat of liquid water.

HTAL1 makes the more exact calculation as shown in Section 2.3.2, Subsection 4.5.

4.2.2.4 Regarding Steady-State Operating Point

TUD: From analysis of Figure 4-1, IHR is dependent on the initial conditions.

For Case E120, if the start value of EWB is less than 16.776°C (e.g. EWB = 14°C; where ODB = 29.4°C and ambient relative humidity = 22%) then the equipment operates at the steady-state operating point EWB = 14°C.

HTAL1: HTAL did not consider this behavior (Figure 4-2) and did not give any information about the dependence of steady-state operating point on start values.

4.2.2.5 Regarding Calculation of the Equipment Capacities

HTAL1 does not consider the power of supply fan in dependence on Part Load Ratio.

TUD has an additional iteration to increase precision regarding consideration of additional supply fan heat resulting from CDF adjustment at part load (Section 2.3.1., Subsection 2). Because of different considerations, the following deviations result (e.g., for Case E170).

Table 4-3. Differences between TUD and HTAL analytical solution results

Solver Values at Steady-state operating point		TUD	HTAL	Deviation (HTAL-TUD)/ TUD*100
EWB	(°C)	17.33	17.40	0.40
Adjusted Net Total Capacity	(W)	7796	7839	0.55
Adjusted Net Sensible Capacity	(W)	5126	5155	0.57
Adjusted Net Latent Capacity	(W)	2670	2684	0.52
Operating time = PLR*24*28	(h)	276.90	275.39	-0.55

This has the following effect on run time: 0.18% increase for E110; 0.54% increase for E170. For Case E170, electricity consumption, the level of disagreement between HTAL and TUD is similar to that arising from the E170 run time difference. A similar disagreement for E170 coil load was also expected, however consideration of variation of performance parameters indicates that compensating differences in the coil load calculations are possible such that TUD's and HTAL's electricity consumptions can have greater disagreement than the coil loads. For Case E140, the Part Load Ratio is too low for the fan heat f(CDF) calculation difference to have a noticeable impact on the results of energy consumption for supply fan (1.4%).

4.3 Summary Comparison during June–August 2000 ([1, 2])

In July 2000 HTAL submitted corrected results for their HTAL1 and HTAL2 Case E120 solutions using extrapolation of valid total capacity of wet-coil data rather than their previous interpolation between wet-coil and dry-coil data. The new results indicate improved agreement for Case E120 versus the TUD results.

In August 2000 TUD submitted new analytical solution results incorporating Patm = 101,325 Pa into their calculations. TUD also revised their humidity ratio calculation to match HTAL's, so that the previously excluded difference between saturation enthalpy and zone enthalpy has now been included. The change in Patm had a negligible effect on results. The corrected humidity ratio calculation had a maximum effect on zone humidity ratio results of about 1.7%, and no effect on the other results.

TUD uses a precise calculation by applying EDB, ODB with two positions exact after the decimal point, and four positions exact for performances.

TUD includes the following to make clearer the solution method, but no effect on results:

- Basis formulas of equations for calculation saturation conditions and zone conditions (equations [3-1] to [3-5])
- Commentary on the use of terms.

4.4 Conclusion

The remaining differences among the solution techniques include:

- HTAL2 uses a realistic controller with a very short (1-s) timestep; TUD and HTAL1 use an ideal controller
- More localized interpolation intervals used by HTAL1 and HTAL2 than used by TUD, so applying of full extremes given data points by TUD could eliminate the uncertainty in the experimental measurements.

- More precise accounting of fan heat as a function of Part Load Ratio by TUD than by HTAL1 and HTAL2, resulting in more exact:

  - EWB from matching SHR of zone load to SHR of the performance
  - Sensible and latent performances
  - Operating time
  - Part Load Ratio
  - CDF factor
  - Energy consumption
  - COP factor
  - Zone humidity ratio from EWB and Setpoint.

  [For the cases with low PLR—e.g., E130 and E140—the deviation is about 1%, maximum deviation of ZHR for case E140 is about 1.8% (see page 124 of [3])].

The resulting process conformed to the initial goal of obtaining high quality solutions by beginning with independent blind solutions and then allowing a third party, and then eventually the solvers themselves to comment on the work, fix errors, and move toward agreement.

The end result is two initially independent solutions (TUD and HTAL2) that revealed significant differences in the two models. The TUD solutions have been not changed from the beginning unless the way for calculation of zone humidity ratio without/with consideration of the term "enthalpy increasing $\Delta h = h_{Sat.} - h_{Zone}$." This change is not important and only has an effect on zone humidity ratio (a maximum fault of 1.7%). The improvement and the fixing of errors in the HTAL model has been carried out and are now highly agreed (< 2%).

Additionally, a third solution (HTAL1) was developed semi-independently after receiving the TUD solution techniques.

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2.3.2 Analytical Solutions by Hochschule Technik + Architektur Luzern (HTAL)

HVAC - BESTEST

INTERNATIONAL ENERGY AGENCY

Analytical and numerical solution for case E100–E200.
Based on test description from September 1998 edition.

OCTOBER 2000

Hochschule Technik+Architektur Luzern

HLK- Engineering
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6048 Horw
SWITZERLAND

SUMMARY

This report describes the analytical solution for HVAC BESTEST, case E100 to E200. Furthermore, a numerical solution is presented in order to verify the analytical results.

The solution technique (analytical solution) is based on the consideration of an ON and an OFF timestep of the cooling device. The length of both steps is calculated for the steady state. Using the so-calculated wet bulb temperature the humidity ratio of the room can be determined. An empirical model of the saturation curve is used for this purpose. Moreover, the COP and the cooling energy consumption are calculated.

A model with discrete timesteps is used for the numerical solution. The length of the timesteps has been chosen as very small in order to guarantee a minimal deviation from the setpoint. From the known room condition the wet bulb temperature is calculated iteratively as there is no analytical solution. This entering wet bulb temperature is used for the interpolation of the performance data. The sensible capacity is also interpolated with the entering dry bulb temperature. An extrapolation is required in certain cases. The energy consumption is calculated by summing up the performances at every timestep. In order to reach a steady state, a preconditioning period of 4 hours is used. The relevant energy is then summed up in a 1-hour simulation loop, representing the whole simulation period.

Several tests showed that it is essential to include the sensitivity of sensible capacity to EDB in case E100 and E200. Otherwise the results are not reliable as they depend on the start value or numerical difficulties make a calculation impossible.

1 ABBREVIATIONS

Variables

ϑ	Celsius temperature
h	enthalpy
p	pressure
R	gas constant
cp	specific heat at constant pressure
ro	thermal energy for evaporation at 0°C
P	performance (loads and capacities)
dt	timestep
m	mass
Q	energy
V	volume
t	time
NBH	number of hours
PLR	part load ratio
CDF	COP degradation factor
λ	ON ratio
ε	stopping criteria to determine wet bulb temperature (small number)

Indices

R	room
s	saturation condition
i	placeholder for points 0, 1, 2
air	variable relates to air
w	variable relates to water
v	variable relates to water vapor
WB	wet bulb
WB'	wet bulb calculated with h = const.
EWB	entering Wet Bulb (= WB)
EDB	entering dry bulb (= setpoint)
UB	upper boundary
LB	lower boundary
sens	sensible
lat	latent
tot	total
tr	transmission through envelope
rem	removed from system
add	added to system
g	heat gain
id	indoor
od	outdoor
fan	fan
comp	compressor
ev	evaporator
init	initial value (set first time)
end	end of summation time
prec	precalculation phase
is	intersection
prev	value is from previous timestep
sum	sum of a variable
ss	steady state
per	period (of simulation or summation)
env	envelope

Example

$P_{sens,\,g} \Rightarrow$ sensible gain performance

Constants

ro	$2.501*10^6$	[J/kg]	
cp_w	4187	[J/(kg*K)]	
cp_{air}	1006	[J/(kg*K)]	
R_{air}	287.1	[J/(kg*K)]	(dry air)
R_{vap}	461.5	[J/(kg*K)]	
cp_v	1861	[J/(kg*K)]	

INTRODUCTION

We present two solutions for the BESTEST calculation, a hybrid simulation/analytic solution which takes a realistic control strategy into account, and an analytic solution for the steady-state behaviour of the system. The hybrid solution was first presented in a draft of May 1999 and subsequently revised in the May 2000 draft.

Our analytic solution was first presented in the May, 2000 draft and is generally compatible with the original analytic solution of H.T. Le and G. Knabe, first submitted to our attention in their draft of October 1999, which contains essentially §2.1 and §2.2 of their contribution to the August, 2000 report.

ANALYTICAL SOLUTION

Our analytic technique was developed in the context of resolving numerical discrepancies between H.T. Le's and G. Knabe's analytic results of October, 1999 and the results in our earlier draft using hybrid techniques.

In the course of this investigation we also developed a fully analytic solution, which relies on a new and independent criterion for establishing steady state conditions, based on our original control model. The results and some of the computational details are, however, apart from apparent minor differences in the psychrometric data used, consistent with those of the analytic model developed by H.T. Le and G. Knabe.

The chronology of the developing process is described in the preceding Section 2.2 of the report ("Comparison of Analytical Solution Results and Resolution of Disagreements").

The calculation procedure consists essentially of the following steps:

1. Determine the steady state operating point on the performance map
2. Calculate the humidity ratio using the psychrometric equations
3. Determine the part-load operation factor and the corresponding COP degradation factor
4. Sum energy consumption, zone and coil loads.

2 CALCULATION METHOD

One ON and one OFF timestep in steady state are considered in order to calculate the ratio λ, which represents the length of the ON timestep.

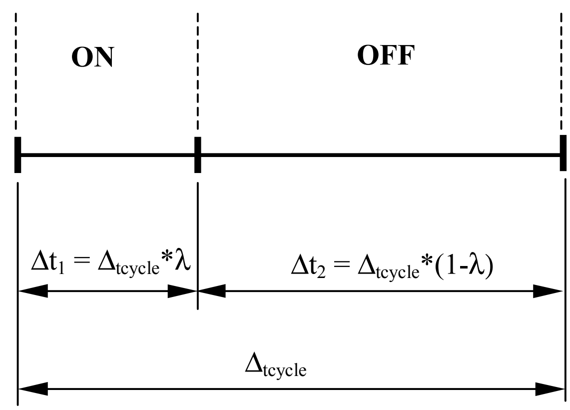


Fig. 2-1

It is assumed that the deviation of the temperature to the setpoint is infinitesimally small. In steady state the energy that is taken away must be equal to the added energy:

$$P_{sens,g} + P_{tr,g} = \lambda * P_{sens} \quad \text{[W]}$$

Eq. 2-1

$$P_{lat,g} = \lambda * P_{lat} \quad \text{[W]}$$

Eq. 2-2

3 LOAD

The sensible and latent loads are given in the test description ([1] Table 1-1a).

For all cases a sensible load caused by transmission has to be added:

$$P_{tr,g} = UA * (\vartheta_{ODB} - \vartheta_R) \quad \text{[W]}$$

Eq. 3-1

4 HUMIDITY RATIO OF ROOM AIR

Equation Eq. 2-1 divided by Eq. 2-2 leads to the ratio r:

$$r = \frac{P_{sens,g} + P_{tr,g}}{P_{lat,g}} = \frac{P_{sens}}{P_{lat}} \quad \text{[-]}$$

Eq. 4-1

4.1 Determination of Wet Bulb Temperature for Wet Coil Cases

ϑ_{WB} has to be calculated such that the resulting unit performances P_{sens} and P_{lat} lead to the ratio 'r' which is defined in Eq. 4-1.

Formula for interpolation from performance table (P stands for P_{sens} respectively P_{lat}):

$$P_{(\vartheta)} = P_1 * \left(\frac{\vartheta_2 - \vartheta}{\vartheta_2 - \vartheta_1} \right) + P_2 * \left(\frac{\vartheta - \vartheta_1}{\vartheta_2 - \vartheta_1} \right) \quad \text{[W]}$$

Eq. 4-2

Equation Eq. 4-1 with P_{sens} and P_{lat} replaced by Eq. 4-2 and simplified gives:

$$r = \frac{P_{1,sens} * (\vartheta_2 - \vartheta_{EWB,ss}) + P_{2,sens} * (\vartheta_{EWB,ss} - \vartheta_1)}{P_{1,lat} * (\vartheta_2 - \vartheta_{EWB,ss}) + P_{2,lat} * (\vartheta_{EWB,ss} - \vartheta_1)} \quad \text{[-]}$$

Eq. 4-3

Solved for ϑ_{EWB} at the steady state:

$$\vartheta_{EWB,ss} = \frac{P_{1,sens} * \vartheta_2 - P_{2,sens} * \vartheta_1 + r * (P_{2,lat} * \vartheta_1 - P_{1,lat} * \vartheta_2)}{P_{1,sens} - P_{2,sens} + r * (P_{2,lat} - P_{1,lat})} \quad \text{[°C]}$$

Eq. 4-4

The correct data from the performance table have to be taken for this calculation. When no extrapolation is required the condition

$$\vartheta_1 \le \vartheta_{EWB,ss} \le \vartheta_2 \quad \text{[°C]}$$

Eq. 4-5

must be fulfilled (ϑ_1 and ϑ_2 are given interval boundaries from the performance table).

4.2 Determination of Wet Bulb Temperature for Dry Coil Cases

For dry coil cases the steady state is reached when the unit has no latent performance (see Fig. 4-1).

The total and sensible performance for a given wet bulb temperature $\vartheta_{EWB:}$ can be attained by linear interpolation from the tabulated data:

$$P_{sens} = \left(\frac{P_{sens,2} - P_{sens,1}}{\vartheta_{EWB,2} - \vartheta_{EWB,1}} \right) * (\vartheta_{EWB} - \vartheta_{EWB,1}) + P_{sens,1}$$

$$P_{tot} = \left(\frac{P_{tot,2} - P_{tot,1}}{\vartheta_{EWB,2} - \vartheta_{EWB,1}} \right) * (\vartheta_{EWB} - \vartheta_{EWB,1}) + P_{tot,1} \quad \text{[W]}$$

Eq. 4-6

At the point of intersection $P_{tot} = P_{sens}$, the equation can be solved to get ϑ_{EWB} at the intersection point:

$$\vartheta_{EWB,ss} = \frac{(P_{tot,1} - P_{sens,1}) * (\vartheta_{EWB,2} - \vartheta_{EWB,1})}{P_{sens,2} - P_{sens,1} - P_{tot,2} + P_{tot,1}} + \vartheta_{EWB,1} \quad \text{[°C]}$$

Eq. 4-7

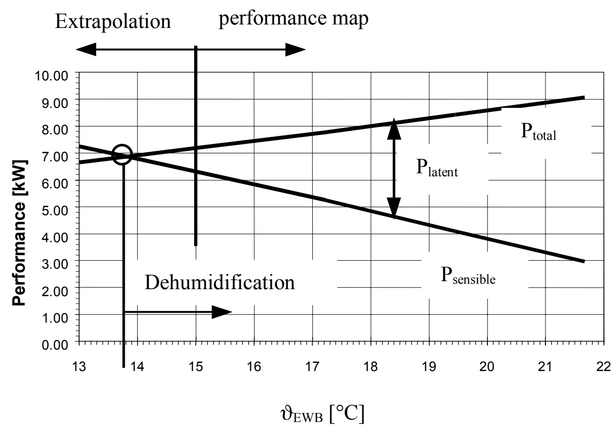


Fig. 4-1: Intersection of total and sensible performance curve

For all cases the intersection of total and sensible performance is determined. The upper and lower boundary of the interval (index 1 and 2 in Table 4-1) in which the intersection occurs is used for the calculation. For all cases except E120, there is no intersection point in the performance table between total and sensible capacity available. Because in case E120 at EWB = 15 °C the sensible capacity is bigger than the total the data at EWB = 17.2 °C are used. Therefore an extrapolation is required in all dry-coil cases (see Fig. 4-1).

Below, for example Case E120:

ODB (°C)	EWB (°C)	Net total Capacity (kW)	Net sensible Capacity **26.7**
29.4	15.0	7.19	7.66
	17.2	7.78	7.45
	19.4	8.42	6.31
	21.7	9.06	5.14

index 1

index 2

Table 4-1

Table 4-2 shows the EWB temperatures, the sensible unit performance and the compressor power at the steady state.

Case	**$\vartheta_{EWB,is}$ [°C]**	**P_{sens} [W]**	**P_{comp} [W]**
E100	14.696	5955.1	2105.9
E110	13.820	6873.4	1598.5
E120	16.792	7661.3	1650.7
E130	14.696	5955.1	2105.9
E140	13.820	6873.4	1598.5

extrapolation required

Table 4-2

In the following sections the humidity ratio for given ϑ_{EDB} and $\vartheta_{EWB,ss}$ is calculated using psychrometric equations.

4.3 Saturation Pressure

The saturation pressure corresponding to $\vartheta_{EWB,ss}$ is calculated using the (empirical) ASHRAE formula [2]:

$$p_{s,ss} = e^{\left(-\frac{5800.2206}{\vartheta_{EWB,ss}+273.15} - 11.89458198 - 0.048640239 * \vartheta_{EWB,ss} + 0.000041764768 * (\vartheta_{EWB,ss}+273.15)^2 - 0.14452093 * 10^{-7} * (\vartheta_{EWB,ss}+273.15)^3 + 6.5459673 * \ln(\vartheta_{EWB,ss}+273.15)\right)}$$

[Pa]

Eq. 4-8

4.4 Saturation Humidity

The humidity ratio on the saturation curve at $\vartheta_{EWB,ss}$ is calculated with $p_{s,ss}$ and the total pressure p_{tot}:

$$x_{s,ss} = \frac{R_{air}}{R_v} * \frac{p_{s,ss}}{p_{tot} - p_{s,ss}}$$

[kg/kg]

Eq. 4-9

Eq. 4-9 can be simplified when the gas-constants of the dry air (R_{air}) and of the steam (R_v) are replaced by their values (steam and dry air are treated as ideal gas):

$$x_{s,ss} = \frac{287.1}{461.5} * \frac{p_{s,ss}}{p_{tot} - p_{s,ss}} \qquad \text{[kg/kg]}$$

$$= 0.622 * \frac{p_{s,ss}}{p_{tot} - p_{s,ss}}$$

Eq. 4-10

4.5 Humidity Ratio of Room Air

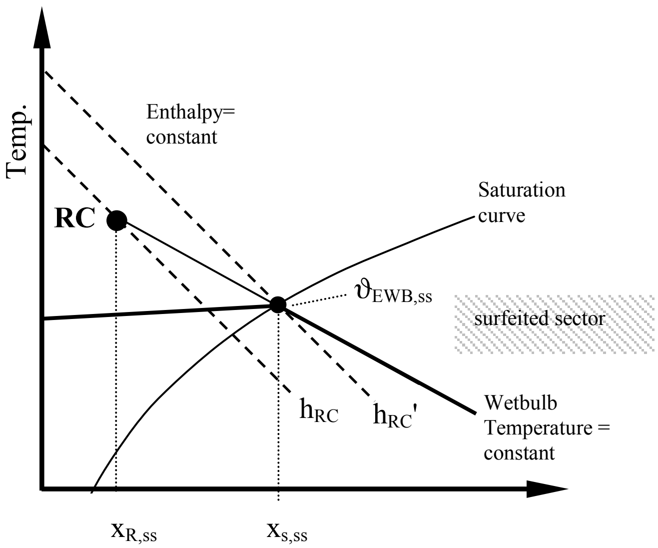


Fig. 4-2: Determination of x_R with wet bulb temperature.

Room air enthalpy is calculated from:

$$h_{RC} = \vartheta_R * cp_{air} + x_{R,ss} * (r_o + \vartheta_R * cp_V) \qquad \text{[kJ/kg*K]}$$

Eq. 4-11

In Eq. 4-11, $x_{R,ss}$ is also unknown. However, recalling the adiabatic saturation process [2, p. 6.13]:

$$h_{RC}' = h_{RC} + (x_{s,ss} - x_{R,ss}) * cp_w * \vartheta_{EWB,ss} \qquad \text{[kJ/kg*K]}$$

Eq. 4-12

h_{RC}' is easily calculated when $\vartheta_{EWB,ss}$ is known. So when h_{RC} in Eq. 4-12 is replaced by Eq. 4-11 and solved for $x_{R,ss}$, the humidity at the room condition, the following equation results:

$$term_1 = x_{s,ss} * \left[r_0 + \vartheta_{EWB,ss} * (cp_v - cp_w)\right]$$

$$x_{R,ss} = \frac{cp_{air} * (\vartheta_{EWB,ss} - \vartheta_R) + term_1}{ro + cp_v * \vartheta_R - cp_w * \vartheta_{EWB,ss}} \qquad \text{[kg/kg]}$$

Eq. 4-13

5 PERFORMANCE

The performance data are obtained by interpolation from the performance table ([1] Table 1-6e) using Eq. 4-2:

$$P_{sens,ss} = P_{sens,1} * \left(\frac{\vartheta_2 - \vartheta_{EWB,ss}}{\vartheta_2 - \vartheta_1}\right) + P_{sens,2} * \left(\frac{\vartheta_{EWB,ss} - \vartheta_1}{\vartheta_2 - \vartheta_1}\right) \qquad \text{[W]}$$

Eq. 5-1

The data which have to be taken from the table are determined by Eq. 4-5.

The interpolation is done in the same manner for $P_{tot,ss}$ and $P_{comp,ss}$.

The latent performance is the difference between $P_{tot,ss}$ and $P_{sens,ss}$:

$$P_{lat,ss} = P_{tot,ss} - P_{sens,ss} \qquad \text{[W]}$$

Eq. 5-2

6 COP CALCULATION

6.1 Part Load Ratio

The part-load ratio is defined as the ratio of the net refrigeration effect to the adjusted net total capacity of the coil:

$$PLR_{ss} = \frac{P_{sens,g} + P_{lat,g} + P_{tr,g}}{P_{tot,ss}} \qquad \text{[-]}$$

Eq. 6-1

The part-load ratio indicates also the ON time of the unit and is therefore identical to the ratio λ (see also Fig. 2-1).

6.2 COP Degradation Factor (CDF)

The COP degradation factor is calculated as a function of PLR ([1] Figure 1-3).

$$CDF_{ss} = 1 - 0.229 * (1 - PLR_{ss}) \qquad 0 \le PLR_{ss} \le 1 \qquad [-]$$

Eq. 6-2

6.3 COP

The coefficient of performance is corrected using the CDF in order to take the part load into consideration.

$$COP_{ss} = CDF_{ss} * \frac{P_{tot,ss}}{(P_{comp,ss} + P_{fan,id} + P_{fan,od})} \qquad [-]$$

Eq. 6-3

7 ENERGY CACULATION

The unit performances at the given operating point (see also subsection 5) are now used to calculate the energy consumption.

7.1 Effective ON Ratio

Due to part load operation the unit has to run longer to compensate for the start-up phase. This effect is considered with CDF and results in a higher ON ratio.

$$\lambda_{eff} = \frac{PLR_{ss}}{CDF_{ss}} \qquad [-]$$

Eq. 7-1

7.2 Cooling Energy Consumption

Compressor and both fans cycle ON/OFF simultaneously so that:

$$Q_{comp} = P_{comp,ss} * \lambda_{eff} * t_{per} \qquad [kWh]$$

Eq. 7-2

$$Q_{fan,id} = P_{fan,id} * \lambda_{eff} * t_{per} \qquad [kWh]$$

Eq. 7-3

$$Q_{fan,od} = P_{fan,od} * \lambda_{eff} * t_{per} \qquad [kWh]$$

Eq. 7-4

7.3 Zone Load

Due to additional equipment start up time when PLR < 1, the effective unit performances are lower than calculated in subsection 5:

$$P_{eff} = P * CDF \qquad [W]$$

Eq. 7-5

therefore the runtime ratio λ_{eff} has to be used for the calculation of the energy removed from the zone (excluding fan heat):

$$Q = P_{eff} * \lambda_{eff} * t_{per} \qquad [kWh]$$

Eq. 7-6

P_{eff} in Eq. 7-6 can be replaced by Eq. 7-5. This leads to the following equation for the zone load:

$$Q = P * CDF * \frac{PLR}{CDF} * t_{per} \Rightarrow$$

$$Q = P * PLR * t_{per} \qquad [kWh]$$

Eq. 7-7

$$Q_{tot} = P_{tot,ss} * PLR * t_{per} \qquad [kWh]$$

Eq. 7-8

$$Q_{sens} = P_{sens,ss} * PLR * t_{per} \qquad [kWh]$$

Eq. 7-9

$$Q_{lat} = Q_{tot} - Q_{sens} \qquad [kWh]$$

Eq. 7-10

7.4 Evaporator Coil Load

The indoor fan heat has to be considered to obtain the evaporator coil load:

$$Q_{tot,ev} = Q_{tot} + Q_{fan,id} \qquad [kWh]$$

Eq. 7-11

$$Q_{sens,ev} = Q_{sens} + Q_{fan,id} \qquad [kWh]$$

Eq. 7-12

$$Q_{lat,ev} = Q_{lat} \qquad [kWh]$$

Eq. 7-13

8 CALCULATION EXAMPLE

The calculation for the case E170 is presented.

$P_{sens,g}$	[W]	2100
$P_{lat,g}$	[W]	1100
$\vartheta_{setpoint}$ (=ϑ_R)	[°C]	22.2
ϑ_{ODB}	[°C]	29.4

8.1 Entering Wet Bulb Temperature at Steady State

The following performance data are used for the calculation:

ODB (°C)	EWB (°C)	Net total Capacity (kW)	Net sensible Capacity at EDB **22.2**	Net latent capacity at EDB **22.2**	compressor power
	17.2	7780	5260	2520	1660
29.4	19.4	8420	4110	4310	1710

index 1

index 2

Table 8-1

$$P_{tr,g} = 1.713 * (29.4 - 22.2) = 12.334$$ [W]

$$r = \frac{2100 + 12.334}{1100} = 1.920$$ [-]

$$\vartheta_{EWB,ss} = \frac{\begin{bmatrix} 5260 * 19.4 - 4110 * 17.2 + \\ 1.920 * (4310 * 17.2 - 2520 * 19.4) \end{bmatrix}}{5260 - 4110 + 1.920 * (4310 - 2520)}$$
$$= 17.402$$

[°C]

$17.2 < 17.402 < 19.4 \Rightarrow$ the correct interval has been chosen.

8.2 Saturation Pressure and Humidity

$$p_{s,ss} = e^{\left(-\frac{5800.2206}{17.402+273.15} - 11.89458198 - 0.048640239*17.402 + 0.000041764768*(17.402+273.15)^2 - 0.14452093*10^{-7}*(17.402+273.15)^3 + 6.5459673*\ln(17.402+273.15)\right)}$$
$$= 1987.877$$

[Pa]

$$x_{s,ss} = 0.622 * \frac{1987.877}{101325 - 1987.877} = 0.012447$$

[kg/kg]

8.3 Humidity Ratio of Room Air

$$term_1 = 0.012447 * [2501.6 + 17.402 * (1.861 - 4.187)] = 30.634$$

$$x_{R,ss} = \frac{1.006 * (17.402 - 22.2) + 30.634}{2501.6 + 1.861 * 22.2 - 4.187 * 17.402}$$
$$= 0.010448$$

[kg/kg]

8.4 Performance

With the previously determined room condition the unit has the following performances:

$$P_{tot,ss} = 7780 * \left(\frac{19.4 - 17.402}{19.4 - 17.2}\right) + 8420 * \left(\frac{17.402 - 17.2}{19.4 - 17.2}\right)$$ [W]
$$= 7838.713$$

$$P_{sens,ss} = 5260 * \left(\frac{19.4 - 17.402}{19.4 - 17.2}\right) + 4110 * \left(\frac{17.402 - 17.2}{19.4 - 17.2}\right)$$ [W]
$$= 5154.503$$

$$P_{lat,ss} = 7838.713 - 5154.501 = 2684.212$$ [W]

$$P_{comp,ss} = 1660 * \left(\frac{19.4 - 17.402}{19.4 - 17.2}\right) + 1710 * \left(\frac{17.402 - 17.2}{19.4 - 17.2}\right) \quad \text{[W]}$$
$$= 1664.587$$

8.5 COP

$$PLR_{ss} = \frac{2100 + 1100 + 12.334}{7838.713} = 0.410 \quad \text{[-]}$$

$$CDF_{ss} = 1 - 0.229 * (1 - 0.410) = 0.865 \quad \text{[-]}$$

$$COP_{ss} = 0.865 * \frac{7838.713}{(1664.587 + 230 + 108)} = 3.385 \quad \text{[-]}$$

8.6 Energy Calculation

Simulation period: February
Length: t_{per} = 28*24 = 672 [h]

$$\lambda_{eff} = \frac{0.410}{0.865} = 0.474 \quad \text{[-]}$$

Electricity

$$Q_{comp} = 1664.587 * 0.474 * 672 / 1000 \quad \text{[kWh]}$$
$$= \mathbf{530.046}$$

$$Q_{fan,id} = 230 * 0.474 * 672 / 1000 \quad \text{[kWh]}$$
$$= \mathbf{73.238}$$

$$Q_{fan,od} = 108 * 0.474 * 672 / 1000 \quad \text{[kWh]}$$
$$= \mathbf{34.390}$$

Zone

$$Q_{tot} = 7838.713 * 0.410 * 672 / 1000 \quad \text{[kWh]}$$
$$= \mathbf{2158.688}$$

$$Q_{sens} = 5154.503 * 0.410 * 672 / 1000 \quad \text{[kWh]}$$
$$= \mathbf{1419.488}$$

$$Q_{lat} = Q_{tot} - Q_{sens} \quad \text{[kWh]}$$
$$= \mathbf{739.200}$$

Evaporator

$$Q_{tot,ev} = 2158.688 + 73.238 \quad \text{[kWh]}$$
$$= \mathbf{2231.926}$$

$$Q_{sens,ev} = 1419.488 + 73.238 \quad \text{[kWh]}$$
$$= \mathbf{1492.726}$$

$$Q_{lat,\,ev} = \mathbf{739.200} \quad \text{[kWh]}$$

NUMERICAL SOLUTION

9 FLOWCHART

The principle of the calculation algorithm is shown in the following flowchart. Each rectangular box represents a calculation module that is explained later in a separate chapter.

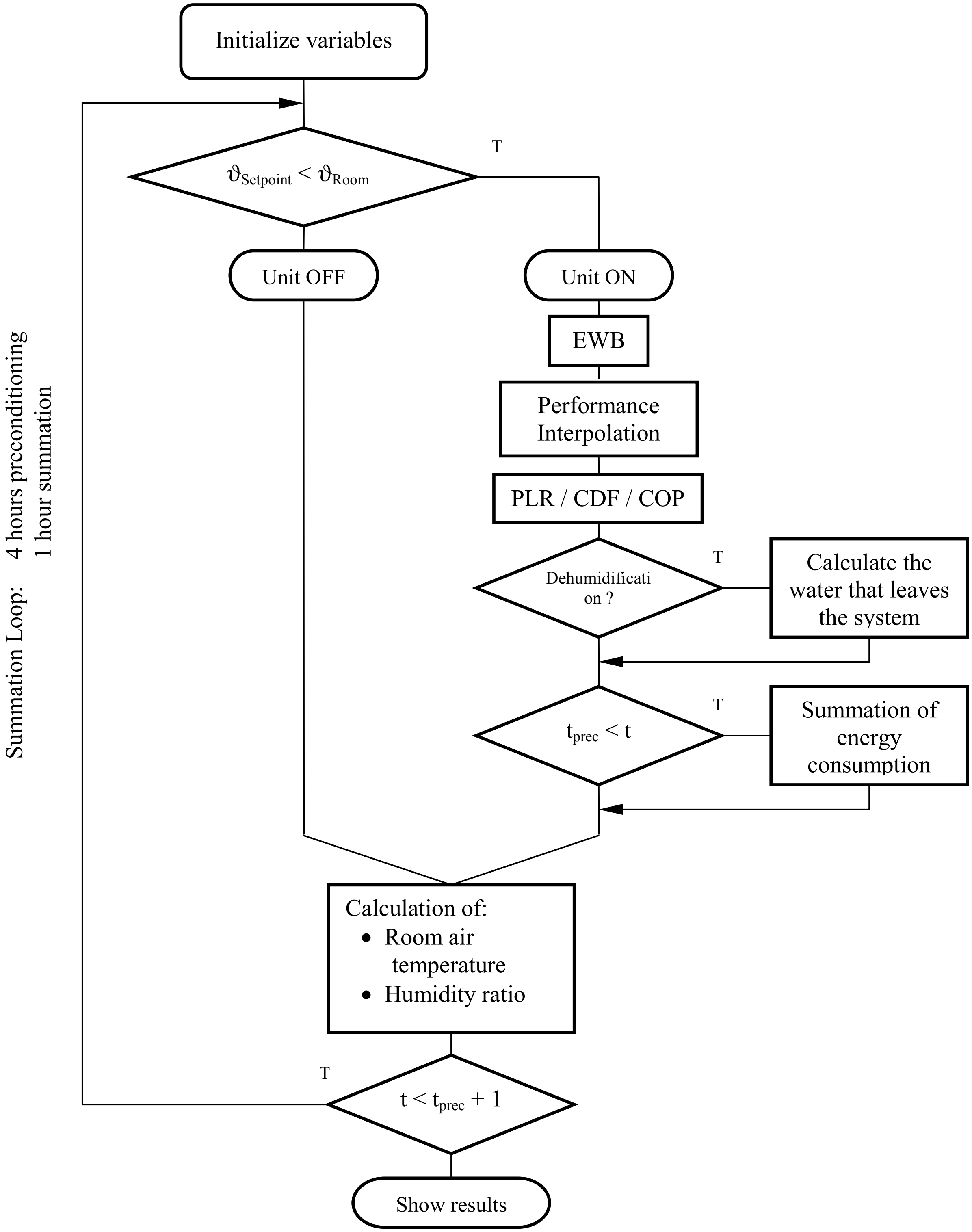

Summation Loop: 4 hours preconditioning
1 hour summation
Initialize variables
$\vartheta_{Setpoint} < \vartheta_{Room}$
T
Unit OFF
Unit ON
EWB
Performance Interpolation
PLR / CDF / COP
Dehumidification ?
T
Calculate the water that leaves the system
$t_{prec} < t$
T
Summation of energy consumption
Calculation of:
• Room air temperature
• Humidity ratio
T
$t < t_{prec} + 1$
Show results

Fig. 9-1

10 WET COIL

10.1 Entering Wet Bulb Temperature

The entering wet bulb temperature is calculated as a function of the current room-air condition (RC: temperature and moisture content). Because an explicit solution is not known, an iterative process is used.

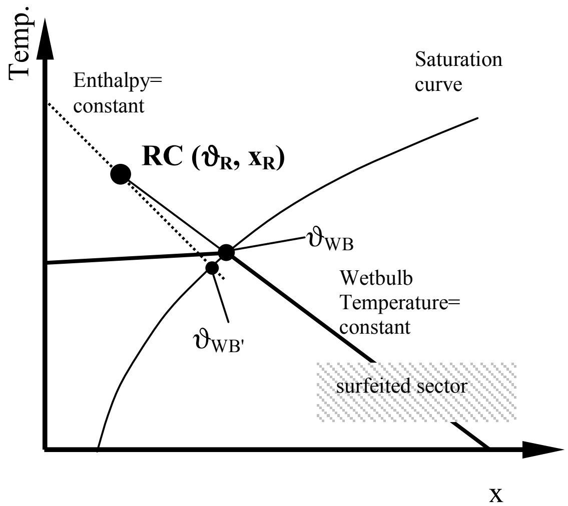


Fig. 10-1: definition of the wet bulb temperature. It is defined as the point of intersection between the temperature constant curve (which intersects the room air condition) and the saturation curve. A first approximation can be made by using the room-enthalpy.

First the temperature at the saturation curve with enthalpy h_R = constant is determined ($\vartheta_{WB'}$). Then a correction is made to account for the deviation of the enthalpy and temperature curve.

Determination of $\vartheta_{WB'}$

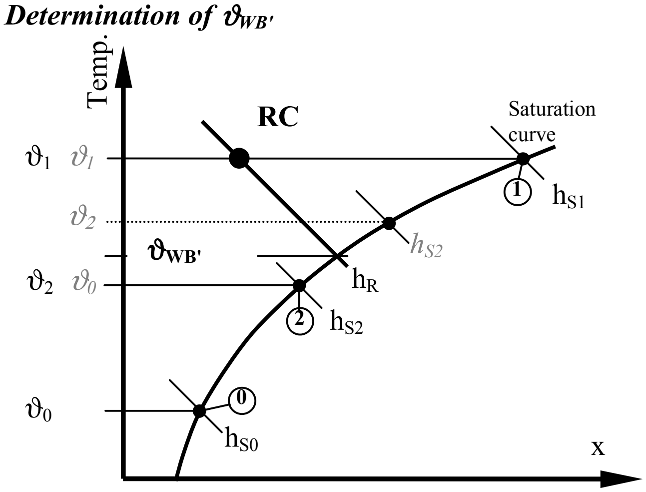


Fig. 10-2

The following inputs must be made for the calculation of the wet bulb temperature. The initial values are needed for the first step of iteration.

R Room air condition

	ϑ_R	room air temperature
	x_R	humidity ratio

Initial values

ϑ_0	lower boundary	$\vartheta_0 = 0°C$
ϑ_1	upper boundary	$\vartheta_1 = \vartheta_R$
ϑ_2	mean value	$\vartheta_2 = 0.5*(\vartheta_0 + \vartheta_1)$

Enthalpy h_R

$$h_R = \vartheta_R * cp_{air} + X_R * (r_o + \vartheta_R * cp_v) \quad \text{[kJ/kg*K]}$$

Eq. 10-1

Enthalpy h_i (i=0 ... 2)
is calculated at the saturation curve with the temperatures ϑ_0, ϑ_1, ϑ_2. These enthalpies are calculated first with the initial temperatures.

The saturation pressure at the temperature ϑ_i is calculated with the ASHRAE formula (see also subsection 4.3).

$$p_{si} = e^{\left(-\frac{5800.2206}{\vartheta_i + 273.15} - 11.89458198 - 0.048640239 * \vartheta_i + 0.000041764768 * (\vartheta_i + 273.15)^2 - 0.14452093 * 10^{-7} * (\vartheta_i + 273.15)^3 + 6.5459673 * \ln(\vartheta_i + 273.15) \right)}$$

[Pa]

Eq. 10-2

The moisture content is then calculated with p_s and the total pressure p (see also subsection 4.4):

$$x_{si} = 0.622 * \frac{p_{si}}{p_{tot} - p_{si}} \quad \text{[kg/kg]}$$

Eq. 10-3

The enthalpy of point 0, 1 and 2 can now be calculated:

$$h_{si} = \vartheta_i * cp_{air} + X_{si} * (r_o + \vartheta_i * cp_v) \quad \text{[kJ/kg*K]}$$

Eq. 10-4

h_{s2} is compared with hR. If Eq. 10-5 is met,

$$\left| h_R - h_{s2} \right| < \varepsilon \quad \text{[kJ/kg*K]}$$

Eq. 10-5

the wet bulb temperature $\vartheta_{WB'}$ is determined:

$$\vartheta_{WB'} = \vartheta_2 \quad \text{[°C]}$$

Eq. 10-6

If Eq. 10-5 is false, the calculation is repeated. ϑ_0, ϑ_1, ϑ_2 are then determined again, in the way that h_R is between h_{s0} and h_{s1} (see Fig. 10-2):

$$\text{if } h_{s0} < h_R \wedge h_R < h_{s2} \Rightarrow \vartheta_0 = \vartheta_0 \wedge \vartheta_1 = \vartheta_2$$

$$\text{if } h_{s2} < h_R \wedge h_R < h_{s1} \Rightarrow \vartheta_0 = \vartheta_2 \wedge \vartheta_1 = \vartheta_1$$

$$\vartheta_2 = 0.5 * (\vartheta_0 + \vartheta_1)$$

Eq. 10-7

The calculation is repeated (a so called technique of nested intervals) until Eq. 10-5 is fulfilled. Eq. 10-6 defines then the wet bulb temperature of the room condition RC ($\vartheta_{WB} = \vartheta_{WB'}$).

Determination of ϑ_{WB}
A new (imaginary) room condition RC' is calculated. It is defined with the same air temperature as RC but the higher enthalpy (Eq. 10-8). The wet bulb temperature is then calculated with the same algorithm as $\vartheta_{WB'}$.

$$h_{R'} = h_R + (x_{S2} - x_R) * cp_W * \vartheta_{WB'} \quad \text{[kJ/kg*K]}$$

Eq. 10-8

Calculation of enthalpy $h_{R'}$: Instead of x_S the value x_{S2} which is already known from the $\vartheta_{WB'}$-calculation is taken. The resulting error is negligible because the difference between ϑ_{WB} and $\vartheta_{WB'}$ is small.

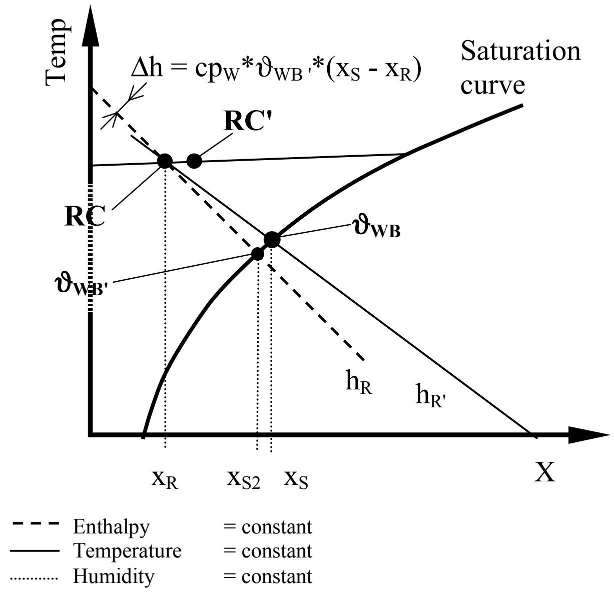


Fig. 10-3: correction to account for that the wet bulb temperature is calculated with intersection of temp = const. and not enthalpy = const.

10.2 Performance Interpolation

The performance of the cooling unit is calculated as a function of the entering wet-bulb and dry-bulb temperatures. The appropriate performance table for the given case is used as input. The sensible capacity is first interpolated with the entering wet bulb temperature and then with the entering dry bulb temperature.

The adjusted net capacities ([1] Table 1-6e) are used.

ODB [°C]	EWB [°C]	Net Total Capacity [kW]	Net Sensible Capacity at EDB **22.2** [kW]	Compressor Power [kW]	Apparatus Dew Point [°C]
	15.0	7.19	6.31	1.62	8.9
	17.2	7.78	5.26	1.66	11.1
29.4	19.4	8.42	4.11	1.71	13.4
	21.7	9.06	2.97	1.76	15.8

Table 10-1

First the interval which contains the given wet bulb temperature is determined. This is done automatically by the calculation routine. Also the dry-bulb temperature interval is determined by the routine. The unit capacities are then calculated by using a linear interpolation algorithm:

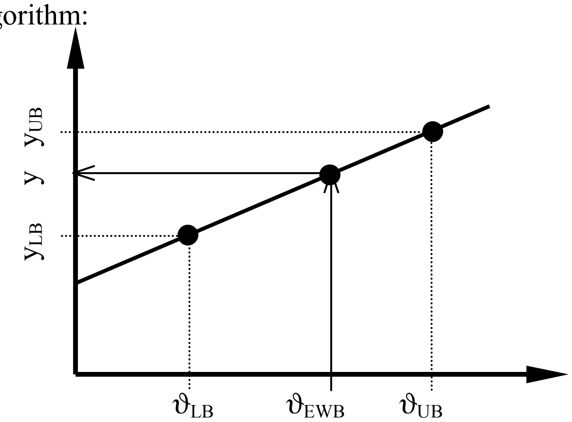


Fig. 10-4: Interpolation of performance data. y stands for the variable that has to be interpolated (net sensible capacity, net total capacity, compressor power)

Linear interpolation using the upper and lower boundary of the appropriate interval:

$$y = \frac{(y_{UB} - y_{LB}) * (\vartheta_{EWB} - \vartheta_{LB})}{(\vartheta_{UB} - \vartheta_{LB})} + y_{LB} \qquad \text{[W]}$$

Eq. 10-9:

linear interpolation in the performance table

Example for ODB = 29.4, EDB = 22.2 and EWB = 16 °C

Interpolation of the net total capacity

ϑ_{LB}	=	15.0	°C
ϑ_{UB}	=	17.2	°C
y_{LB}	=	7190	W
y_{UB}	=	7780	W

$$y = \frac{(7780 - 7190) * (16 - 15)}{(17.2 - 15)} + 7190 = 7460 \text{ W}$$

Extrapolation
If the given entering wet bulb temperature is below 15.0 or above 21.7 °C (the given range in the performance list), the performance data will be extrapolated:

ϑ_{EWB} < 15 °C
For the upper and lower bound temperatures the values of the first interval can be inserted in Eq. 10-9. This leads to:

$$y = \frac{(y_{UB} - y_{LB}) * (\vartheta_{EWB} - 15)}{2.2} + y_{LB} \qquad \text{[W]}$$

Eq. 10-10

ϑ_{EWB} > 21.7 °C

$$y = \frac{(y_{UB} - y_{LB}) * (\vartheta_{EWB} - 19.4)}{2.3} + y_{LB} \qquad \text{[W]}$$

Eq. 10-11

The following variables are interpolated:

$P_{(t),\,comp}$	= f (EWB)	compressor power
$P_{(t),\,tot}$	= f (EWB)	total unit capacity
$P_{(t),\,sens}$	= f (EWB, EDB)	sensible unit capacity[2]

10.3 COP Calculation

The COP is calculated at every timestep when the unit is turned on. This COP is then multiplied with the COP degradation factor to take the part load ratio into consideration.

$$COP'_{(t)} = \frac{P_{(t),tot}}{(P_{(t),comp} + P_{fan,id} + P_{fan,od})} \qquad \text{[-]}$$

Eq. 10-12

[2] For the interpolation with both, EWB and EDB temperatures, Eq. 10-9 is used.

The part load ratio of the momentary timestep is the ratio of the net refrigeration effect to the adjusted net total capacity of the coil. At a steady state of the room temperature the net refrigeration effect is equal to the total load (see Eq. 10-34 for the calculation of $P_{tr,g}$).

$$PLR_{(t)} = \frac{P_{sens,g} + P_{lat,g} + P_{tr,g}}{P_{(t),tot}} \quad [-]$$

Eq. 10-13

The COP degradation factor is calculated as a function of PLR ([1] Figure 1-3, p. 22)

$$CDF_{(t)} = 1 - 0.229 * (1 - PLR_{(t)}) \qquad 0 \le PLR_{(t)} \le 1 \quad [-]$$

Eq. 10-14

The effective COP can then be calculated:

$$COP_{(t)} = COP'_{(t)} * CDF_{(t)} \quad [-]$$

Eq.10-15

Runtime of the unit: this is the time during which the unit is switched on in the summation-period.

$$t_{ON,sum} = \sum_{t=t_{prec}}^{t=t_{end}} t_{ON(t)} \quad [s]$$

Eq. 10-16

The mean COP during the runtime is:

$$COP_{mean} = \frac{\sum_{t=t_{prec}}^{t_{end}} COP_{(t)} * dt}{t_{ON,sum}} \quad [-]$$

Eq. 10-17

10.4 Effective Performance

As the start-up phase of the unit causes a decrease of the average performance this effect has to be taken into account using the CDF factor:

$$P_{(t),tot,eff} = P_{(t),tot} * CDF \quad [w]$$

Eq. 10-18

$$P_{(t),sens,eff} = P_{(t),sens} * CDF \quad [w]$$

Eq. 10-19

10.4.1 Dehumidification

Dehumidification occurs only when the following equation is met:

$$P_{(t),sens,eff} < P_{(t),tot,eff} \quad [W]$$

Eq. 10-20

$P_{(t),\,tot}$ is defined as the sum of latent and sensible performance:

$$P_{(t),tot,eff} = P_{(t),sens,eff} + P_{(t),lat,eff} \quad [W]$$

Eq. 10-21

The removed water from the system is then calculated as follows:

$$m_{w,rem} = \frac{\left| P_{(t),tot,eff} - P_{(t),sens,eff} \right| * dt}{r_{(\vartheta)}} \quad [kg]$$

Eq. 10-22

The evaporation energy is a function of temperature:

$$r_{(\vartheta)} = ro + \vartheta_R * (cp_V - cp_W) \quad [kg]$$

Eq. 10-23

$r_{(\vartheta)}$ in Eq. 10-22 replaced by Eq. 10-23 leads to the following equation:

$$m_{w,rem} = \frac{\left| P_{(t),tot,eff} - P_{(t),sens,eff} \right| * dt}{ro + \vartheta_R * (cp_V - cp_W)} \quad [kg]$$

Eq. 10-24

10.4.2 Unit Is Turned Off

When the unit is turned off, the performance data are set to zero:

$$P_{(t),tot,eff} = 0$$
$$P_{sens,eff} = 0$$
$$m_{W,rem} = 0$$
$$P_{fan,id} = 0$$
$$P_{fan,od} = 0$$
$$P_{comp} = 0$$

10.5 Summation of Energy Consumption

The energy is summed after the preconditioning-period. A period of one hour is calculated. The energy is then multiplied with the number of hours for the month February (NBH_{FEB} = 672 [h])

The start time for the summation is therefore $t = t_{prec}$ and the end time:

$$t_{end} = t_{prec} + 3600 \qquad [s]$$

Eq. 10-25

total (sensible + latent) energy removed by the system:

$$Q_{tot} = NBH * \sum_{t=t_{prec}}^{t_{end}} P_{(t),tot,eff} * dt \qquad [J]$$

Eq. 10-26

The sensible energy removed by the system gives the following equation. This energy is the same as the sensible envelope load.

$$Q_{sens} = NBH * \sum_{t=t_{prec}}^{t_{end}} P_{(t),sens,eff} * dt \qquad [J]$$

Eq. 10-27

The latent energy is obtained by deducting the sensible energy from the total:

$$Q_{lat} = NBH * \sum_{t=t_{prec}}^{t_{end}} (P_{(t),tot,eff} - P_{(t),sens,eff}) * dt \qquad [J]$$
$$= Q_{tot} - Q_{sens}$$

Eq. 10-28

Fan energy consumption

Indoor fan:

$$Q_{fan,\ id} = NBH * \sum_{t=t_{prec}}^{t_{end}} P_{fan,id} * dt \qquad [J]$$

Eq. 10-29

The actual indoor fan power is for all cases $P_{fan,id} = 230$ [W]

outdoor fan:

$$Q_{fan,\ od} = NBH * \sum_{t=t_{prec}}^{t_{end}} P_{fan,od} * dt \qquad [J]$$

Eq. 10-30

The outdoor fan power is for all cases $P_{fan,\ od}$ = 108 [W]

Compressor power

$$Q_{comp} = NBH * \sum_{t=t_{prec}}^{t_{end}} P_{comp} * dt \qquad [J]$$

Eq. 10-31

To get the energy that is removed by the evaporator the energy of the indoor fan has to be added[3]:

total energy removed by the evaporator:

$$Q_{tot,ev} = Q_{tot} + Q_{fan,\ id} \qquad [J]$$

Eq. 10-32

sensible energy removed by the evaporator:

$$Q_{sens,ev} = Q_{sens} + Q_{fan,\ id} \qquad [J]$$

Eq. 10-33

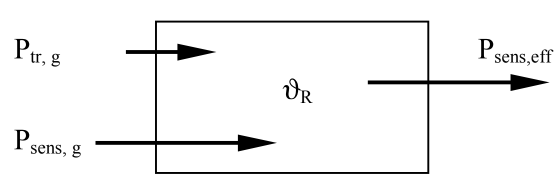
The apparatus dew point is not used for the calculation.

10.6 Calculation of Room Air Condition

At the end of every timestep the room air condition (defined with air temperature and humidity ratio) is calculated. In the following timestep the air temperature is used to decide if the unit cycles on. The humidity ratio affects the performance of the unit.

10.6.1 Air Temperature

The air temperature is calculated with a heat balance at the end of every timestep. Only the room air can store thermal energy.



[3] Q_{tot} and Q_{sens} are calculated with adjusted net capacities, which exclude the indoor fan energy.

Transmitted energy through the building envelope is calculated using the UA value:

$$P_{tr,g} = UA * (\vartheta_{ODB} - \vartheta_R) \quad \text{[W]}$$

Eq. 10-34

Added energy to the room from t = 0 to t:

$$Q_{(t),add} = \sum_{t=1}^{t} (P_{sens,g} + P_{tr,g} - P_{sens,eff}) * dt \quad \text{[J]}$$

Eq. 10-35

The resulting air temperature at the moment "t" is:

$$\vartheta_{(t),R} = \vartheta_{setpoint} + \frac{Q_{(t),add}}{V_R * \rho_{air} * (cp_{air} + X_R * cp_v)} \quad \text{[°C]}$$

Eq. 10-36

Mean air temperature at the moment "t":

$$\vartheta_{(t)R,mean} = \frac{\sum_{t=t_{prec}}^{t_{end}} \vartheta_{(t)R} * dt}{t - t_{prec}} \quad \text{[°C]}$$

Eq. 10-37

10.6.2 Humidity Ratio

The humidity ratio is set to a initial value at the beginning of the calculation. Then at every timestep the added and removed water is determined. The total water in the air is summed up and the humidity ratio of the present timestep is calculated.

$$m_{(t)w,add} = \frac{P_{lat,g} * dt}{ro + \vartheta_R * (cp_v - cp_w)} \quad \text{[kg]}$$

Eq. 10-38

The effective added moisture is received when the removed water ($m_{w,\ rem}$ from Eq. 10-22) is subtracted from $m_{(t)w,\ add}$. The moisture content is summed up in variable $m_{(t)w,\ tot}$. The moisture that is included at the beginning of the calculation is determined by using the initial humidity ratio.

$$m_{(t)w,tot} = x_{R,init} * V_R * \rho_{air} + \sum_{t=1}^{t} (m_{w,add} - m_{w,rem}) * dt \quad \text{[kg]}$$

Eq. 10-39

According to the test specification the initial humidity ratio ($x_{R,\ init}$) is set equal to the outdoor humidity ratio. The required preconditioning periods are listed in Table 10-2:

Case	$X_{R,\ init}$ [kg/kg]	$t_{precond}$ [h]
E100	0.01	4
E110	0.01	4
E120	0.01	3
E130	0.01	28
E140	0.01	36
E150	0.01	3
E160	0.01	3
E165	0.01	3
E170	0.01	4
E18	0.01	3
E185	0.01	3
E190	0.01	9
E195	0.01	9
E200	0.01	6

Table 10-2

The resulting humidity ratio is obtained by dividing the mass of the water by the mass of the air:

$$x_{(t)R} = \frac{m_{(t)w,tot}}{V_R * \rho_{air}} \quad \text{[kg/kg]}$$

Eq. 10-40

Mean humidity ratio at the moment "t":

$$x_{(t)R,mean} = \frac{\sum_{t=t_{prec}}^{t_{end}} x_{(t)R} * dt}{t - t_{prec}} \quad \text{[kg/kg]}$$

Eq.10-41

10.7 Envelope Load

The envelope loads are summed up at the end of every timestep:

Sensible envelope load

$$Q_{sens,env} = \sum_{t=t_{prec}}^{t} (P_{sens,g} + P_{tr,g}) * dt \quad \text{[J]}$$

Eq. 10-42

Latent envelope load

$$Q_{sens,lat} = \sum_{t=t_{prec}}^{t} P_{lat,g} * dt \qquad [J]$$

Eq. 10-43

11 DRY COIL TESTS

The calculation for dry coil test cases is done with the same algorithm as for wet coils. The initial humidity ratio of 0.01 [kg/kg] leads to a room condition where a dehumidification occurs until a steady state is reached (that is where $P_{tot} = P_{sens}$).

12 CALCULATION EXAMPLE

In the following a calculation example for the case E170 is presented. Because the whole process is an iterative procedure, one ON and one OFF timestep are described.

12.1 ON Timestep

The step at time t = 14,600 s is presented. t = 14,600 includes the preconditioning period, that means it is t = 200 s of the summation loop. From the previous timestep the following values of the variables are known:

$\vartheta_{R,\ prev}$	22.204	[°C]
$x_{R,\ prev}$	0.01045	[kg/kg]

Entering wet bulb temperature:
termination criterion $\varepsilon = 0.001$

Calculation of $\vartheta_{WB'}$
Initial values

ϑ_0	=	0.000	[°C]
ϑ_1	=	22.204	[°C]
ϑ_2	=	$0.5*(\vartheta_0 + \vartheta_1) = 11.102$	[°C]

$$h_R = 22.204 * 1.006 + 0.01045 * (2501 + 22.204 * 1.861)$$

= **48.90** [kJ/kg]

Point 0

$$p_{s0} = e^{\left(-\frac{5800.2206}{0+273.15} - 11.89458198 - 0.048640239*0 + 0.000041764768*(0+273.15)^2 - 0.14452093*10^{-7}*(0+273.15)^3 + 6.5459673*\ln(0+273.15)\right)}$$

$$= 611.21 \qquad [Pa]$$

$$x_{s0} = 0.622 * \frac{611.21}{101325 - 611.21} = 0.00378 \qquad [kg/kg]$$

$$h_{s0} = 0 * 1.006 + 0.00378 * (2501 + 0 * 1.861) \qquad [kJ/kg]$$

= **9.44**

Point 1

$$p_{s1} = e^{\left(-\frac{5800.2206}{22.204+273.15} - 11.89458198 - 0.048640239*22.204 + 0.000041764768*(22.204+273.15)^2 - 0.14452093*10^{-7}*(22.204+273.15)^3 + 6.5459673*\ln(22.204+273.15)\right)}$$

$$= 2676.26$$

[Pa]

$$x_{s1} = 0.622 * \frac{2677.86}{101325 - 2677.86} = 0.01688 \qquad [kg/kg]$$

$$h_{s1} = 22.204 * 1.006 + 0.01688 * (2501 + 22.204 * 1.861)$$

[kJ/kg]

= **65.27**

Point 2

$$p_{s2} = e^{\left(-\frac{5800.2206}{11.102+273.15} - 11.89458198 - 0.048640239*11.102 + 0.000041764768*(11.102+273.15)^2 - 0.14452093*10^{-7}*(11.102+273.15)^3 + 6.5459673*\ln(11.102+273.15)\right)}$$

$$= 1321.67$$

$$x_{s2} = 0.622 * \frac{1321.67}{101325 - 1321.67} = 0.00822 \qquad [kg/kg]$$

$$h_{s2} = 11.102 * 1.006 + 0.00822 * (2501 + 11.102 * 1.8161)$$

[kJ/kg]

= **31.90**

$\left|48.90 - 31.9\right| < 0.001$: false $\Rightarrow$ the calculation has to be repeated until this equation is met.

Determination of new points:

$h_{s2} < h_R$ and $h_R < h_{s1} \Rightarrow$

$\vartheta_0 = 11.102$

$\vartheta_1 = 22.204$

$\vartheta_2 = 0.5 * (11.102 + 22.204) = 16.653$

The calculation is now repeated with these new points.

After 14 iteration steps: $\vartheta_{WB'} = 17.354$ [°C]

$x_{s2} = 0.01241$ [kg/kg]

Calculation of ϑ_{WB}

$h_{R'} = 48.90 + (0.01241 - 0.01045) * 4.187 * 17.354$

$= 49.04$

[kJ/kg]

The calculation of $\vartheta_{WB'}$ with $h_{R'}$ is repeated until $\left|49.04 - h_{s2}\right| < 0.001$

After 15 further iteration steps the wet bulb temperature is determined:

ϑ_{WB} = 17.4008[°C]

Performance Interpolation

Interpolation of performance data with the entering wet bulb temperature.

ODB [°C]	EWB [°C]	Net total Capacity [kW]	Net sensible capacity at EDB [°C]		Compressor power [kW]
			22.2 [kW]	**23.3** [kW]	
29.4	15.0	7.19	6.31	6.87	1.62
	17.2	7.78	5.26	5.81	1.66
	19.4	8.42	4.11	4.67	1.71
	21.7	9.06	2.97	3.50	1.76

lower bound

upper bound

Table 12-1

Net total capacity

$$P_{(14600)tot} = \frac{(8420 - 7780) * (17.4008 - 17.2)}{(19.4 - 17.2)} + 7780$$

$= 7838.42$ [W]

Compressor power

$$P_{(14600)comp} = \frac{(1710 - 1660) * (17.4008 - 17.2)}{(19.4 - 17.2)} + 1660$$

$= 1664.56$ [W]

Net sensible capacity

First, the net sensible capacities for the entering dry bulb temperatures of 22.2 and 23.3°C have to be determined by interpolating with EWB=17.4008°C. This gives the following results:

ODB [°C]	EWB [°C]	Net sensible capacity at EDB [°C]	
		22.2 [kW]	**23.3** [kW]
29.4	19.4	5.155	5.706

Interpolating with EDB=$\vartheta_{R, prev}$=22.204 °C:

$$P_{(14600)sens} = \frac{(5706 - 5155)*(22.204 - 22.2)}{(23.3 - 22.2)} + 5155$$
$$= 5157.00 \ [W]$$

PLR, CDF, and COP

$$P_{tr,g} = 1.713 * (29.4 - 22.2) = 12.3336 \quad [W]$$

$$PLR_{(14600)} = \frac{2100 + 1100 + 12.3336}{7838.42} = 0.4098 \quad [-]$$

$$CDF_{(14600)} = 1 - 0.229 * (1 - 0.4098) = 0.865 \quad [-]$$

$$COP_{(14600)} = 0.865 * \frac{7838.42}{(1664.56 + 230 + 108)} = 3.385 \quad [-]$$

Mean COP:

$$t_{ON,sum,prev} = 94 \quad [s]$$
$$t_{(14601)ON,sum} = 94 + 1 = 95 \quad [s]$$

$$COP_{sum,prev} = 318.21 \quad [-]$$
$$COP_{(14601)sum} = 318.21 + 3.385 = 321.60 \quad [-]$$

$$COP_{mean} = \frac{321.60}{95} = \mathbf{3.385} \quad [-]$$

Effective Performance

$$P_{(14600)tot,eff} = 7838.42 * 0.865 = 6779.048 \quad [W]$$

$$P_{(14600)sens,eff} = 5157.00 * 0.865 = 4460.81 \quad [W]$$

Dehumidification
Water that is removed from system this timestep:

$$m_{w,rem} = \frac{|6779.048 - 4460.81| * 1}{2501 * 10^3 + 22.204 * (1861 - 4187)} \quad [kg]$$
$$= 0.0009465$$

Summation of energy consumption
The momentary energy is added to the already consumed energy during the summation period:

Total energy removed by the system:

$$Q_{tot,\ prev} = \sum_{t=14400}^{t=14599} P_{(t),tot} * dt = 637248.9 \quad [J]$$

$$Q_{(14600)tot} = 637248.9 + 6779.0 * 1 = 644028.0 \quad [J]$$

Sensible energy removed by the system:

$$Q_{sens,\ prev} = 419042.9 \quad [J]$$

$$Q_{(14600)sens} = 419042.9 + 4460.81 * 1$$
$$= 423503.7 \quad [J]$$

Latent energy removed by the system:

$$Q_{(14600)lat} = 644028.0 - 423503.7$$
$$= 220524.3 \quad [J]$$

Supply-fan energy consumption:

$$Q_{fan,id,prev} = 21620 \quad [J]$$

$$Q_{(14600)fan,id} = 21620 + 230 * 1$$
$$= 21850 \quad [J]$$

Outdoor-fan energy consumption:

$$Q_{fan,od,prev} = 10152 \quad [J]$$

$$Q_{(14600)fan,od} = 10152 + 108 * 1$$
$$= 10260 \quad [J]$$

Compressor energy consumption:

$$Q_{comp,\ prev} = 146470.9 \quad [J]$$

$$Q_{(14600)comp} = 146470.9 + 1664.6 * 1$$
$$= 158135.5 \quad [J]$$

Total energy removed by the evaporator:

$$Q_{(14600)tot,ev} = 644028.0 + 21850 = 665878 \quad [J$$

Sensible energy removed by the evaporator:

$$Q_{(14600)tot,ev} = 423503.7 + 21850 = 445353.7 \quad [J]$$

Calculation of room air condition

Air temperature:

$Q_{add,prev} = 643.7$ [J]

$Q_{(14600)add} = 643.7 + (2100 + 12.3 - 4460.8) * 1 = -1704.8$ [J]

$\vartheta_{(14600),R} = 22.2 + \frac{-1704.8}{129.6 * 1.201 * 1006}$ [°C]

$= \mathbf{22.189}$

Mean air temperature:

$\vartheta_{sum,prev} = 4417.65$ [°C*s]

$\vartheta_{(14600)R,mean} = \frac{4417.65 + 22.189 * 1}{14600 - 14400} = 22.199$ [°C]

Humidity ratio:

$m_{(t)w,add} = \frac{1100 * 1}{2501 * 10^3 + 22.189 * (1861 - 4187)} = 4.490 * 10^{-4}$ [kg]

$m_{w,tot,prev} = 1.6259$ [kg]

$m_{(14601)w,tot} = 1.6259 + (4.490 * 10^{-4} - 9.465 * 10^{-4}) = 1.6254$ [kg]

$x_{(t)R} = \frac{1.6254}{129.6 * 1.201} = \mathbf{0.0104428}$ [kg/kg]

Mean humidity ratio:

$x_{R,sum,prev} = 2.079$ [kg/kg]

$x_{(t)R,mean} = \frac{2.079 + 0.0104428 * 1}{14600 - 14400} = 0.0104449$ [kg/kg]

12.2 OFF Timestep

The step at time t = 14601 s is presented. The temperature from the previous timestep (see 12.1) $\vartheta_{(14601)R}$ = 22.193 °C is lower than the setpoint (22.2 °C). That means that the unit is turned off.

Set all performance variables to zero

$P_{(t),tot} = 0$

$P_{sens} = 0$

$m_{W,rem} = 0$

$P_{fan,id} = 0$

$P_{fan,od} = 0$

$P_{comp} = 0$

Calculation of room air condition

Air temperature:

$Q_{add,prev} = -1702.3$ [J]

$Q_{(14601)add} = -1702.3 + (2100 + 12.3 - 0) * 1 = 410.0$ [J]

$\vartheta_{(14601),R} = 22.2 + \frac{410.0}{129.6 * 1.201 * 1006}$ [°C]

$= \mathbf{22.2026}$

Mean air temperature:

$\vartheta_{sum,prev} = 4439.8$ [°C*s]

$\vartheta_{(14601)R,mean} = \frac{4439.8 + 22.2026 * 1}{14601 - 14400} = 22.1992$ [°C]

Humidity ratio:

$m_{(14601)w,add} = \frac{1100 * 1}{2501 * 10^3 + 22.2026 * (1861 - 4187)} = 4.490 * 10^{-4}$ [kg]

$m_{w,tot,prev} = 1.6254$ [kg]

$m_{(14602)w,tot} = 1.6254 + (4.490 * 10^{-4} - 0) = 1.6259$ [kg]

$x_{(14601)R} = \frac{1.6259}{129.6 * 1.201} = \mathbf{0.0104457}$ [kg/kg]

Mean humidity ratio:

$x_{R,sum,prev} = 2.0890$ [kg/kg]

$x_{(14601)R,mean} = \frac{2.0890 + 0.0104457 * 1}{14601 - 14400} = 0.0104449$ [kg/kg]

12.3 Total Energy Consumption

The total energy is got by a multiplication of the energy consumed in one hour with the number of hours in February. The units are converted from [J] to [kWh]:

Simulation period: February $\Rightarrow$ 672 hours

Cooling energy consumption

Supply-fan energy consumption:

$$Q_{fan,id} = 672 * \frac{392380}{(3.6 * 10^6)} = 73.24 \qquad \text{[kWh]}$$

Outdoor-fan energy consumption:

$$Q_{fan,od} = 672 * \frac{184248}{(3.6 * 10^6)} = 34.4 \qquad \text{[kWh]}$$

Compressor energy consumption:

$$Q_{comp} = 672 * \frac{2839784.5}{(3.6 * 10^6)} = 530.1 \qquad \text{[kWh]}$$

Evaporator coil load

Latent energy removed by the system:

$$Q_{lat} = 672 * \frac{3960330.8}{(3.6 * 10^6)} = 739.3 \qquad \text{[kWh]}$$

Sensible energy removed by the system:

$$Q_{sens} = 672 * \frac{7605099.1}{(3.6 * 10^6)} = 1419.62 \qquad \text{[kWh]}$$

Sensible energy removed by the evaporator:

$$Q_{sens,ev} = 1419.62 + 73.24 = 1492.9 \qquad \text{[kWh]}$$

Total energy removed by the evaporator:

$$Q_{tot,ev} = 1419.6 + 73.2 + 739.3 = 2232.1 \qquad \text{[kWh]}$$

Envelope loads

Sensible envelope load:

$$Q_{sens,env} = 672 * \frac{7604401}{(3.6 * 10^6)} = 1419.5 \qquad \text{[kWh]}$$

Latent envelope load:

$$Q_{lat,env} = 672 * \frac{3960000}{(3.6 * 10^6)} = 739.2 \qquad \text{[kWh]}$$

13 Remarks concerning the numerical solution

The results showed that in Cases E100 and E200, it is essential to incorporate sensitivity of sensible capacity to EDB. Without this consideration numerical problems were encountered or the algorithm did not work properly with the given initial values. This means that in Case E100 the final humidity ratio (at steady state) was dependent on the initial humidity ratio and therefore the results were not reliable.

All other cases show minor changes (< 0.5 %) in the results compared to a model that does not consider sensitivity of sensible capacity to entering dry bulb temperature.

References

[1] BESTEST Test description
J. Neymark / R. Judkoff
September 1998 edition

[2] ASHRAE 1997 Handbook of Fundamentals

2.4 Analytical Solution Results Tables

(For abbreviations and acronyms used here, see Section 2.6.)

COP: Mean, and (Max-Mean)/ Mean

	Mean COP				**(Max - Min)/ Mean COP**			
	TUD	HTAL1	HTAL2	(Max-Min) / Mean	TUD	HTAL1	HTAL2	(Max-Min) / Mean
E100	2.39	2.39	2.39	0.05%	0.000	0.000	0.000	
E110	3.38	3.38	3.38	0.02%	0.000	0.000	0.000	
E120	3.59	3.59	3.59	0.10%	0.000	0.000	0.000	
E130	1.89	1.91	1.91	1.08%	0.000	0.000	0.000	
E140	2.75	2.77	2.77	0.72%	0.000	0.000	0.000	
E150	3.63	3.63	3.63	0.07%	0.000	0.000	0.001	
E160	3.83	3.84	3.84	0.18%	0.000	0.000	0.000	
E165	2.93	2.93	2.93	0.02%	0.000	0.000	0.000	
E170	3.37	3.39	3.39	0.67%	0.000	0.000	0.000	
E180	4.04	4.04	4.04	0.06%	0.000	0.000	0.000	
E185	2.85	2.85	2.85	0.15%	0.000	0.000	0.000	
E190	3.39	3.41	3.41	0.69%	0.000	0.000	0.000	
E195	2.29	2.31	2.31	0.68%	0.000	0.000	0.000	
E200	3.62	3.62	3.62	0.02%	0.000	0.000	0.000	

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Indoor Drybulb Temperature: Mean and (Max-Min)/ Mean

	Mean IDB (°C)				**(Max - Min)/ Mean IDB (°C)**			
	TUD	HTAL1	HTAL2	(Max-Min) / Mean	TUD	HTAL1	HTAL2	(Max-Min) / Mean
E100	22.20	22.20	22.22	0.07%	0.000	0.000	0.002	
E110	22.20	22.20	22.21	0.06%	0.000	0.000	0.002	
E120	26.70	26.70	26.71	0.04%	0.000	0.000	0.002	
E130	22.20	22.20	22.19	0.06%	0.000	0.000	0.001	
E140	22.20	22.20	22.19	0.07%	0.000	0.000	0.002	
E150	22.20	22.20	22.22	0.07%	0.000	0.000	0.002	
E160	26.70	26.70	26.71	0.05%	0.000	0.000	0.002	
E165	23.30	23.30	23.32	0.07%	0.000	0.000	0.002	
E170	22.20	22.20	22.20	0.00%	0.000	0.000	0.001	
E180	22.20	22.20	22.21	0.02%	0.000	0.000	0.001	
E185	22.20	22.20	22.21	0.03%	0.000	0.000	0.001	
E190	22.20	22.20	22.19	0.03%	0.000	0.000	0.001	
E195	22.20	22.20	22.20	0.02%	0.000	0.000	0.001	
E200	26.70	26.70	26.71	0.05%	0.000	0.000	0.000	

Humidity Ratio: Mean and (Max-Min)/ Mean

	Mean Humidity Ratio				**(Max - Min)/ Mean Hum. Ratio**			
	TUD	HTAL1	HTAL2	(Max-Min) / Mean	TUD	HTAL1	HTAL2	(Max-Min) / Mean
E100	0.0074	0.0073	0.0073	1.20%	0.000	0.000	0.000	
E110	0.0065	0.0064	0.0064	1.84%	0.000	0.000	0.000	
E120	0.0079	0.0079	0.0079	0.23%	0.000	0.000	0.000	
E130	0.0074	0.0073	0.0073	1.20%	0.000	0.000	0.00	
E140	0.0065	0.0064	0.0064	1.84%	0.000	0.000	0.00	
E150	0.0082	0.0082	0.0082	0.33%	0.000	0.000	0.000	
E160	0.0100	0.0099	0.0099	0.29%	0.000	0.000	0.000	
E165	0.0093	0.0092	0.0092	0.75%	0.000	0.000	0.000	
E170	0.0104	0.0105	0.0105	0.81%	0.000	0.000	0.001	
E180	0.0162	0.0162	0.0162	0.25%	0.000	0.000	0.001	
E185	0.0161	0.0161	0.0161	0.06%	0.000	0.000	0.001	
E190	0.0158	0.0159	0.0159	0.65%	0.000	0.000	0.001	
E195	0.0154	0.0154	0.0154	0.47%	0.000	0.000	0.001	
E200	0.0111	0.0111	0.0111	0.35%	0.000	0.000	0.000	

Space Cooling Electricity Consumption

	Total (kWh,e)				Compressor (kWh,e)				Supply Fan (kWh,e)				Condenser Fan (kWh,e)			
				(Max-Min)				(Max-Min)				(Max-Min)				(Max-Min)
	TUD	HTAL1	HTAL2	/ Mean	TUD	HTAL1	HTAL2	/ Mean	TUD	HTAL1	HTAL2	/ Mean	TUD	HTAL1	HTAL2	/ Mean
E100	1531	1531	1531	0.02%	1319	1319	1319	0.01%	144	144	144	0.07%	68	68	68	0.06%
E110	1076	1077	1077	0.11%	888	889	889	0.16%	128	128	128	0.18%	60	60	60	0.11%
E120	1013	1011	1011	0.17%	841	839	839	0.21%	117	117	117	0.01%	55	55	55	0.00%
E130	111	110	110	0.98%	95	94	94	1.04%	10	10	10	1.04%	5	5	5	1.81%
E140	69	69	68	1.01%	57	57	56	0.93%	8	8	8	1.40%	4	4	4	1.50%
E150	1206	1207	1207	0.00%	999	999	999	0.02%	141	141	141	0.09%	66	66	66	0.17%
E160	1140	1139	1139	0.14%	950	949	949	0.14%	129	129	129	0.08%	61	61	61	0.03%
E165	1498	1500	1500	0.12%	1279	1280	1280	0.12%	149	149	149	0.14%	70	70	70	0.13%
E170	641	638	638	0.53%	533	530	530	0.56%	74	73	73	0.53%	35	34	34	0.45%
E180	1083	1082	1082	0.07%	908	908	908	0.05%	119	119	119	0.23%	56	56	56	0.13%
E185	1545	1543	1543	0.11%	1340	1339	1338	0.12%	139	139	139	0.07%	65	65	65	0.08%
E190	165	164	164	0.65%	138	138	138	0.63%	18	18	18	0.94%	9	9	9	0.38%
E195	252	250	250	0.86%	219	217	217	0.84%	23	23	23	0.97%	11	11	11	1.09%
E200	1476	1477	1477	0.11%	1249	1250	1250	0.11%	154	155	155	0.09%	73	73	73	0.10%

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Coil Loads: Total, Sensible, and Latent

	Total (kWh,thermal)				Sensible (kWh,thermal)				Latent (kWh,thermal)			
				(Max-Min)				(Max-Min)				(Max-Min)
	TUD	HTAL1	HTAL2	/ Mean	TUD	HTAL1	HTAL2	/ Mean	TUD	HTAL1	HTAL2	/ Mean
E100	3800	3800	3800	0.00%	3800	3800	3800	0.00%	0	0	0	
E110	3765	3765	3765	0.01%	3765	3765	3765	0.01%	0	0	0	
E120	3749	3749	3749	0.01%	3749	3749	3749	0.01%	0	0	0	
E130	219	219	219	0.07%	219	219	219	0.07%	0	0	0	
E140	198	198	197	0.34%	198	198	197	0.34%	0	0	0	
E150	4518	4517	4518	0.02%	3778	3778	3779	0.02%	739	739	739	0.03%
E160	4501	4500	4500	0.01%	3761	3761	3761	0.01%	739	739	739	0.02%
E165	4537	4537	4538	0.02%	3798	3798	3799	0.02%	739	739	739	0.01%
E170	2232	2232	2233	0.03%	1493	1493	1493	0.03%	739	739	739	0.03%
E180	4495	4495	4494	0.03%	1538	1538	1538	0.03%	2957	2957	2956	0.05%
E185	4535	4535	4534	0.03%	1578	1578	1578	0.03%	2958	2957	2956	0.04%
E190	578	577	578	0.07%	208	208	208	0.10%	370	370	370	0.05%
E195	601	601	601	0.03%	232	232	232	0.02%	370	370	370	0.04%
E200	5498	5498	5498	0.00%	4277	4277	4277	0.00%	1221	1221	1221	0.01%

Zone Loads: Total, Sensible, and Latent

	Total (kWh,thermal)				Sensible (kWh,thermal)				Latent (kWh,thermal)			
				(Max-Min)				(Max-Min)				(Max-Min)
	TUD	HTAL1	HTAL2	/ Mean	TUD	HTAL1	HTAL2	/ Mean	TUD	HTAL1	HTAL2	/ Mean
E100	3656	3656	3656	0.00%	3656	3656	3656	0.00%	0	0	0	
E110	3637	3637	3637	0.00%	3637	3637	3637	0.00%	0	0	0	
E120	3632	3632	3632	0.00%	3632	3632	3632	0.00%	0	0	0	
E130	209	209	209	0.03%	209	209	209	0.03%	0	0	0	
E140	190	190	190	0.03%	190	190	190	0.03%	0	0	0	
E150	4376	4376	4376	0.00%	3637	3637	3637	0.00%	739	739	739	0.00%
E160	4371	4371	4371	0.00%	3632	3632	3632	0.00%	739	739	739	0.00%
E165	4388	4388	4388	0.00%	3649	3649	3649	0.00%	739	739	739	0.00%
E170	2159	2159	2159	0.00%	1420	1420	1420	0.00%	739	739	739	0.00%
E180	4376	4376	4376	0.00%	1420	1420	1420	0.00%	2957	2957	2957	0.00%
E185	4396	4396	4396	0.00%	1439	1439	1439	0.00%	2957	2957	2957	0.00%
E190	559	559	559	0.01%	190	190	190	0.03%	370	370	370	0.00%
E195	579	579	579	0.01%	209	209	209	0.03%	370	370	370	0.00%
E200	5343	5343	5343	0.00%	4122	4122	4122	0.00%	1221	1221	1221	0.00%

Fan Heat and Latent Loads Check

	Sensible Coil - Zone Load (Fan Heat) (kWh,th)			(Max-Min)	Latent Coil-Zone Load (kWh,th) (Should = 0)			(Max-Min)
	TUD	HTAL1	HTAL2	/ Mean	TUD	HTAL1	HTAL2	/ Mean
E100	144	144	144	0.03%	0	0	0	
E110	128	128	128	0.23%	0	0	0	
E120	117	117	117	0.27%	0	0	0	
E130	10	10	10	2.02%	0	0	0	
E140	8	8	8	7.67%	0	0	0	
E150	141	141	142	0.57%	0	0	0	
E160	129	129	129	0.32%	0	0	0	
E165	149	149	150	0.53%	0	0	0	
E170	74	73	74	0.54%	0	0	0	
E180	118	119	118	0.34%	1	0	-1	
E185	139	139	139	0.36%	1	0	-1	
E190	18	18	18	1.11%	0	0	0	
E195	23	23	23	0.44%	0	0	0	
E200	154	155	155	0.09%	0	0	0	

Sensitivities for Space Cooling Electricity Consumption, COP and Coil Loads

	Del Qtot (kWh,e)				Del Qcomp (kWh,e)				Del Q IDfan (kWh,e)				Del Q ODfan (kWh,e)			
	TUD	HTAL1	HTAL2	Abs(Max-Min)/ Mean	TUD	HTAL1	HTAL2	Abs(Max-Min)/ Mean	TUD	HTAL1	HTAL2	Abs(Max-Min)/ Mean	TUD	HTAL1	HTAL2	Abs(Max-Min)/ Mean
E110-E100	-454	-454	-453	0.25%	-431	-430	-430	0.33%	-16	-16	-16	1.73%	-7	-7	-7	0.31%
E120-E110	-64	-66	-66	4.35%	-47	-50	-50	6.37%	-11	-11	-11	1.97%	-5	-5	-5	1.30%
E120-E100	-518	-520	-520	0.37%	-478	-480	-480	0.40%	-27	-27	-27	0.37%	-13	-13	-13	0.35%
E130-E100	-1420	-1421	-1421	0.09%	-1224	-1225	-1225	0.08%	-134	-134	-134	0.12%	-63	-63	-63	0.07%
E140-E130	-42	-41	-41	1.42%	-38	-38	-38	1.22%	-2	-2	-2	0.29%	-1	-1	-1	2.97%
E140-E110	-1007	-1009	-1009	0.18%	-831	-833	-833	0.24%	-120	-120	-120	0.10%	-56	-56	-56	0.02%
E150-E110	130	129	129	0.89%	111	110	110	1.11%	13	13	13	0.84%	6	6	6	1.63%
E160-E150	-66	-67	-68	2.32%	-49	-50	-50	3.14%	-12	-12	-12	0.09%	-6	-6	-6	1.80%
E165-E160	357	360	361	0.92%	328	331	331	0.85%	20	20	20	1.56%	9	9	9	1.15%
E170-E150	-565	-569	-569	0.60%	-466	-469	-469	0.69%	-68	-68	-68	0.40%	-32	-32	-32	0.45%
E180-E150	-124	-124	-125	0.62%	-91	-91	-92	0.73%	-22	-23	-23	0.68%	-11	-11	-11	0.95%
E180-E170	442	445	444	0.69%	375	378	378	0.76%	45	45	45	0.48%	21	21	21	0.40%
E185-E180	462	461	461	0.21%	432	431	431	0.27%	21	21	21	0.89%	10	10	10	0.20%
E190-E180	-917	-918	-918	0.08%	-770	-770	-770	0.10%	-101	-101	-101	0.10%	-47	-47	-47	0.08%
E190-E140	96	96	96	0.60%	82	81	81	0.55%	10	10	10	0.56%	5	5	5	0.53%
E195-E190	87	86	86	1.38%	80	79	79	1.32%	5	5	5	1.10%	2	2	2	3.94%
E195-E185	-1292	-1293	-1293	0.07%	-1121	-1122	-1121	0.06%	-117	-117	-117	0.20%	-55	-55	-55	0.12%
E195-E130	142	141	141	0.77%	123	122	123	0.76%	12	12	12	0.92%	6	6	6	0.50%
E200-E100	-55	-53	-54	2.48%	-70	-69	-69	1.78%	10	11	11	1.84%	5	5	5	2.28%

Sensitivities for COP and Coil Loads

	Delta COP (kWh,t)				Del Q coil,t (kWh,t)				Del Q coil,s (kWh,t)				Del Qcoil,lat (kWh,t)			
	TUD	HTAL1	HTAL2	Abs(Max-Min)/ Mean	TUD	HTAL1	HTAL2	Abs(Max-Min)/ Mean	TUD	HTAL1	HTAL2	Abs(Max-Min)/ Mean	TUD	HTAL1	HTAL2	Abs(Max-Min)/ Mean
E110-E100	0.99	0.99	0.99	0.06%	-35	-35	-35	0.85%	-35	-35	-35	0.85%	0	0	0	
E120-E110	0.21	0.21	0.21	1.41%	-16	-16	-17	3.65%	-16	-16	-17	3.65%	0	0	0	
E120-E100	1.20	1.20	1.20	0.20%	-51	-52	-52	0.86%	-51	-52	-52	0.86%	0	0	0	
E130-E100	-0.50	-0.48	-0.48	3.98%	-3581	-3581	-3581	0.01%	-3581	-3581	-3581	0.01%	0	0	0	
E140-E130	0.86	0.86	0.86	0.08%	-21	-21	-22	2.38%	-21	-21	-22	2.38%	0	0	0	
E140-E110	-0.63	-0.61	-0.61	3.12%	-3567	-3567	-3568	0.03%	-3567	-3567	-3568	0.03%	0	0	0	
E150-E110	0.25	0.25	0.25	0.87%	752	752	753	0.10%	13	13	14	4.59%	739	739	739	0.03%
E160-E150	0.21	0.21	0.21	2.09%	-17	-17	-18	8.10%	-17	-17	-18	6.73%	0	0	0	403.08%
E165-E160	-0.90	-0.91	-0.91	0.72%	36	37	38	4.16%	36	37	38	3.65%	0	0	0	291.46%
E170-E150	-0.26	-0.24	-0.24	8.16%	-2285	-2286	-2286	0.03%	-2285	-2286	-2286	0.03%	0	0	0	300.00%
E180-E150	0.42	0.41	0.41	1.22%	-22	-23	-25	9.69%	-2241	-2240	-2241	0.05%	2218	2218	2217	0.07%
E180-E170	0.68	0.65	0.65	3.82%	2263	2263	2261	0.08%	45	45	45	1.78%	2218	2218	2217	0.07%
E185-E180	-1.20	-1.19	-1.19	0.55%	40	40	40	0.61%	40	40	40	0.48%	0	0	0	198.53%
E190-E180	-0.66	-0.63	-0.63	4.05%	-3918	-3918	-3916	0.04%	-1330	-1330	-1330	0.05%	-2588	-2587	-2586	0.06%
E190-E140	0.64	0.64	0.64	0.58%	380	379	380	0.26%	10	10	11	7.94%	370	370	370	0.05%
E195-E190	-1.09	-1.10	-1.10	0.72%	24	24	24	1.68%	24	24	24	0.84%	0	0	0	344.58%
E195-E185	-0.55	-0.54	-0.54	2.10%	-3934	-3934	-3933	0.03%	-1346	-1347	-1346	0.04%	-2588	-2587	-2587	0.05%
E195-E130	0.40	0.40	0.40	1.21%	382	382	382	0.03%	12	12	12	0.89%	370	370	370	0.04%
E200-E100	1.23	1.23	1.23	0.14%	1697	1697	1697	0.00%	476	476	476	0.01%	1221	1221	1221	0.01%

i220401a.xls r:s145..ak192; 05/18/01

2.5 References for Part II

These are references for Sections 2.1 and 2.2. References for Section 2.3 are included at the end of each analytical solution report (see end of 2.3.1 and 2.3.2).

ASHRAE Handbook of Fundamentals. (2001). Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

Dürig, M. (2000a). Hochschule Technik + Architektur Luzern. E-mail communications with J. Neymark. March 2000.

Dürig, M. (2000b). Hochschule Technik + Architektur Luzern. E-mail communication with J. Neymark. March 15, 2000.

Dürig, M., Glass, A., and Zweifel, G. (2000). *Analytical and Numerical Solution for Cases E100–E200. Based on Test Description September 1998 Edition.* Hochschule Technik + Architektur Luzern. Draft, August 2000.

Glass, A. (2000). Hochschule Technik + Architektur Luzern. E-mail communications with J. Neymark. February 14, 2000, and others from February–March 2000.

Knabe, G., and H-T. Le. (2000). *Analytical Solution HTA Luzern - TU Dresden*. Technische Universität Dresden. 17 Mar 2000. Submitted at Madrid, Spain, March 2000. Note: This document erroneously indicated TUD used an atmospheric pressure of 100,000 Pa for psychrometric calculations; the actual value was 101,000 Pa. See Le 2000.

Le, H-T. (2000). Technische Universität Dresden. E-mail communication with J. Neymark. June 20, 2000.

Le, H-T., and G. Knabe. (1998). *HVAC BESTEST: Results of Analytical Solutions.* Dresden University of Technology. E-mail from Le to Judkoff. June 19, 1998.

Le, H-T., and G. Knabe. (1999a). *Solutions Techniques for Dry Coil Conditions, "Analytical Solutions for Case E110."* Dresden University of Technology. Fax from Le to Neymark. May 19, 1999.

Le, H-T., and G. Knabe. (1999b). *Solutions Techniques for Dry Coil Conditions, "Analytical Solutions for Case E170."* Dresden University of Technology. Fax from Le to Neymark. April 30, 1999.

Zweifel, G. and Dürig, M. (1999). *Analytical and Numerical Solution for Cases E100–E200. Based on Test Description September 1998 Edition.* Hochschule Technik + Architektur Luzern. Draft, May 1999.

2.6 Abbreviations and Acronyms for Part II

ASHRAE	American Society for Heating, Refrigerating, and Air-Conditioning Engineers
COP	Coefficient Of performance
EDB	Entering Dry-Bulb temperature
EWB	Entering Wet-Bulb temperature
HTAL	Hochschule Technik + Architektur Luzern

HTAL1	Analytical solution results by HTAL
HTAL2	Initial HTAL “numerical” solution technique applying realistic controller with 1-s timestep
IDB	Indoor Dry-Bulb temperature
IEA	International Energy Agency
Mean	Average of TUD, HTAL 1, and HTAL2 solutions
Max	Maximum of TUD, HTAL1, and HTAL2 solutions
Min	Minimum of TUD, HTAL1, and HTAL2 solutions
SHC	Solar Heating and Cooling (Programme)
TUD	Technische Universität Dresden; in Section 2.4 this refers specifically to analytical solution results by TUD

PART III:

Production of Simulation Results

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 Europe and the United States. The objectives of developing the simulation results were:

- To demonstrate the applicability and usefulness of the Building Energy Simulation Test for Heating, Ventilating, and Air-Conditioning Equipment Models (HVAC BESTEST) test suite
- To improve the test procedure through field trials
- To identify the range of disagreement that may be expected for simulation programs relative to the analytical solution results that constitute a reliable set of theoretical results for these specific test cases (see Part IV).

The field trial effort took about 3 years and involved several revisions to the HVAC BESTEST specifications and subsequent re-execution of the computer simulations. The process was iterative in that executing the simulations led to the refinement of HVAC BESTEST, and the results of the tests led to the improvement and debugging of the programs. This process underscores the importance of International Energy Agency (IEA) participation in this project; such extensive field trials, and resulting enhancements to the tests, were much more cost effective with the participation of the IEA Solar Heating and Cooling (SHC) Programme Task 22 experts.

Table 3-1 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 analytical solutions used in this study. The analytical solution results constitute a reliable set of theoretical results. Therefore, the primary purpose of including simulation results for the E100–E200 cases in Part IV is to allow simulationists to compare their relative agreement (or disagreement) with the analytical solution results *versus* the relative agreement of the Part IV simulation results with the analytical solution results (i.e., a comparison with the state of the art in simulation). Perfect agreement among simulations and analytical solutions is not necessarily expected. The Part IV results give an indication of what sort of reasonable agreement is possible between simulation results and the analytical solution results.

Acronyms used in Sections 3.2 through 3.6 are given in Section 3.7. References cited in Section 3.2 through 3.6 are given in Section 3.8.

3.2 Selection of Simulation Programs and Modeling Rules for Simulations

The countries participating in this IEA task 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 calculational 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 country 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 HVAC BESTEST (see Section 1.2.2, 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., detailed component data are not given for the compressor, condenser, and thermal expansion device).

Table 3-1. Participating Organizations and Computer Programs

Model	Authoring organization	Implemented by
CA-SIS V1	Electricité de France, France	Electricité de France, France
CLIM2000 2.1.6	Electricité de France, France	Electricité de France, France
DOE-2.1E-088	LANL/LBNL/ESTSC,[a,b,c], USA	CIEMAT,[d] Spain
DOE-2.1E-133	LANL/LBNL/JJH,[a,b,e], USA	NREL/JNA,[f], USA
ENERGYPLUS 1.0.0.023	LBNL/UIUC/CERL/OSU/GARD Analytics/FSEC/DOE-OBT,[a,g,h,i,j,k]	GARD Analytics, USA
PROMETHEUS	Klimasystemtechnik, Germany	Klimasystemtechnik, Germany
TRNSYS 14.2-TUD with ideal controller model	University of Wisconsin, USA; Technische Universität Dresden, Germany	Technische Universität Dresden, Germany
TRNSYS 14.2-TUD with real controller model	University of Wisconsin, USA; Technische Universität Dresden, Germany	Technische Universität Dresden, Germany

[a]LANL: Los Alamos National Laboratory
[b]LBNL: Lawrence Berkeley National Laboratory
[c]ESTSC: Energy Science and Technology Software Center (at Oak Ridge National Laboratory, USA)
[d]CIEMAT: Centro de Investigaciones Energéticas, Medioambientales y Tecnologicas
[e]JJH: James J. Hirsch & Associates
[f]NREL/JNA: National Renewable Energy Laboratory/J. Neymark & Associates
[g]UIUC: University of Illinois Urbana/Champaign
[h]CERL: U.S. Army Corps of Engineers, Construction Engineering Research Laboratories
[i]OSU: Oklahoma State University
[j]FSEC: University of Central Florida, Florida Solar Energy Center
[k]DOE-OBT: U.S. Department of Energy, Office of Building Technology, State and Community Programs, Energy Efficiency and Renewable Energy

To minimize the potential for user error, we encouraged more than one modeler to develop input files for each program. This was done for DOE-2.1E (USA/Spain). Where disagreement in the inputs or results was found, we requested that the two modelers resolve the differences. Where only a single modeler was involved, we strongly recommended that another modeler familiar with the program check the inputs carefully.

3.3 Improvements to the Test Specification as a Result of the Field Trials

Based on comments by the other IEA SHC Task 22 participants during the field trials, observations from our own DOE-2.1E.simulations, and comments by industry engineers, 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.

The contribution of the IEA SHC Task 22 participating countries was particularly valuable because the Task 22 experts supplied continuous feedback throughout the 3-year field trial effort. Their feedback resulted in several revisions to the HVAC BESTEST specifications and subsequent re-execution of the computer simulations. This iterative process led to refinement of HVAC BESTEST, and the results of the tests led to the improvement and debugging of the programs. The process underscores the leveraging of resources for the IEA countries participating in this project. Such extensive field trials, and resulting enhancements to the tests, would not have occurred without the participation of the IEA SHC Task 22 experts.

Revisions to HVAC BESTEST were isolated in various addenda to the original (and subsequent) user's manuals (Neymark and Judkoff 1998–2000). Most of the revisions outlined below were made during the earlier stages of the project.

- Improved description of manufacturer performance data included:
  - Fan heat assumed by the manufacturer (which is not equal to the listed fan power)
  - Additional tables that list gross capacities and adjusted net capacities
  - Clarification text about the validity of listed performance data, instructions for interpolation and extrapolation, and instructions for modeling dry coil conditions
  - Clearer definition of part load ratio (PLR) and adjustment of the COP=f(PLR) (performance degradation, CDF) curve.
- General test specification improvements included:
  - Improved glossary and overall improvement in definition of terms
  - Use of a draw-through rather than a blow-through indoor fan
  - Adjustment of load inputs to achieve Air-Conditioning and Refrigeration Institute (ARI) conditions
  - Notation of relative humidity of the weather data, along with a discussion about the weather-data solar time convention
  - Modeling rules that require consistent modeling methods (rather than most detailed modeling methods), and clarification of model initialization and simulation period
  - Clarification about thermostat control strategy
  - Operating assumptions that include: perfectly mixed zone air, zone loads distributed evenly throughout the zone, no system hot gas bypass, and no compressor unloading.
- Additional equivalent inputs included:
  - Discussion of bypass factor with a calculation appendix
  - Indoor fan performance data with a calculation appendix
  - Evaporator coil details
  - Minimum supply air temperature.

3.4 Examples of Error Trapping with HVAC BESTEST Diagnostics

This section summarizes a few examples that demonstrate how the HVAC BESTEST procedure was used to isolate and correct bugs in the reference programs. Further description may be found in the individual code reports presented in the next section.

Simulations were performed for each test case with the participating computer programs using four sets of hourly typical meteorological year (TMY) weather data modified to give constant outdoor dew point temperature and constant outdoor dry bulb temperature (ODB) for 3 consecutive months. These artificial weather data were applied because they allow steady-state analysis, which facilitated the development of analytical solutions for comparison with simulation results. At each stage of the exercise, output data from the simulations were compared to each other according to the diagnostic logic of the test cases (see Part I, Appendix F). In the final stages of the exercise, the analytical solutions were compared. The test diagnostics revealed (and led to the correction of) bugs, faulty algorithms, input errors, or some combination of those in every one of the programs tested. Several examples follow.

3.4.1 DOE2.1E Version W-54 (NREL)

DOE-2 is a whole-building simulation program, the development of which has been sponsored by the U.S. Department of Energy (DOE). NREL used DOE-2.1E's RESYS2 (residential system) for its model.

3.4.1.1 Minimum Supply Temperature Bug in RESYS2 (36% compressor+ODfan consumption issue)

In the earliest stage of the HVAC BESTEST test specification development and testing (around August of 1994, before the IEA SHC Task 22 began), a problem was identified for the RESYS2 system in a version older than DOE-2.1E W-54. Identical input decks (for a much different preliminary version of the test specification) were used—the only difference between the input decks was the designation of the DOE-2 SYSTEM-TYPE as PSZ versus RESYS2. Table 3-2 and Figure 3-1 show the comparison results.

Table 3-2. DOE-2.1E. Version W-54 System Disagreements: RESYS2 versus PSZ

DOE-2.1E System	Compr[a]+OD[b] fan Elec.[c] (kWh)	Total Coil Clg.[d] (kWh)	Latent Clg. (kWh)	"COP"[e]
PSZ	2,587	7,532	1,202	2.9
RESYS2	1,646	7,767	1,524	4.8

[a]Compr = compressor
[b]OD = outdoor
[c]Elec = electricity consumption
[d]Clg. = total evaporator coil load
[e]"COP" = (Total Coil Clg.)/(Compr+ODfan Elec.)

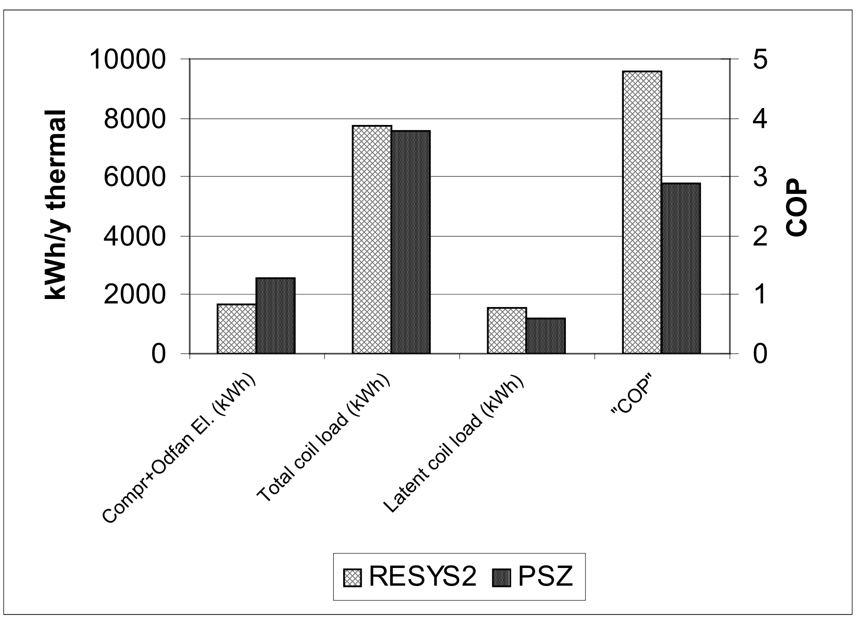

Figure 3-1. DOE-2.1E RESYS2 minimum supply temperature bug

In response to this 36% consumption difference and the unreasonably high COP for the RESYS2 result, the code author explained that a bug had been found in the RESYS2 system model. In that model, when the indoor fan mode is set to INTERMITTENT, the capacity calculation that sets the minimum supply temperature (TCMIN) used the wrong value, resulting in the unrealistically high COP for the RESYS2. The program authors corrected this coding error (Hirsch 1994). This is an example of how systematic checking for internal consistency among outputs required by HVAC BESTEST led to the discovery of an error.

3.4.2 DOE2.1E JJH Version 133 (NREL)

This summary is for the version of DOE-2.1E distributed by James J. Hirsch & Associates, Camarillo, California, USA. NREL used DOE-2.1E's RESYS2 (residential system) for its model.

3.4.2.1 Issues Transmitted to Code Authors

For DOE-2.1E, some disagreements and inconsistencies were also discovered.

- Fan heat discrepancy (up to 5% total coil load inconsistency at low sensible heat ratio [SHR]); more discussion of this is included with the DOE-2.1E ESTSC version 088 summary of CIEMAT's results (below).
- The indoor (ID) fan does not precisely cycle on/off with compressor (2% total consumption disagreement at mid-PLR).
- COIL-BF-FT multiplier insensitivity (1% total consumption disagreement for E185).

The code authors have been notified about these issues, which are covered in more detail in the DOE-2.1E/NREL modeler report.

3.4.2.2 Changes from Version 117 to Version 133

Before January 10, 2000, the preliminary DOE-2.1E simulation work was done with an older version (117, current version = 133). The simulation work spotted two problems (listed below) in the older version. Note that although these problems were already fixed for version 133, they still existed in many previous versions—if there had been a systematic test in place sooner, we may have been able to address these problems sooner.

- Minimum entering wet bulb (EWB) was "hardwired" at 60°F (3% consumption disagreement for dry coil E110).
- Bypass factor f(PLR) multiplier = 0.99 at full load (should be 1.00); 0.4% total consumption disagreement for mid-PLR case E170.

For the hardwired minimum EWB issue see Figure 3-2 for a comparison of results from case E110 for version 117, version 133, and the analytical solutions. To help visualize the differences, the y1-axis minimum has been raised and the ID fan energy was multiplied by 5. See the DOE-2.1E/NREL modeler report for more detail on these issues.

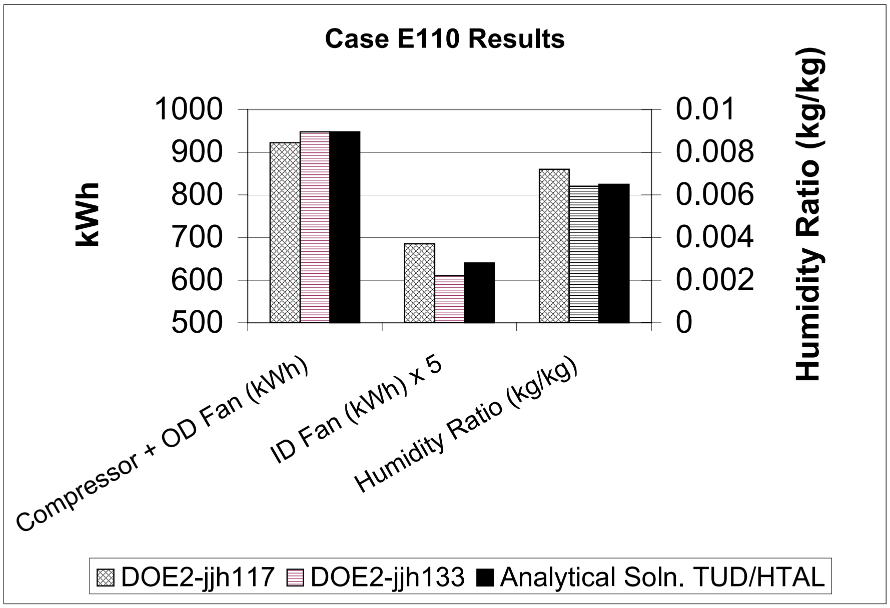

Figure 3-2. Release of Hardwired EWBmin = 60°F for Case E110

3.4.3 DOE2.1E ESTSC Version 088 (CIEMAT)

This summary applies to the version of DOE-2.1E distributed by the Energy Science and Technology Software Center of Oak Ridge National Laboratory, Oak Ridge, Tennessee, USA. CIEMAT used DOE-2.1E's PTAC (Packaged Terminal Air Conditioner) system for its model.

3.4.3.1 Issues Transmitted to Code Authors

Two issues were identified for the code authors:

- The ID fan does not precisely cycle on/off with compressor (2% total consumption disagreement at mid-PLR); this is the same problem that NREL found for the RESYS2 system in the JJH version.
- There is a "fan heat" discrepancy (up to 2% sensible coil load inconsistency at low SHR).

In terms of the fan heat discrepancy, in all cases except E130 and E140, fan heat (fan energy consumption) was not equal to sensible cooling coil load minus sensible zone load. Table 3-3, developed by CIEMAT, indicates these fan heat differences for each case.

A fan heat modeling error, or some other inconsistency between DOE-2.1E's LOADS and SYSTEMS subroutines, could cause this output problem.

Although both NREL and CIEMAT observed this problem, the observed discrepancy for their models differed. CIEMAT gave the following possible reasons for the differences between CIEMAT's and NREL's observed differences, including:

- Use of PTAC versus RESYS2
- Blow-through fan for PTAC, versus draw-through fan for RESYS2 which also:
  - Affects supply air temperature
  - Requires minor differences in curve fit data input for performance equivalence
- Different input methods for fan performance parameters
  - NREL: KW/CFM and DELTA-T
  - CIEMAT: static pressure and efficiency.

Note: NREL's earlier sensitivity tests indicate that this is not likely to be the source of the difference.

Table 3-3. Sensible Coil Load versus Sum of Sensible Zone Load and Fan Consumption for DOE-2.1E PTAC System

CASE	SENS COIL LOAD	SENS ROOM LOAD + SUPPLY FAN ENERGY	ERROR
E100	3841.47	3799.605	1.10%
E110	3803.58	3768.518	0.93%
E120	3763.48	3740.053	0.63%
E130	215.778	215.66	0.05%
E140	195.53	195.45	0.04%
E150	3803.58	3768.518	0.93%
E160	3777.178	3749.504	0.74%
E165	3828.258	3788.549	1.05%
E170	1486.857	1479.473	0.50%
E180	1553.184	1528.865	1.59%
E185	1608.094	1571.905	2.30%
E190	203.007	202.641	0.18%
E195	225.64	225.054	0.26%
E200	4313.176	4269.653	1.02%
MEAN ERROR			**0.81%**

ERROR = [(sens.zone load+fan energy)–(sens.coil load)]/(sens.zone load + fan energy)*100%

3.4.4 TRNSYS-TUD with Realistic Controller (TUD)

TRNSYS is considered to be the most advanced program that DOE has sponsored for simulating active solar systems. Originally written at the University of Wisconsin, Technische Universität Dresden (TUD) acquired a license for the source code and has since developed new source code for TUD's own calculation routines. This new version is designated TRNSYS-TUD, and some new routines developed at TUD were tested for this project. For this project TUD ran two different TRNSYS simulation models: one with an ideal controller and one with a realistic controller. The problems described below occurred in the model that incorporated the realistic controller.

3.4.4.1 Problem with Use of Single Precision Variables (45% total consumption disagreement at low PLR, 14% disagreement at mid-PLR)

For the initial set of TRNSYS-TUD results at low part loads (cases E130, E140, E190, and E195), large (43%–48%) errors were found in the sensible and latent coil loads; these errors also propagated through to energy consumption predictions. For the mid-PLR case E170, the error was also high (at 14% for energy consumption).

Diagnostic logic indicated that the problem could be with the application of the part load curve. On further review, the code authors discovered a problem with the use of single-precision variables in one of the calculation subroutines associated with the model that applies a realistic system controller. This caused rounding errors that became worse as PLR decreased.

Figure 3-3 documents the results of TUD's simulations before and after this problem was fixed. The figure includes a comparison with TUD's analytical solution results and shows that when the appropriate variables were changed to double-precision variables, the simulation results improved a great deal.

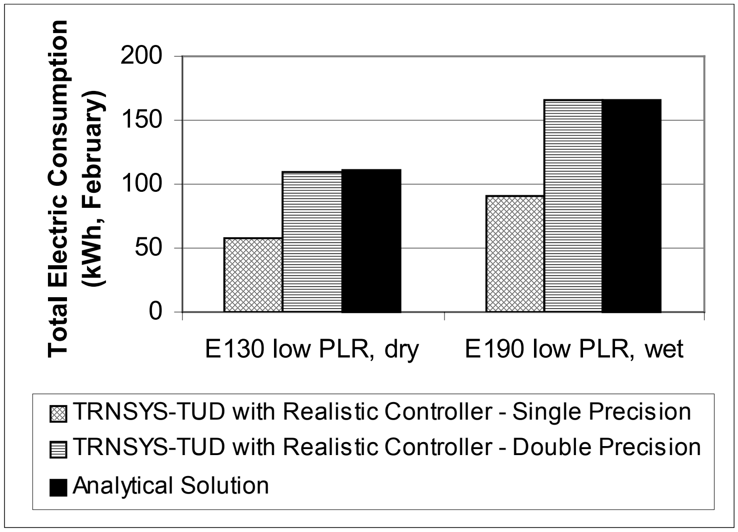

Figure 3-3. TRNSYS-TUD single precision variable bug

3.4.4.2 Wrong Data Compiled for Latent Coil Load Output (3% latent coil disagreement for E170, 4% total consumption disagreement for E150, E160)

After repairing the single-precision variable problem from March 1999, the September 1999 results for TRNSYS with the realistic controller showed that the latent coil load was 3% less than the latent zone load for case E170 (mid-PLR; see Figure 3-4). After checking by TUD—a review that also indicated 4% energy consumption variation from their analytical solution results for cases E150 and E160—an error was found and corrected. Quoting from TUD's modeler report:

> *A type (module) is used to save the entering dry bulb temperature (EDB) and the entering humidity ratio (EHR) for calculation of the equipment performance. This type was called at the end of every timestep but before the printer. By calling of this type all the defined equations in [the subroutine] are updated. After that the printer prints the just updated values. So the results were inconsistent. This type has been set now after the printer and it works well.*

Interestingly, two iterations were necessary to fix this error. TUD's first attempt to fix the problem indicated improved agreement for some outputs. However, the results summary for March 2000 indicated that for case E170 (mid-PLR), their latent coil load was 6% greater than their latent zone load (NREL 2000). For the next cycle of results they went back, undid their previous code revision, and incorporated the final fix (described above; Le 2000). The new TRNSYS-TUD results for the realistic controller now give much better agreement with the analytical solution results. This iterative process underscores the diagnostic power of HVAC BESTEST for ensuring that the problem that needed correction got corrected.

As a result of the fix, latent coil loads now match latent zone loads for all cases. Figure 3-4 illustrates the incremental improvements to TUD's software where "Qtotal" signifies total energy consumption and "Qcoil latent" signifies latent coil load. Note that compensating errors initially (in March 1999) showed agreement for E150 total consumption.

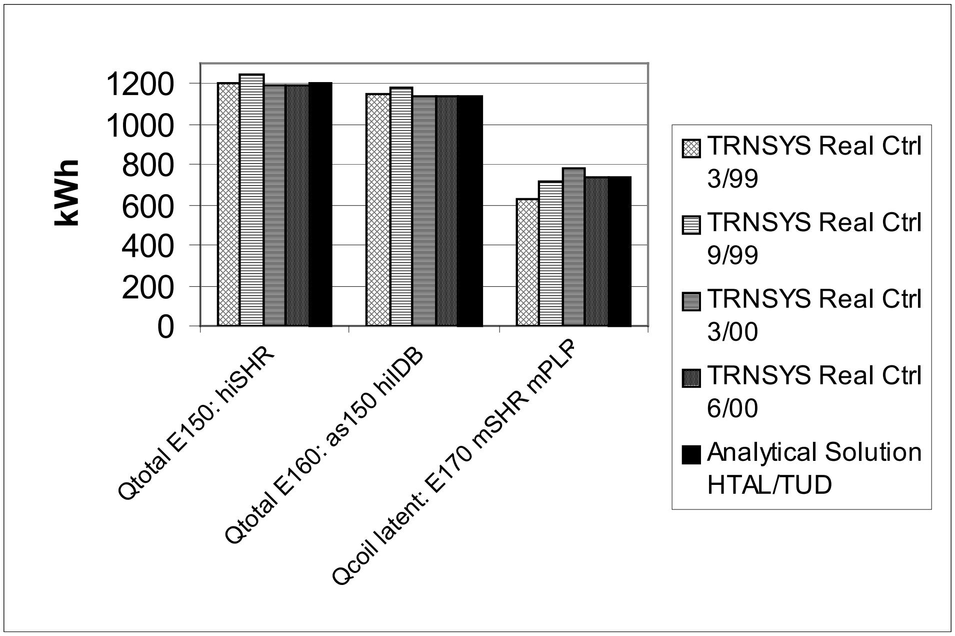

Figure 3-4. TRNSYS-TUD data gathering bug fix

3.4.4.3 Additional Observations about Diagnosing the TRNSYS Errors

It is interesting to observe, in the case of the second TRNSYS-TUD error, that only after the first error was fixed did the second error became apparent. This classic example of how some errors mask others further underscores the utility of the HVAC BESTEST diagnostics in being able to uncover both problems.

Additionally, the second example indicates the enhanced robustness of having an analytical truth standard. Although the error was initially flagged by the inconsistency among the latent coil loads and latent zone loads, the consumption errors were not apparent because at the time we had no agreeing analytical verification solutions and because of noise among the simulation results. Only after the TUD code authors compared their TRNSYS results with their own analytical solution (which yielded results very close to the final analytical solutions) were they able to identify the important discrepancies in the consumption results that initiated this revision.

3.4.5 CLIM2000 (EDF)

CLIM2000, developed by the French national electric utility Electricité de France (EDF), is an advanced modular building energy simulation software program.

Four different models were used in this study:

- The initial set of results used an older mechanical system model. Previous validation exercises had made EDF aware of some problems with this model. The code authors submitted results for two possible modeling methods appropriate to that model (see “CLIM2000-1a” and “CLIM2000-1b” in Figure 3-5).
- The second set of results used a new mechanical system model on which development had begun before this validation project started. The new model was completed midway through this project (see “CLIM2000-2” in Figure 3-5).
- The third and fourth sets of results incorporated improvements to the new model that resulted from work on this project (see “CLIM2000-3” and “CLIM2000-4” in Figure 3-5).

Figure 3-5 summarizes the results of the different models, along with the analytical solutions of TUD/ Hochschule Technik + Architektur Luzern (HTAL). Details on analysis of the results and changes to the models are given in the sections that follow.

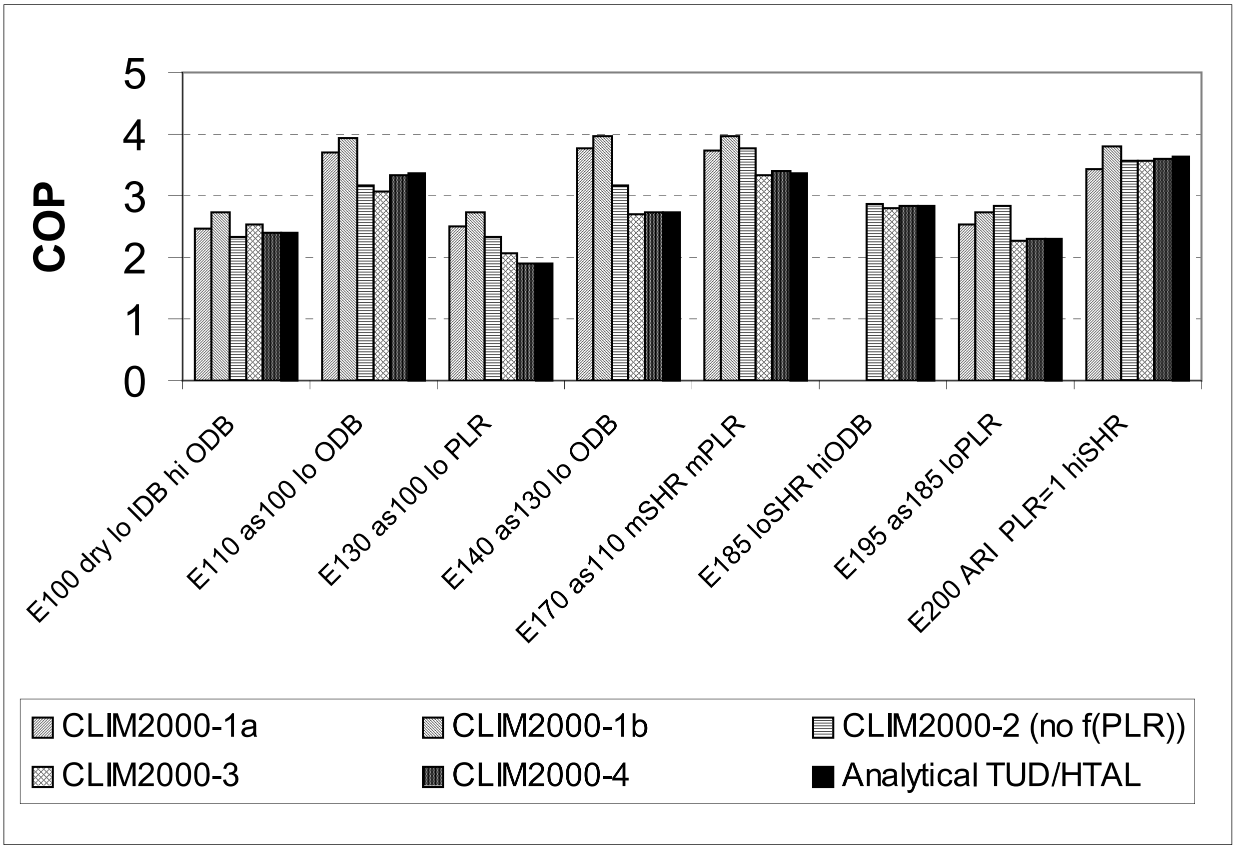

Figure 3-5. CLIM2000 new model comparison and improvements

3.4.5.1 Evaluation of the New Mechanical Equipment Model (up to 50% COP effect)

From these results, it is apparent that EDF's new mechanical equipment model (designated by "CLIM2000-2") is a big improvement over the previous model (for most but not all cases—see E195). The previous model had up to 50% disagreement in the worst case (E140 low PLR, low ODB). Also, the new model was able to model case E185 (high latent load); the old model was the only simulation that was unable to model that case. Using BESTEST in this way—where the changes to the model are already in progress—ensures that intended changes are actually made. From Figure 3-5, it is apparent that much of the intended improvement did occur, however, some problems remained as described in the next sections.

3.4.5.2 No COP=f(PLR) Sensitivity (20% consumption disagreement at low PLR, 13% disagreement at mid-PLR)

The results for the second set of runs indicated that the new model was not sensitive to performance (COP) degradation at part load. Figure 3-6 (extracted from EDF's modeler report) gives COP sensitivity comparison for the new model, a comparison that indicates there is no COP=f(PLR) sensitivity. See especially the highlighted (arrows) results in Figure 3-6 (i.e., for E130–E100, E140–E110, E190–E180, and E195–E185). These all check the PLR effect over default conditions as described in the appendix to Part I on diagnostic logic. (Note: In Figure 3-6, "BESTEST3" in the header above the graph is EDF's designator for the same set of results that we have called "CLIM2000-2" in Figure 3-5.)

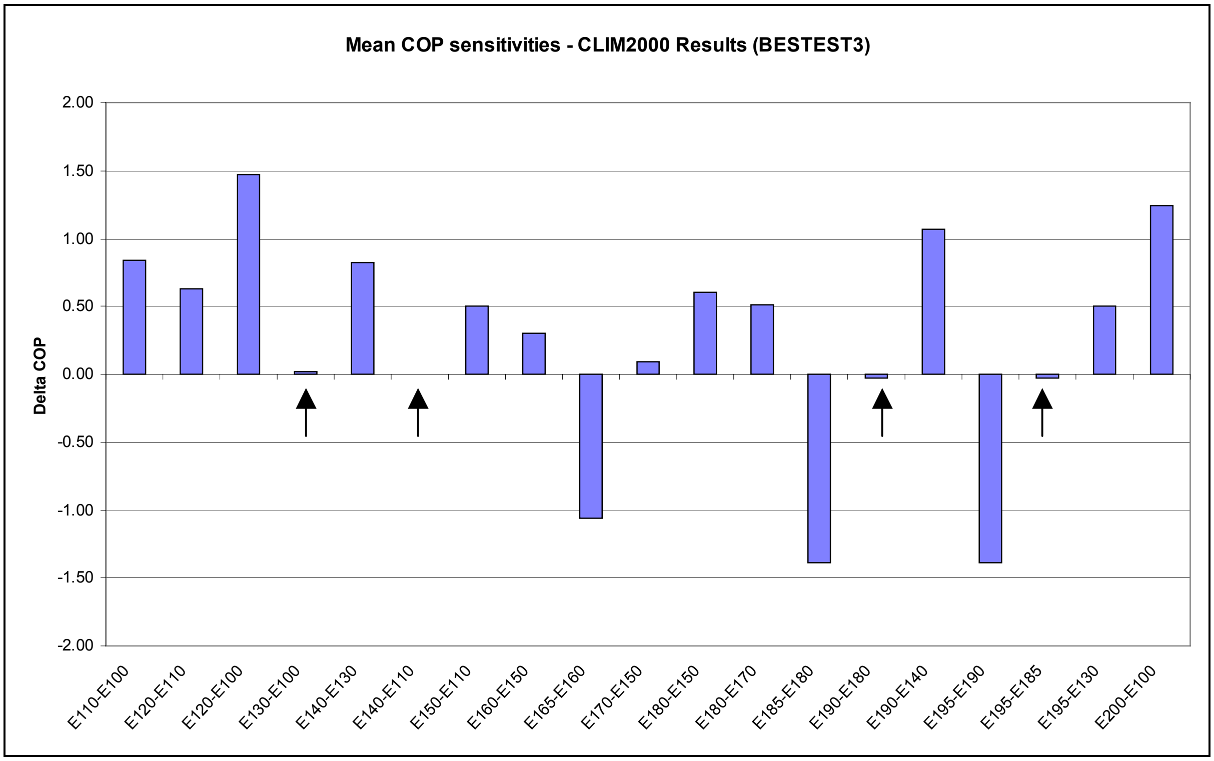

Figure 3-6. Mean COP sensitivities—CLIM2000-2 results

Figure 3-5 indicates that the lack of sensitivity gives a 20% COP (or consumption) error at low PLR (E130, E195), and a 13% error at mid-PLR (E170). These errors occurred because no COP degradation for cycling was implemented in the model. Based on these results, EDF decided to implement COP degradation for part load cycling into CLIM2000. In the “CLIM2000-3” results, the part load COP has better agreement (as indicated in Figure 3-5; especially notice cases E130, E140, E170, and E195).

3.4.5.3 Improved Performance Map Interpretation (up to 9% consumption effect for dry coils [E110])

After EDF completed its third set of results, there remained 7% and 10% disagreements in total consumption versus the analytical solutions for cases E100 and E110, respectively. In EDF’s fourth modeling, the code authors improved CLIM2000’s interpretation of the performance data by automating data extrapolation with EWB, including recognition of dry coil conditions, and manually extrapolating for EDB. Their results for the dry-coil cases are now within 1% of the analytical solutions. Results of this improvement are designated by CLIM2000-4 in Figure 3-5; especially notice cases E100 and E110 in the figure.

3.4.5.4 Comments on Compensating Errors and Diagnostics

The agreement for CLIM2000-2 in E100 caused by compensating errors is noteworthy. It is interesting to see how the correction for PLR is indicated as needed in other cases (e.g., E140) but not indicated as needed in E100, even though it should have been (although to a lesser degree than in E140). When the insensitivity to PLR was corrected in the CLIM2000-3 results, the E100 results then helped reveal a performance map interpretation error in CLIM2000-3 (previously concealed in E100, but perhaps not in E110).

Also, CLIM2000-2 has good agreement in cases E200 and E185—full load and near-full-load cases with moderate and high latent loads (high and low SHR), respectively—that do not test either sensitivity to part loading or operation at dry coil conditions. Because of their character, cases E200 and E185 are not sensitive to the algorithmic changes by EDF after CLIM2000-2, although they were useful in identifying problems with CLIM2000-1.

It is apparent, then, that it is the variety of cases founded by the diagnostic logic associated with HVAC BESTEST that allows a greater number of possible errors to be identified—even those errors that may sometimes be concealed by compensating disagreements.

3.4.6 CA-SIS (EDF)

CA-SIS is the whole-building hourly simulation program that EDF developed and used for building energy studies. The calculation engine for CA-SIS is the TRNSYS solver, property of the University of Wisconsin's Solar Energy Laboratory (USA), marketed in France by CSTB (Centre Scientifique et Technique du Batiment).

EDF used three different sets of simulations in this study:

1) The initial set of results featured:
    a. No extrapolation of the performance map
    b. No accounting for fan heat in the coil load
    c. No accounting for CDF in the energy consumptions of the compressor, indoor fan, and outdoor fan—although the initial results indicate COP sensitivity to part loading, EDF cites compensating errors as the cause (Hayez 2000)
    d. Zone humidity ratio variation between time steps limited using mathematical relaxation to be able to run all cases and to improve convergence (see the CA-SIS modeler report [Appendix III-D]).
2) The second set of results featured the following differences from the first set:
    a. Manual extrapolation of the performance data
    b. Accounting for fan heat in the coil load
    c. Accounting of CDF for the compressor
    d. Additional parameter variation between time steps limited using mathematical relaxation (zone humidity [more limited than before], zone temperature, and envelope load) for case E200; see the CA-SIS modeler report (Appendix III-D).
3) The third set of results featured the following differences from the second set:
    a. Revised manual extrapolation of the performance data including dry coil performance limits
    b. Accounting of CDF for indoor and outdoor fans
    c. Zone temperature variation between time steps was limited using mathematical relaxation for all cases.

The COP results for these models, along with the TUD/HTAL analytical solution results, are summarized in Figure 3-7. Figure 3-8 includes specific results used to diagnose the causes for various disagreements; relevant outputs have been scaled for convenience so that we could include them all in this figure. As a result of this modeling and analysis work, the following specific problems were identified for CA-SIS.

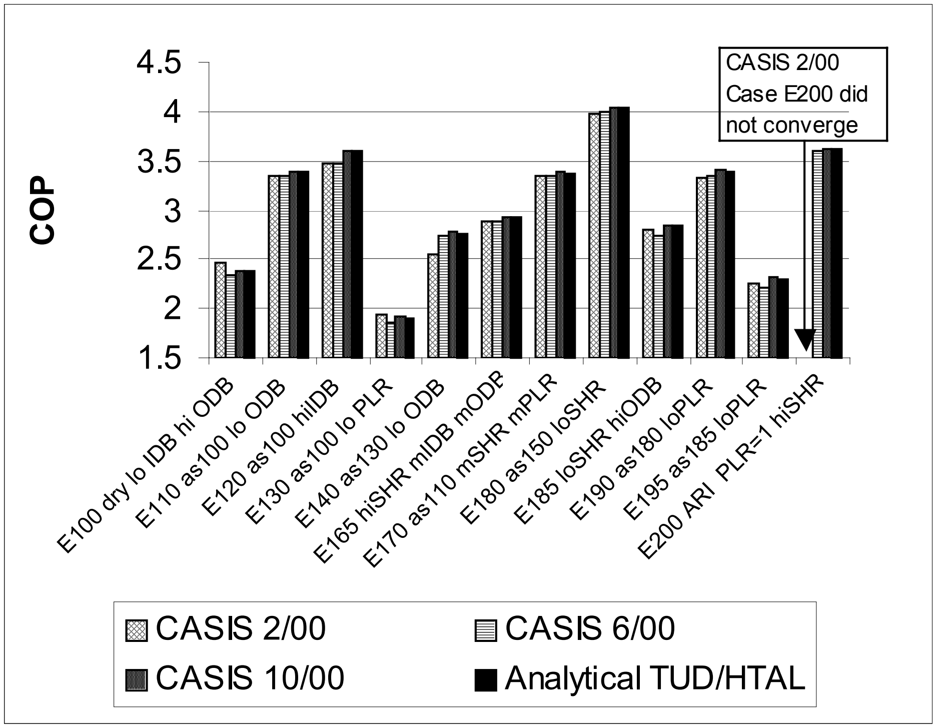

Figure 3-7. COP improvements to CASIS models

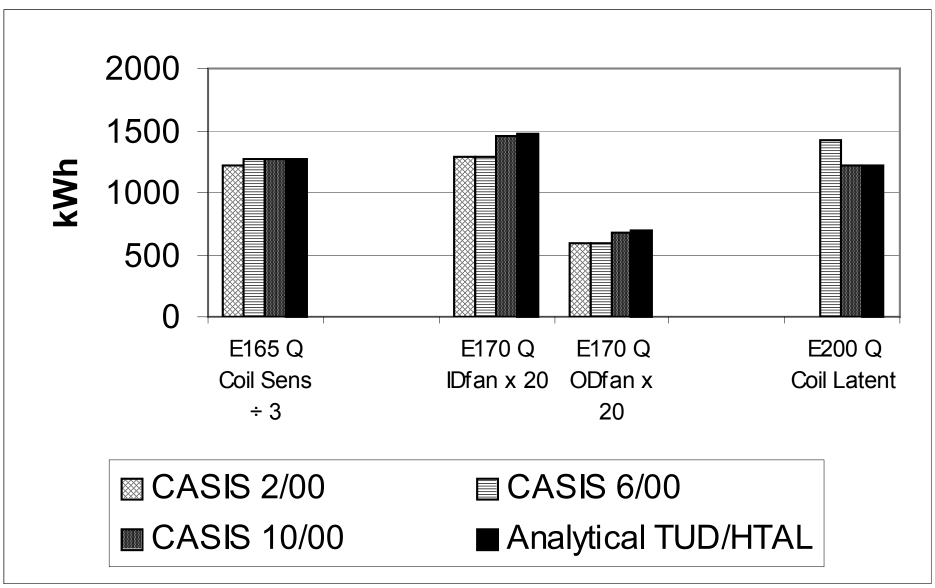

Figure 3-8. CASIS fixes: fan heat, fan CDF, and convergence algorithm

3.4.6.1 Fan Heat Not Included in Total Coil Load (4% sensible coil load effect for all cases)

In Figure 3-8 it is apparent from the CA-SIS 2/00 results for sensible coil load (shown by the group of results labeled “E165 Q Coil Sens”) that sensible coil load was about 4% lower than the analytical solution results. This difference was seen in all the sensible coil load outputs and was traced to fan heat not being accounted for in the coil load. After fan heat was included in the coil load (see “CA-SIS 6/00” results), the sensible coil loads agreed with the analytical solution results. Because multiple changes were made between the first and second models (see above) and because of probable compensating errors, it was not possible to isolate this effect in the compressor power results. However, the effect on consumption should also reasonably have been 4% for dry coil cases, and less than that for wet coil cases (decreasing with decreasing SHR).

3.4.6.2 Indoor and Outdoor Fan Power Did Not Include COP=f(PLR) Degradation (2% total consumption effect at mid PLR)

In Figure 3-8, it is apparent from the "CA-SIS 2/00"and "CA-SIS 6/00" results for indoor fan energy (shown by the group of results labeled "E170 Q IDfan × 20" and "E170 QODfan × 20"), that indoor fan energy consumption was about 12% lower than the analytical solution results for case E170. These differences were traced to CDF not being accounted for in the indoor and outdoor fan consumptions. This has a 2% effect on total energy consumption for these cases, and would have a higher percentage of effect at a lower PLR (and a lower percentage of effect at a higher PLR).

3.4.6.3 Convergence Algorithm (E200, full load case, could not be run)

The Figure 3-7 COP results for the group labeled "E200 ARI" and Figure 3-8 results for the group labeled "E200 Q Coil Latent" do not indicate results for "CA-SIS 2/00" case. This is because initially CA-SIS had convergence problems for case E200, and could not run it. To fix the problem, in its second run of E200, EDF applied limits to variation between time steps (using mathematical relaxation) to the following parameters: zone humidity ratio (more limited than before), zone temperature, and envelope load. With this change, the E200 COP results in Figure 3-7 are improved. EDF has also changed the default value of a coefficient used in CA-SIS's convergence algorithm.

The E200 latent coil load results in Figure 3-8 indicate an error in the calculation of latent coil load. This remaining latent coil load disagreement turns out to have been related to improper extrapolation of the performance map (Hayez 2000). For its third CA-SIS simulation, EDF revised the performance map inputs for CA-SIS to indicate the limits of performance for dry coil conditions. Integrating this new performance map resulted in an agreeing latent coil load for the CA-SIS 10/00 results, as shown in Figure 3-8.

3.4.6.4 No Automated Extrapolation of Performance Data (possibly up to 10% consumption effect in E110)

In the initial "CA-SIS 2/00" results, the COP for all the dry-coil cases differs from the TUD analytical solutions by 0.04 to 0.20 units of COP (1.2%–7.3%); see especially case E140 in Figure 3-7. The dry-coil cases require a small degree of extrapolation, as well as careful attention to which of the given manufacturer data points correspond to wet coil conditions (all valid data) and which correspond to dry coil conditions (and do not give valid compressor consumptions and total capacities). In the "CA-SIS 6/00" simulations, EDF manually extrapolated the performance data and input those data to CA-SIS again to obtain new results. Figure 3-7 indicates that for some cases, this had no effect. For other cases (e.g., E100 and E130), COP varied such that there was still disagreement. Because the "CA-SIS 6/00" runs also included other adjustments (see above), it is difficult to say if this change was solely responsible for the variation.

For the "CA-SIS 10/00" results, the manual extrapolation was improved to include the difference in behavior between dry coil and wet coil conditions on the performance map. For this set of results, the CA-SIS COPs give much better agreement with the analytical solution results. Because the first try at manual extrapolation may have been errant, and because EDF did not present individual sensitivity tests of each change in its modeler report, it is difficult to state exact quantitative effects of performing extrapolations in the CA-SIS results. However as shown in the CLIM2000 modeler report, EDF's CLIM2000 results for E110 and E100 indicate that 7%–10% error in total energy consumption is possible when small extrapolations of the performance data are not performed.

3.4.7 EnergyPlus (GARD Analytics)

EnergyPlus is the program recently released by DOE, and is the building energy simulation program that will be supported by DOE. GARD Analytics (GARD) used EnergyPlus's "Window Air-Conditioner" system for its model.

GARD submitted eight iterations of simulation results. Table 3-4 describes input file and software modifications for each iteration; a single results set was submitted corresponding to changes described in each row of the table. Version Beta 5-07 was used for the initial results set.

Table 3-4. Summary of EnergyPlus Changes That Were Implemented

Version	Input File Changes	Code Changes
Beta 5-07		
Beta 5-12 through Beta 5-14		• DX coil calculations modified to account for cycling • Modified method of calculating SHR and coil bypass factor
Beta 5-15 through Beta 5-18	• Changed DX (direct expansion) coil object names	• Changed name of DX coil object to better represent its algorithmic basis (no impact on results)
Ver 1-01 through Ver 1-11	• Changed from blow-through to draw-through fan configuration	• Changed to double precision • Modified method of calculating coil outlet conditions • Added draw-through fan option to Window Air-Conditioner model
Ver 1-12 through Ver 1-14	• New equipment performance curves • Adjusted fan mass flow and efficiency to achieve desired mass flow and fan power	
Ver 1-15 through Ver 1-17	• Went back to specified values for fan mass flow and efficiency	• Partial implementation of moist air specific heat • Fan power calculated using a standard initial density for volume to mass flow conversion
Ver 1-18 through Ver 1-19	• Changed basis of CDF curve from net to gross • Opened up minimum/maximum limits for performance curves	• Complete implementation of moist air specific heat • Heat of vaporization (h_{fg}) calculation modified for latent loads
Ver 1-20 through Ver 1-23	• Went back to original CDF curve (modified curve used with Version 1-19 was incorrect) • Changed from FAN:SIMPLE:CONSTVOLUME to FAN:SIMPLE:ONOFF • Used CDF curve for fan power to account for cycling	• Implemented optional PLR curve for fan cycling • Changed moisture initializations to use outdoor humidity ratio

The COP results for selected cases are summarized in Figure 3-9 for the EnergyPlus simulations and the TUD/HTAL analytical solution results. Note that the differences in results have been magnified in this figure by increasing the minimum value on the y-axis. Figure 3-10 includes specific results used to diagnose the causes of various disagreements. For the initial run with Beta 5-07, a number of disagreements with the analytical solution results were identified:

- Low indoor fan electrical power and fan heat; see Figure 3-10 results labeled "E170 Q ID Fan x 50"
- Reported cooling coil loads apparently not adjusted for part load cycling (although actual load removed from the zone appears to have been adjusted); see Figure 3-10 results labeled "E140 Q Coil Total"
- Sensible coil load about 1% higher than total coil load in the dry-coil cases.

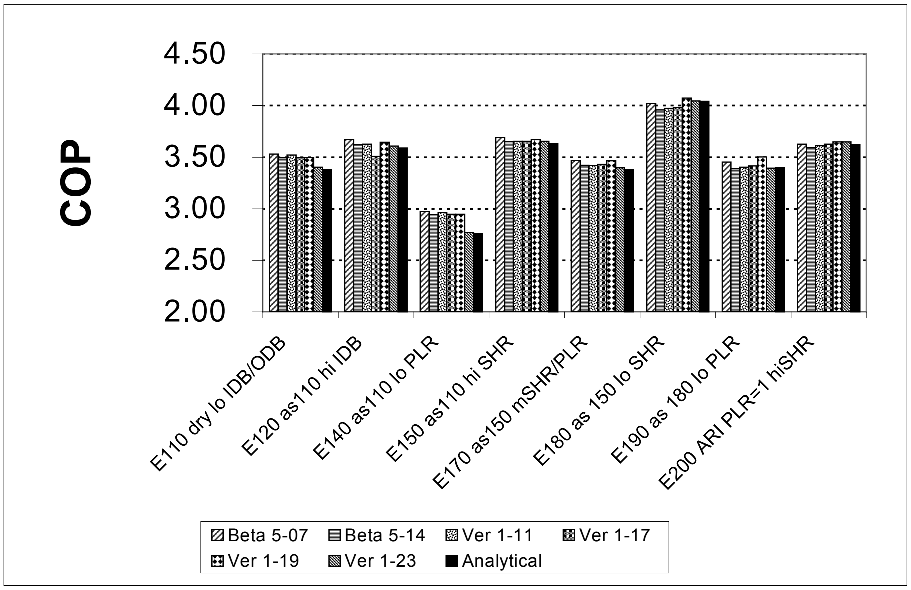

Figure 3-9. COP improvements to EnergyPlus models

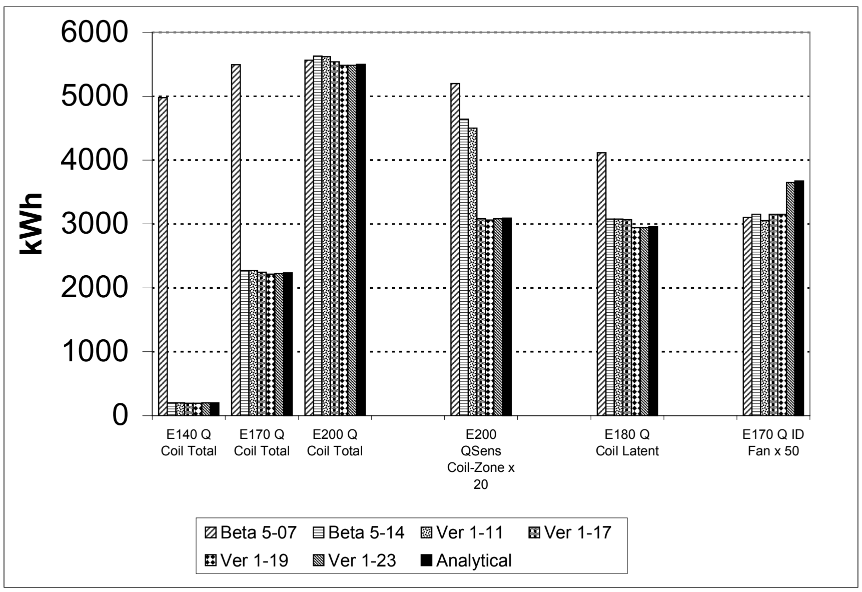

Figure 3-10. EnergyPlus fixes: various results

The process of correcting these disagreements engendered the improvements to EnergyPlus described below.

3.4.7.1 Reported Cooling Coil Loads Not Adjusted for Part Load Operation (up to 2500% effect on total coil load, negligible effect on energy consumption)

In Figure 3-10 it is apparent from the Beta 5-07 results for total coil load (designated by the results labeled "Q Coil Total" for cases E140, E170, and E200) that the total cooling coil load is in error, with the greatest error found in cases with lower PLR. For Beta 5-14 the reporting of cooling coil loads was corrected to account for run time during cycling operation. Because for Beta 5-07 the actual load extracted from the space was already being adjusted for cycling (similar magnitude disagreements do not exist for COP of cases E140 and E170 in Figure 3-9), it appears that this problem had a negligible effect on COP and energy consumption.

3.4.7.2 Modified Calculation of SHR and BF (1%–2% total consumption effect)

The problem of sensible coil loads being greater than total coil loads was addressed by modifying the methods of calculating SHR and BF. With the reasonable assumption that the coil load reporting error had negligible effect on energy consumption, the difference in COP between Beta 5-14 and Beta 5-07 (shown in Figure 3-9) illustrates the 1%–2% energy consumption effect of this modification, with a similar degree of change for all cases.

Along with the remaining differences in COP that are apparent from Figure 3-9 for Beta 5-14, GARD noted a number of other disagreements that were previously masked:

- Total coil loads were generally greater than for the analytical solutions (see "E200 Q Coil Total" results in Figure 3-10), and were 1%–2% greater than the sums of total zone load plus fan energy consumption
- The mean IDB for E200 moved from 26.7°C (good) to 27.1°C (high)
- Previous Beta 5-07 disagreements in terms of low ID fan power remain (see "E170 Q ID Fan × 50" results in Figure 3-10).

These disagreements with the analytical solutions prompted further improvements, described below.

3.4.7.3 Draw-through Fan, Double-Precision Variables, and Modified Calculation of Coil Outlet Conditions (0.1%–0.7% total consumption effect)

Changes leading up to Version 1-11 included:

- Modified method for calculating coil outlet conditions
- Use of double precision throughout EnergyPlus (this change was prompted by other issues not related to HVAC BESTEST)
- Addition of draw-through fan option to the window air-conditioner system.

Unfortunately, the effects of each of these changes were not disaggregated in the testing. The combined effects of these changes are illustrated in Figure 3-10, where the results for Beta 5-14 are compared to those from Ver 1-11 for the set of results labeled "E200 Qsens Coil-Zone × 20" (the difference between sensible coil loads and sensible zone loads, magnified by a factor of 20). This set of results indicates a 5% change in the loads-based calculated fan heat. The overall effect of these changes on COP (and consumption) is <1% as illustrated in Figure 3-9 comparing the difference between results of Ver 1-11 and Beta 5-14.

Along with remaining differences in COP apparent for Ver 1-11 in Figure 3-9, GARD noted other remaining disagreements:

- Total coil load remained 1%–2% greater than total zone load plus fan heat; similarly, the latent coil loads were 3% greater than for the analytical solution results—see "E180 Q Coil Latent" results in Figure 3-10
- The mean IDB for E200 moved from 27.1°C (high) to 27.5°C (higher)
- Previous Beta 5-07 disagreements of low ID fan power became worse compared with analytical solution results (see "E170 Q ID Fan × 50" results in Figure 3-10).

3.4.7.4 Change to Standard Air Density for Fan Power Calculation (1% decrease in sensible coil load)

For versions 1-12 through 1-17, results for changes to the software were aggregated with input file changes (notably the revision of system performance curves) so that assessing the effect of software revisions—including the implementation of moist air specific heat and the use of standard air properties for calculating supply air mass flow rates—was difficult. However, in Figure 3-10 (for the set of results labeled "E200

Qsens Coil-Zone × 20" for Ver 1-17 versus Ver 1-11), the bulk of the remaining fan heat discrepancy appears to have been addressed in version 1-17 when the fan power calculation was changed to incorporate standard air density. This change appears to have resulted in a 1% change in sensible and total coil load (see results for "E200 Q Coil Total" in Figure 3-10). The effect on ID fan energy appears to be about 3% (see results for "E170 ID Fan Q × 50" for Ver 1-17 versus Ver 1-11 in Figure 3-10), which translates to a 0.3% total power effect. The total electricity consumption effect would be greater in cases where the fan is running continuously (e.g. because of outside air requirements) even though the compressor is operating at lower part loads.

3.4.7.5 Modified Heat of Vaporization for Converting Zone Latent Load into HVAC System Latent Load (0.4%–2.5% total consumption effect for wet coil cases only)

For versions 1-18 and 1-19, the effects of input file changes were likely negligible (CDF curve revision), or the changes may have only affected specific cases. Enabling extrapolation of performance curves appears to have had the greatest effect in E120—see Figure 3-9 results for E120, Ver 1-19 versus Ver 1-17. Therefore, changes in results for the wet coil cases are likely caused primarily by changes to the software. Versions 1-18 through 1-19 include the following changes to the software:

- Changed heat of vaporization (h_{fg}) used for converting a zone latent load into a coil load
- Changed airside HVAC-model specific heat (c_p) from dry air to moist air basis.

From Figure 3-10, the case E180 latent coil load results (designated by "E180 Q Coil Latent") for Ver 1-19 versus Ver 1-17 indicate that the fixes to the software improved the latent coil load results, with a 4% effect on latent coil load for E180 and the other wet coil cases (not shown here). In Figure 3-9, the difference between Ver 1-19 and Ver 1-17 illustrates the effect on COP, with the greatest effect (2.2%–2.5%) seen for cases with the lowest SHR (e.g., cases E180 and E190). GARD also noted that changing the airside HVAC-model specific heat (c_p) from a dry air to a moist air basis improved consistency between coil and zone loads and removed other small discrepancies.

3.4.7.6 ID Fan Power Did Not Include COP f(PLR) Degradation (2% total consumption effect at mid PLR)

In Figure 3-10, using the set of results labeled "E170 Q ID Fan × 50" (fan energy use magnified by a factor of 50), it is apparent that indoor fan consumption was about 15% lower than the analytical solution results for case E170. This difference was traced to CDF not being accounted for in the ID fan consumption. Application of COP=f(PLR) was implemented by Ver 1-23, and better agreement with the analytical solution indoor fan energy consumption was the result. The difference in results for Ver 1-23 and Ver 1-19 in Figure 3-9 indicates a 2% effect on total energy consumption for the mid-PLR case E170, with a higher percentage of effect as PLR decreases (e.g., see Figure 3-9 results for case E140 or E190).

3.4.7.7 General Comment About Improvements to EnergyPlus

Each individual error found in EnergyPlus by itself did not have >3% effect on consumption results. However, these multiple errors do not necessarily compensate each other, and may be cumulative in some cases. Furthermore, some errors that have small effect on total consumption for these cases (e.g. fan model errors when the indoor fan is cycling with the compressor) could have larger effects on total consumption for other cases (e.g., if the indoor fan is operating continuously while the compressor cycles). Therefore, correcting these errors was important.

3.4.8 PROMETHEUS (KST)

PROMETHEUS is a whole-building hourly simulation program developed and used by Klimasystemtechnik (KST) of Berlin, Germany, for the company's energy and design consulting work.

During September 1998, the PROMETHEUS modelers reported that the CDF was externally applied to PROMETHEUS. The significance of CDF to the simulation results is documented in the CLIM2000 results summary (presented earlier in this section). As a result of this work the code authors planned to implement a COP=f(PLR) algorithm into PROMETHEUS (Behne 1998). However, because KST was unable to participate in the project after January 2000, it was unable to complete the refinement of its simulation results.

Results, Conclusions, and Recommendations

3.5 Interpretation of Results

The tables and graphs in Part IV present the final results from all the simulation programs and analytical solutions used in this study. Because the analytical solution results constitute a reliable set of theoretical results (a mathematical truth standard), the primary purpose of including simulation results for the E100–E200 cases in Part IV is to allow simulationists to compare their relative agreement (or disagreement) with the analytical solution results. Perfect agreement among simulations and analytical solutions is not necessarily expected. The Part IV results give an indication of what sort of agreement is possible between simulation results and the analytical solution results.

As we have explained previously, the analytical solution results do not represent absolute truth; instead, they represent a mathematical truth standard for cases E100–E200. Given the underlying physical assumptions in the case definitions, the analytical solutions are a mathematically provable and deterministic result for each case. In this context the underlying physical assumptions about the mechanical equipment as defined in cases E100–E200 are representative of typical manufacturer data, normally used by building design practitioners, with which many whole-building simulation programs are designed to work.

It is important to reiterate the difference between a mathematical truth standard and an absolute truth standard. In the former we accept the given underlying physical assumptions while recognizing that these assumptions represent a simplification of physical reality. The ultimate or absolute validation standard would be comparison of simulation results with a *perfectly performed* empirical experiment, the inputs for which are *perfectly specified* to the simulationists. In reality an experiment is performed and the experimental object specified within some acceptable band of uncertainty. Such experiments are possible but fairly expensive. We recommend that a set of empirical validation experiments be developed for future work.

One must rely on engineering judgment to assess the significance of results that disagree. For simulation results that disagree significantly with the analytical solution 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 task has indicated that when programs show disagreement with analytical solution results, other simulation results, or both, we often find a bug or a questionable algorithm.

As expected, because of iterative correction of input errors, software bugs, and clarification of the test specification, the quality of simulation results improved with each iteration of the field trials. Improvements to the simulation programs are evident when the initial results set is compared to the current results set. Initial simulation results for COP obtained after the first round of simulations, before the analytical solutions were developed, are shown in Figure 3-11 (abbreviations along this graph's x-axis are shorthand for the case descriptions given in Part I). These results indicate that there was initially 2%–30% *average* disagreement versus the mean of the simulated COP results. Corresponding disagreement of energy consumption results ranged from 4%–40%.

Figure 3-12 includes the current set of COP results for all the simulations and analytical solutions; abbreviations along this figure's x-axis are the same as in Figure 3-11. After correcting software errors using HVAC BESTEST diagnostics, the mean results of COP and total energy consumption for the programs are, on average, within <1% of the analytical solution results, with average variations of up to 2% for the low PLR dry coil cases (E130 and E140).

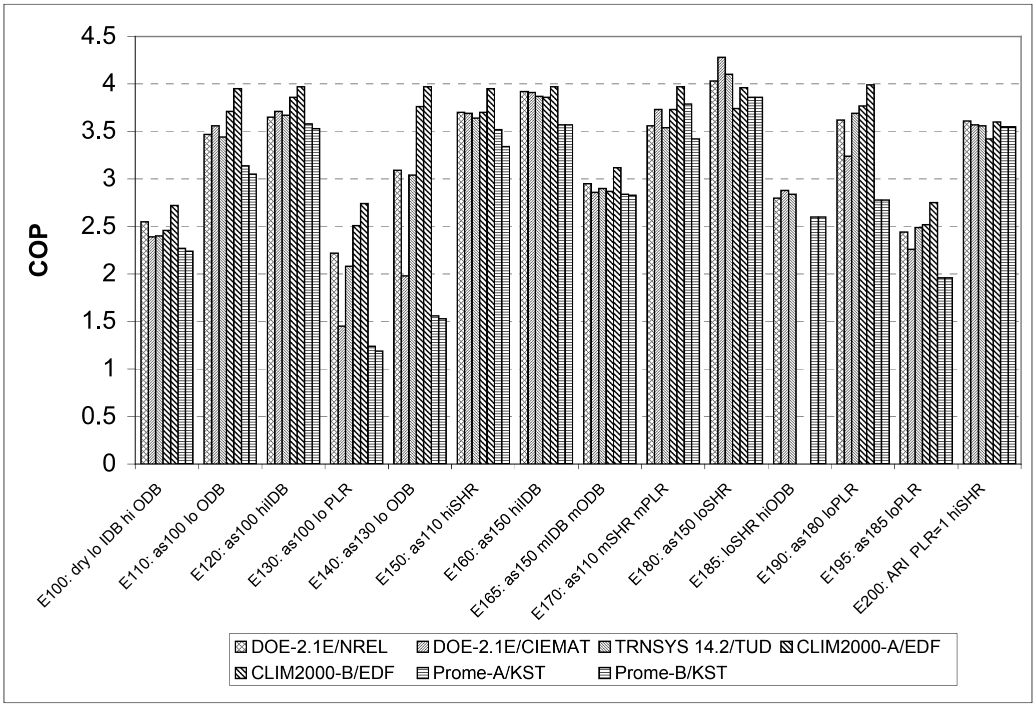

Figure 3-11. HVAC BESTEST—mean COP, before BESTESTing

(Abbreviations along the x-axis are shorthand for the case descriptions; see Part I for full case descriptions.)

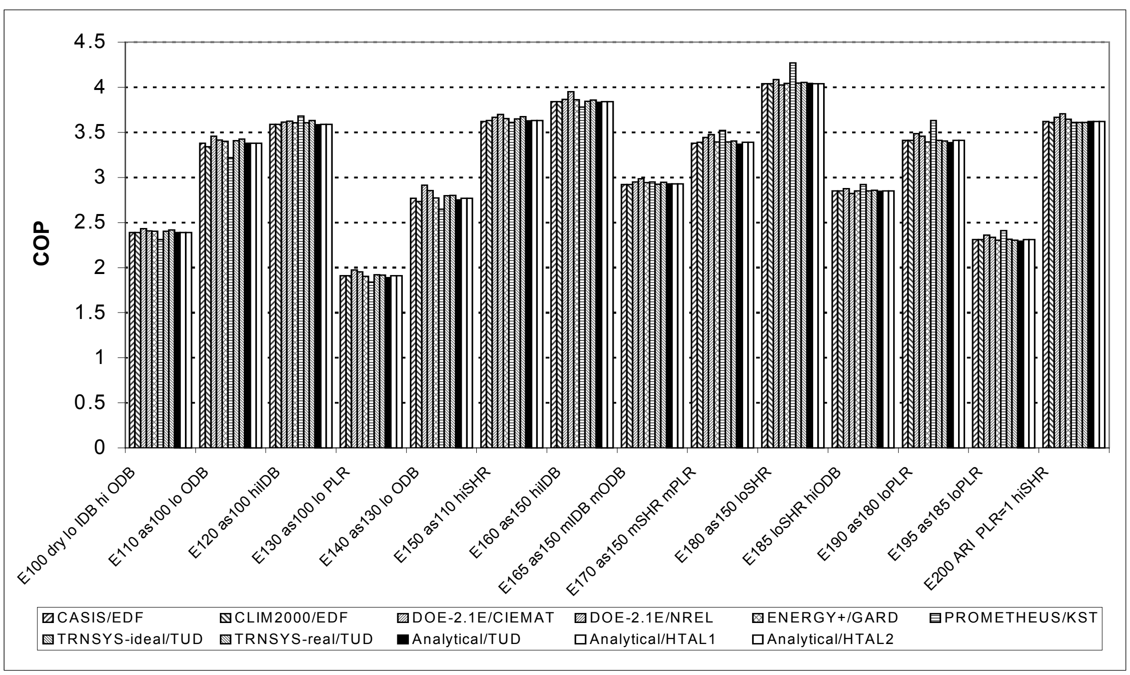

Figure 3-12. HVAC BESTEST—mean COP, after BESTESTing

(Abbreviations along the x-axis are shorthand for the case descriptions; see Part I for full case descriptions.)

Ranges of disagreement are further summarized in Table 3-5 for predictions of various outputs. This range of disagreement for each case is based on the difference between each simulation result versus the mean of the analytical solution results, divided by the mean of the analytical solution results. The outputs are disaggregated for dry coil performance (no dehumidification) and for wet coil performance (dehumidification moisture condensing on the coil). This summary excludes results for the PROMETHEUS participants; although they suspected an error(s) in their software, they were unable to complete the project.

Table 3-5. Ranges of Disagreement[a] among Simulation Results

Cases	Dry Coil (E100-E140)	Wet Coil (E150-E200)
COP and Total Electric Consumption	0% - 6%[a]	0% - 3%[a]
Zone Humidity Ratio	0% - 11%[a]	0% - 7%[a]
Zone Temperature	0.0°C - 0.7°C (0.1°C)[b]	0.0°C - 0.5°C (0.0°C – 0.1°C)[b]

[a] Percentage disagreement for each case is based on the difference between each simulation result (excluding PROMETHEUS) versus the mean of the analytical solution results, divided by the mean of the analytical solution results.
[b] Excludes results for TRNSYS-TUD with realistic controller.

The higher level of disagreement in the dry coil cases, which occurs for the case with the lowest PLR, is related to some potential problems that have been documented for DOE-2.1E (ESTSC version 088 and JJH version 133) in both the CIEMAT and NREL results (see Section 3.4). The larger disagreements for zone humidity ratio were caused by disagreements for the CLIM2000 and DOE-2.1E/CIEMAT results. The disagreement in zone temperature results is primarily from the TRNSYS-TUD results, where a realistic controller was applied on a short timestep (36 s); all other simulation results applied ideal control.

Based on results after "HVAC BESTESTing," the programs appear reliable for performance-map modeling of space cooling equipment when the equipment is operating close to design conditions. In the future, HVAC BESTEST cases may explore modeling at "off-design" conditions and the effects of using more realistic control schemes.

3.5.1 Test Cases for Future Work

In the course of this work it became apparent from comments of the participants and observations of the test specification authors that some additional test cases would be useful for improved error trapping and diagnostics with HVAC BESTEST. In future work, we suggest that some extensions to the test suite, outlined in the sections that follow, be considered.

3.5.1.1 Mechanical Equipment

Completion of the proposed E300-series cases (Neymark and Judkoff 2001) is the highest priority. These cases will not be possible to solve analytically, but will be developed as comparative test cases. They are dynamic test cases that utilize unrevised dynamic annual site weather data. They also help to scale the significance of disagreements that are less obvious with steady-state test cases. The cases test the ability to model the following:

- Quasi-steady-state performance using dynamic boundary conditions: dynamic internal gains loading, and actual (dynamic) typical meteorological year 2 (TMY2) weather data

- Latent loading from infiltration
- Outside air mixing
- Periods of operation away from typical design conditions
- Thermostat setup (dynamic operating schedule)
- Undersized system performance
- Economizer with a variety of control schemes
- Variation of PLR (using dynamic weather data)
- ODB and EDB performance sensitivities (using dynamic loading and dynamic weather data).

The proposed E300-series cases also address the important question of the ability of simulation software to model equipment performance at off-design conditions. These cases include a set of expanded performance data (beyond what is normally provided with typical manufacturer catalog data) and may include a test of the ability to extrapolate from a set of typical manufacturer catalog performance data.

It would also be interesting to add cases with more realistic control schemes. Such cases could include:

- Five-minute minimum on/off or hysteresis control, or both. Preliminary work by TUD documented in the TRNSYS-TUD modeler report suggests that it might be interesting to try:
    - Case E140 with 5-min minimum on and 5-min minimum off
    - Case E130 with 2°C hysteresis
    - Five-minute minimum off (a common manufacturer setting)
    - Combination of minimum on/off and hysteresis
    - Proportional control
    - Adding equipment run time to outputs
- Variation of part load performance based on more detailed data.

Other cases that are either under development or being considered for development as part of IEA SHC Task 22 involve:

- Heating equipment such as furnaces, heat pumps
- Radiant floor slabs for heating and cooling.

Additional possible cases include:

- Variable-air volume fan performance and control
- Outside dew point temperature (humidity ratio) effect on performance (see the DOE-2.1E/NREL modeler report [Appendix III-A])
- Repeat one or two of the E100–E200 series cases using expanded performance data
- Fan heat testing using continuous fan operation at low compressor part load
- PLR testing using ARI conditions for ODB, EWB, and EDB
- Combination of mechanical equipment tests with a realistic building envelope (although combining these adds noise, which makes diagnostics more difficult).

Obtaining additional simulation results would also be useful. Possible additional programs to test include: ESP, FSEC 3.0, HVACSIM+, the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) HVAC2 Toolkit, and others.

For the longer term, there has been discussion of trying to gather data that would allow highly detailed equivalent primary-loop component models of, for example, compressors, condensers, evaporators, and expansion valves, to be incorporated into the test specification. Incorporating and verifying data for such models to enhance the current HVAC BESTEST specification is expected to be a major effort. Additional long-term work would also include:

- Thermal storage equipment
- Air-to-air heat exchanger
- More complex systems associated with larger buildings including:
  - Large chillers
  - Chilled water loops
  - Cooling towers, and related circulation loops
  - More complex air handling systems
  - Other "plant" equipment
- ASHRAE RP-865 air-side analytical test field trials.

3.5.1.2 Envelope

IEA SHC Task 22 is updating reference results for existing IEA envelope BESTEST (Judkoff and Neymark 1995a) and HERS BESTEST (Judkoff and Neymark 1995b) cases with ground-coupled heat transfer through floor slabs and basement walls. Task 22 is also considering development of cases to test the ability to model multizone envelope heat transfer (Kataja and Kalema 1993).

Based on comments about this project and previous work, we can identify a number of other interesting areas relating to envelope models for which BESTEST cases could be developed. These include:

- Expanded ground coupling cases
- Expanded infiltration tests (e.g., testing algorithms that vary infiltration with wind speed)
- Varying radiant fraction of heat sources
- Moisture adsorption/desorption
- Daylighting controls.

ASHRAE has developed a series of envelope analytical tests under RP-1052 (Spitler, Rees, and Dongyi 2001). Future work could also include field trials of these tests.

Finally, the current IEA BESTEST envelope tests should be updated periodically to include:

- New simulation results for the current set of programs and also other simulation results from, for example, APACHE, CA-SIS, CLIM2000, EnergyPlus, ESP, SUNREL, TRNSYS
- Application of updated TMY2 weather data
- Additional radiative exchange tests (see Table 2-51 of IEA BESTEST, Judkoff and Neymark 1995a).

3.6 Conclusions and Recommendations

3.6.1 Conclusions

Additional test cases for mechanical space cooling equipment have been added to IEA's existing method for systematically comparing whole-building energy software and determining the algorithms responsible for prediction differences. Similar to previous test suites that applied the HVAC BESTEST method, these new cases have a variety of uses, including:

- Comparing output from building energy simulation programs to a set of analytical solutions that constitutes a reliable set of theoretical results given the underlying physical assumptions in the case definitions
- Comparing several building energy simulation programs to determine the degree of disagreement among them
- Diagnosing the algorithmic sources of prediction differences among several building energy simulation programs
- Comparing predictions from other building energy programs to the analytical solutions and simulation results in this report
- 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.

The previous IEA BESTEST envelope test cases (Judkoff and Neymark 1995a) have been code-language-adapted and formally approved by ANSI and ASHRAE as a Standard Method of Test, ANSI/ASHRAE Standard 140-2001. The BESTEST procedures are also being used as teaching tools for simulation courses at universities in the United States and Europe.

Adding mechanical equipment tests to the existing envelope tests gives building energy software developers and users an expanded ability to test a program for reasonableness of results and to determine if a program is appropriate for a particular application. The current set of steady-state tests (cases E100–E200) represents the beginning of work in this area. A set of additional cases has been proposed; these new (E300-series) cases are briefly described in Section 3.5.1, where additional cases for future consideration beyond the E300 series are also discussed.

Part II of this report includes analytical solution results for all the cases. Because the analytical solution results constitute a reliable theoretical solution and the range of disagreement among the analytical solutions is very narrow compared to the range of disagreement among the simulation results, the existence of the analytical solutions improves the diagnostic capability of the cases. This means that a disagreeing simulation result for a given test implies a stronger possibility that there is an algorithmic problem, coding error, or input error than when results are compared only with other simulation programs.

The procedure has been field-tested using a number of building energy simulation programs from the United States and Europe. The method has proven effective at isolating the sources of predictive differences. The diagnostic procedures revealed bugs, faulty algorithms, limitations, and input errors in every one of the building energy computer programs tested in this study—CA-SIS, CLIM2000, DOE-2.1E, ENERGYPLUS, PROMETHEUS, and TRNSYS-TUD. Table 3-6 summarizes the notable examples.

Many of the errors listed in Table 3-6 were significant, with up to 50% effect on total consumption or COP for some cases. In other cases for individual programs, some errors had relatively minor (<2%) effect on total consumption. However, where a program had multiple errors of smaller magnitude, such errors did not necessarily compensate each other, and may have been cumulative in some cases. Furthermore, some errors that have small effect on total consumption for these cases (e.g., a fan heat calculation error when the indoor fan is cycling with the compressor), could have larger effects for other cases not included with the current tests, but planned for later tests (e.g. if the indoor fan were operating continuously, independent of compressor cycling). Therefore, correcting the minor errors as well as the major errors was important.

Table 3-6. Summary of Software Problems Found Using HVAC BESTEST

Software	Error Description[a]	% Disagreement[a,b]	Resolution
CA-SIS	No extrapolation of performance data allowed	Possibly up to 10% power[c] (E110, E100)	Manually fixed[d]
CA-SIS	Convergence algorithm problem	E200 would not run	Fixed
CA-SIS	Fan heat not added to coil load	4% sensible coil load ([4% power[c] f(SHR)[e])	Fixed
CA-SIS	Indoor and outdoor fan power not f(PLR)	2% power[c] (mid PLR)	Fixed
CLIM2000	Verify new model improvements[f]	Up to 50% COP change from earlier model	Verified improved
CLIM2000	Compressor and fan powers exclude COP = f(PLR) degradation	20% power[c] (low PLR) 13% power[c] (mid PLR)	Fixed
CLIM2000	Performance map extrapolation problem	9% power[c] (E110)	Fixed
DOE-2.1E (JJH ver < W54)	Minimum supply temperature coding error in early RESYS2[g] system	36% COP (earlier base case[h])	Fixed
DOE-2.1E (JJH ver 133)	Coil-Zone load difference disagrees with fan power for RESYS2[g] at low SHR.	5% sensible coil load (at low SHR)	Authors notified
DOE-2.1E (JJH ver 133)	Bypass factor = f(EWB, ODB) insensitivity	1% power[c] (E185)	Authors notified
DOE-2.1E (JJH 133, ESTSC 088)	Indoor fan power not f(PLR) (RESYS2 and PTAC)[g]	2% total power (at mid PLR)	Authors notified
DOE-2.1E (ESTSC v 088)	Coil-Zone load difference disagrees with fan power for PTAC[g] at low SHR.	2% sensible coil load (at low SHR)	Authors notified
ENERGYPLUS	Reported coil loads not f(PLR)	Up to 2500% coil load (at low PLR); 0% power[c]	Fixed
ENERGYPLUS	Calculation of SHR and bypass factor	1-2% power[c]	Fixed
ENERGYPLUS	Heat of vaporization for latent coil calculation	0.4-2.5% power[c]	Fixed
ENERGYPLUS	Indoor fan power not f(PLR)	2% power[c] (mid PLR)	Fixed
PROMETHEUS	Compressor COP = f(PLR) calculated in external post-processor to software	20% power[c] (low PLR) if no post-processor calc.	Authors notified
TRNSYS-TUD (realistic control)	Use of some single precision variables	45% power[c] (lo PLR) 14% power[c] (midPLR)	Fixed
TRNSYS-TUD (realistic control)	Wrong data compiled for coil latent load output.	4% power[c] (E150) 3% latent coil load (E170)	Fixed

a Acronyms used in this column are described in the Nomenclature.
b Specific cases or conditions relevant to the described disagreement are included in parenthesis.
c Total system power
d Current results include non-automated version of the fix.
e Percentage disagreement is greatest at high SHR and decreases as SHR decreases.
f The software authors used HVAC BESTEST to document the improvement of their new model relative to a previous model that they were in the process of replacing when IEA SHC Task 22 began.
g In DOE-2.1E the RESYS2 system is for modeling typical residential equipment (e.g. a unitary split system), and PTAC is for modeling a packaged terminal air-conditioning system.
h This error was discovered using a preliminary version of HVAC BESTEST which had a base case different from E100.

Performance of the tests resulted in quality improvements to all the building energy simulation programs used in the field trials. Many of the errors found during the project stemmed from incorrect code implementation; however, there were also instances where no algorithm existed internal to a program to account for equipment performance degradation with decreasing part-load ratio (PLR). Some of these bugs may well have been present for many years. The fact that they have just now been uncovered shows the power of BESTEST and also suggests the importance of continuing to develop formalized validation methods. It is only after coding bugs have been eliminated that the assumptions and approximations in the algorithms can be evaluated where necessary.

Checking a building energy simulation program with HVAC BESTEST requires about 1 person-week for an experienced user. (This estimate is based on a poll of the participants, and does not include time for finding/fixing coding errors or revising documentation.) Because the simulation programs have taken many years to produce, HVAC BESTEST provides a very cost-effective way of testing them. As we continue to develop new test cases, we will adhere to the principle of parsimony so that the entire suite of BESTEST cases may be implemented by users within a reasonable time span.

After using HVAC BESTEST diagnostics to correct software errors, the mean results of COP and total energy consumption for the programs are, on average, within <1% of the analytical solution results, with average variations of up to 2% for the low PLR drycoil cases (E130 and E140). This summary excludes results for one of the participants, who suspected an error(s) in their software but was unable to complete the project. Based on results after HVAC BESTESTing, the programs appear reliable for performance-map modeling of space cooling equipment when the equipment is operating close to design conditions.

Manufacturers typically supply catalog equipment performance data for equipment selection at given design (near-peak) load conditions. Data for off-design conditions, which can commonly occur in buildings with outside air requirements or high internal gains, are not included. In practice, simulation tools often use data from the manufacturer to predict energy performance. In general, the current generation of programs appears most reliable when performance for zone air and ambient conditions that occur within the bounds of the given performance data is being modeled. However, preliminary sensitivity tests indicate significant differences in results when extrapolations of performance data are required. Additional cases have been proposed to explore simulation accuracy at off-design conditions (Neymark and Judkoff 2001). Those cases, which are in the field-trial process, include a set of expanded performance data beyond what is normally provided with typical manufacturer catalog data. Obtaining such expanded performance data required significant effort. We reviewed three equipment selection software packages typically used by HVAC engineers for specifying equipment. However, none of these were satisfactory for developing the range of data we desired, so the data we ultimately obtained was custom-generated by a manufacturer. In general for the state of the art in annual simulation of mechanical systems to improve, manufacturers need to either readily provide expanded data sets on the performance of their equipment, or improve existing equipment selection software to facilitate generation of such data sets.

Within the BESTEST structure, there is room to add new test cases when required. 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. Near-term continuing work (E300-series cases, not included in this report) is focused on expanding the mechanical space cooling cases to include:

- Dynamic performance using dynamic loading and actual (dynamic) TMY2 weather data
- Latent loading from infiltration
- Outside air mixing
- Periods of system operation away from typical design conditions
- Thermostat setup (dynamic operating schedule)
- Undersized system performance
- Economizer with a variety of control schemes
- Variation of PLR (using dynamic weather data)
- ODB and EDB performance sensitivities (using dynamic loading and weather data).

For the longer term we hope to add test cases that emphasize special modeling issues associated with more complex building types and HVAC systems as listed in Section 3.5.1.

3.6.2 Recommendations

The previous IEA BESTEST procedure (Judkoff and Neymark 1995a), developed in conjunction with IEA SHC Task 12, has been code-language-adapted and approved as a Standard Method of Test for evaluating building energy analysis computer programs (ANSI/ASHRAE Standard 140-2001). This method primarily tests envelope-modeling capabilities. We anticipate that after code-language adaptation, HVAC BESTEST will be added to that Standard Method of Test. In the United States, the National Association of State Energy Officials (NASEO) Residential Energy Services Network (RESNET) has also adopted HERS BESTEST (Judkoff and Neymark 1995b) as the basis for certifying software to be used for Home Energy Rating Systems under the association's national guidelines. The BESTEST procedures are also being used as teaching tools for simulation courses at universities in the United States and Europe. We hope that as the procedures become better known, developers will automatically run the tests as part of their normal in-house quality control efforts. The large number of requests (more than 800) that we have received for the envelope BESTEST reports indicates that this is beginning to happen. Developers should also include the test input and output files with their respective software packages to be used as part of the standard benchmarking process.

Clearly, there is 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, including:

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

Future work should therefore encompass:

- Continued production of a standard set of analytical tests
- Development of a set of diagnostic comparative tests that emphasize the modeling issues important in large commercial buildings, such as zoning and more tests for heating, ventilating, and air-conditioning systems
- Development of a sequentially ordered series of high-quality data sets for empirical validation.

Continued support of model development and validation activities is essential because occupied buildings are not amenable to classical controlled, repeatable experiments. The few buildings that are truly useful for empirical validation studies have been designed primarily as test facilities.

The energy, comfort, and lighting performance of buildings depend on the interactions among a large number of transfer mechanisms, components, and systems. Simulation is the only practical way to bring a systems integration problem of this magnitude within the grasp of designers. Greatly reducing the energy intensity of buildings through better design is possible with the use of simulation tools (Torcellini, Hayter, and Judkoff 1999). However, building energy simulation programs will not be widely used unless the design and engineering communities have confidence in these programs. Confidence and quality can best be encouraged by combining a rigorous development and validation effort with user-friendly interfaces, minimizing human error and effort.

Development and validation of whole-building energy simulation programs is one of the most important activities meriting the support of national energy research programs. The IEA Executive Committee for Solar Heating and Cooling should diligently consider what sort of future collaborations would best support this essential research area.

3.7 Acronyms for Part III

These are acronyms used in Sections 3.2 through 3.6.

ARI: Air-Conditioning and Refrigeration Institute

BF: Bypass Factor

CDF: COP Degradation Factor is a multiplier (≤1) applied to the full-load system COP. CDF is a function of PLR

CIEMAT: Centro de Investigaciones Energéticas, Medioambientales y Technologicas

COP: Coefficient of Performance, the ratio using same unites of the net refrigeration effect to the cooling energy consumption, where the net refrigeration effect is heat removed by the coil excluding the fan, heat, and cooling energy consumption is the site electricity consumption of the compressor, air distribution fan, condenser fan, and related auxiliaries.

DOE: U.S. Department of Energy

EDB: Entering Dry-Bulb temperature, the temperature of a thermometer would measure for air entering the evaporator coil

EDF: Electricité de France

EWB: Entering Wet-Bulb temperature, the temperature of a wet-bulb portion of a psychrometer would measure if exposed to air entering the evaporator coil

GARD: GARD Analytics

HTAL: Hochschule Tecknik+Architektur Luzern

ID: Indoor

IDB: Indoor Dry-Bulb temperature, the temperature a thermometer would measure if exposed to indoor air

IEA: International Energy Agency

KST: Klimasystemtechnik

NREL: National Renewable Energy Laboratory

OD: Outdoor

ODB: Outdoor Dry-Bulb temperature, the temperature a thermometer would measure if exposed to outdoor air; this is the temperature of the air entering the condenser coil

PLR: Part Load Ratio, the ratio of net refrigeration effect to adjusted net capacity where the net capacity is the gross total capacity of the system less the fan power (heat); gross total capacity is the rate of both sensible and latent heat removal by the cooling coil for a given set of operating conditions

SHC: Solar Heating and Cooling Programme (of the IEA)

SHR: Sensible Heat Ratio, is the ratio of sensible heat removal to total (sensible + latent) heat removal by the evaporator coil

TMY2: Typical Meteorological Year 2

TUD: Technische Universität Dresden

3.8 References for Part III

These are references for Sections 3.2 through 3.6. References for Appendix III are included at the end of each modeler report.

ANSI/ASHRAE Standard 140-2001. (2001). *Standard Method of Test for the Evaluation of Building Energy Analysis Computer Programs*. Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

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Judkoff, R.; Neymark, J. (1995a). *International Energy Agency Building Energy Simulation Test (IEA BESTEST) and Diagnostic Method*. NREL/TP-472-6231. Golden, CO: National Renewable Energy Laboratory.

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NREL. (2000). *HVAC BESTEST Summary of 5th Set of Results Cases E100 - E200.* Compiled by J. Neymark & Associates for NREL. Golden, CO: National Renewable Energy Laboratory.

Neymark J.; Judkoff, R. (1998–2000). Addenda to draft versions of *International Energy Agency Building Energy Simulation Test and Diagnostic Method for Mechanical Equipment (HVAC BESTEST)*. Golden, CO: National Renewable Energy Laboratory. These addenda include:

- "September 1998 Summary of Revisions to HVAC BESTEST (Sep 97 Version)," September 1998
- "May 1999 Summary of Comments and Related Clarifications to HVAC BESTEST, September 1998 and February 1999 User's Manuals," May 1999
- "Revisions to E100-E200 Series Cases," December 16, 1999
- "Re: Revisions to E100 Series Cases," e-mail from Neymark to participants January 3, 2000; 2:21 P.M.
- "E100-E200 series cases," e-mail from Neymark to participants January 3, 2000; 2:23 P.M.
- "HVAC BESTEST: E100-E200 two minor but important points," e-mail from Neymark to participants, January 10, 2000; 8:10 P.M.

All relevant information from these addenda was included in the final version of Part I.

Neymark J.; Judkoff, R. (2001). *International Energy Agency Building Energy Simulation Test and Diagnostic Method for Mechanical Equipment (HVAC BESTEST), Volume 2: E300, E400, E500 Series Cases.* Golden, CO: National Renewable Energy Laboratory. Draft. October 2001.

Spitler, J.; Rees, S.; Dongyi, X. (2001). *Development of an Analytical Verification Test Suite for Whole Building Energy Simulation Programs—Building Fabric.* Draft Final Report for ASHRAE 1052-RP. Stillwater, OK: Oklahoma State University School of Mechanical and Aerospace Engineering.

Torcellini, P.; Hayter, S.; Judkoff, R. (1999). *Low Energy Building Design: The Process and a Case Study. ASHRAE Transactions* 1999 105(2). Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

3.9 APPENDIX III: Simulation Modeler Reports

In Appendix III, we present reports written by the modeler(s) 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 program
- Bugs, faulty algorithms, documentation problems, or input errors uncovered using the HVAC BESTEST diagnostics
- Source code or input modifications made because of the diagnostic results
- Comments on agreement or disagreement of results compared to 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 programs, HVAC BESTEST, or both.

Modelers also filled out a pro-forma description that defines many of the algorithms within their programs. The pro-forma description is presented at the end of each individual modeler report. These pro-forma reports, which appear as they were submitted with minimal reformatting and editing, are also presented here.

Appendix III-A

DOE-2.1E

National Renewable Energy Laboratory/J. Neymark & Associates

United States

January 25, 2001

1. Introduction

Software: DOE-2.1E J.J. Hirsch version 133

Authoring Organization: Lawrence Berkeley National Laboratory & Los Alamos National Laboratory & J.J. Hirsch & Associates

Authoring Country: USA

Modeling Methodology:

For modeling unitary direct expansion cooling systems, DOE-2.1E uses performance maps based on manufacturer performance data. The user may either use the default performance maps supplied with the software, or generate custom performance maps based on published performance data for a specific unit.

Zone loads are calculated hourly, separately from the equipment performance. That is, DOE-2.1E first calculates just the loads for the entire simulation period assuming a constant zone temperature. Then it calculates the system performance using the previously calculated hourly zone loads for the entire simulation period, with an adjustment to the zone loads to account for the actual zone temperature provided by the mechanical system and its controls.

Overall performance for total capacity and efficiency is described as:

hourly property = (value at ARI rating conditions)*(multiplier f(EWB,ODB))*(multiplier f(PLR))

where:

ARI ≡ Air Conditioning and Refrigeration Institute

EWB ≡ Entering Wet-bulb Temperature

ODB ≡ Outdoor Dry-bulb Temperature

PLR ≡ Part Load Ratio, (Coil total energy removal)/(Gross Total Capacity)

The f(EWB,ODB) curve fits can be bilinear or biquadratic functions; the f(PLR) curve fits can be linear, quadratic or cubic. The sensible capacity also follows a similar function, however, to account for additional variation as a function of EDB, DOE-2 has a hardwired (not controllable by the user) assumption of:

$SCAP_T$ = ("COOL-SH-CAP")*("COOL-SH-FT"(EWB,ODB))-{1.08*CFM*(1.0-CBF)*(80-EDB)},

where:

$SCAP_T$ ≡ sensible capacity

CFM ≡ airflow rate at fan rating conditions

CBF ≡ coil bypass factor

(2.1A Reference Manual, p. IV.238).

Extrapolation of curve fits can be limited in DOE2, using either a cap on the dependent variable results, or a cap on ODB and EWB. The cap on EWB is hardwired as being EWB = ODB - 10. Bypass factor also includes variation as a function of fan speed.

DOE-2.1E automatically identifies when a dry coil condition has occurred and does calculations accordingly. F(EWB,ODB) curve fit data is meant for wet coils only. Where possible f(T) data points assume EDB = 80°F, however at lower EWB, it was necessary to use data for EDB < 80°F (and normalize that data to be consistent with EDB-80°F data) to give proper information to curve fit routines; this was true for sensible capacity and bypass factor performance maps, but not necessary for EIR or total capacity maps.

Initial zone air conditions (temperature and humidity) are the ambient conditions.

Ideal controls can be modeled. It is possible to account for minimum on/off times by adjusting f(PLR) curves. If needed the COOL-CLOSS-FPLR curve used by the RESYS2 also has a limits feature that allows for modeling minimum on/off system operating requirements.

The HVAC BESTEST user's manual does not give enough information for comparing component models (e.g., disaggregation of individual heat exchangers, etc.) so no attempt was made to check if specific equipment models in PLANT could have been applied to this work.

2. Modeling Assumptions

Some inputs must be calculated from the given data. For example fan power is described as kW/cfm. Such inputs are not noted below because they are a simple calculation directly from the inputs. Those inputs noted below are included either because they may be inferred from information in the test specification, or the DOE-2.1E algorithms use assumptions (e.g. specific heat of air) slightly different from the test specification

- FLOOR-WEIGHT=30: recommendation by DOE-2 for lightweight construction. This input is relatively unimportant for the near adiabatic envelope where conduction is already < 1% of total sensible internal gains for most cases (is about 5% in low PLR cases).
- THROTTLING-RANGE = 0.1: Minimum setting in DOE-2; exact ideal on/off control is not possible, some proportionality is required.
- MIN-SUPPLY-T = 46: This minimum supply temperature setting is low enough so that the supply air temperature is not limited by this input.
- SUPPLY-DELTA-T = 0.789 (is delT from fan heat): The DOE-2 Engineer's Manual indicates using moist air specific heat (ρ (cp) 60 = 1.10), therefore that factor was used for calculating this input.
- COIL-BF and related bypass factor f(EWB,ODB) curve: DOE-2's recommendation for calculating bypass factors uses dry air specific heat (ρ (cp) 60 = 1.08) rather than moist air specific heat given in the user's manual appendix. (2.1A Reference Manual, p. IV.247). The input decks therefore used the ρ (cp) 60 = 1.08 factor in determining leaving air conditions relevant to bypass factor calculations.
- COOL-FT-MIN = 65 (ODB extrapolation minimum that allows extrapolation down to EWB = 55)
- MIN-UNLOAD-RATIO = 1; no compressor unloading
- MIN-HGB-RATIO = 1; no hot gas bypass
- OUTDOOR-FAN-T = 45; default, limit below which fans do not run. At this setting the fans will always cycle on/off with the compressor for cases E100-E200.

3. Modeling Options

SYSTEM-TYPE: RESYS2 model

A number of SYSTEM-TYPEs are possible and reasonable for modeling the HVAC BESTEST DX system, including: RESYS2, RESYS, PSZ, and PTAC. Choice of system type affects: default performance curves and features available with the system.

Of these, according to a DOE-2 documentation supplement (21EDOC.DOC), neither PTAC nor RESYS have had the improved part load (cycling) model for packaged systems incorporated (this uses the COOL-CLOSS-FPLR curve rather than the COOL-EIR-FPLR curve). Additionally, RESYS2 has more peripheral feature options than RESYS (possibly useful later on), and one of the code authors (Hirsch) recommended the RESYS2 over the RESYS system type. Then either the PSZ or RESYS2 models could have worked since custom performance curves are applied. Therefore, RESYS2 was somewhat arbitrarily chosen because its default performance curves should be closer to the HVAC BESTEST performance data. When RESYS2 versus PSZ is used for SYSTEM-TYPE with HVAC BESTEST custom performance curves applied to both and no other changes between the models, sensitivity tests give results that are virtually identical (no variation more significant than to the 5th significant digit, and then only for some of outputs).

SYSTEM-FANS: SUPPLY-KW & SUPPLY-DELTA-T

SYSTEM-FANS includes two different possibilities for determining indoor distribution fan power and heat. This is by either providing values for:

- SUPPLY-KW and SUPPLY-DELTA-T (rated fan power and temperature rise due to fan heat)
- SUPPLY-STATIC, SUPPLY-EFF (rated static pressure and efficiency) and SUPPLY-MECH-EFF (mechanical efficiency, only relevant if fan motor outside air stream). When the motor is located in the air stream SUPPLY-STATIC and SUPPLY-EFF can also be the total pressure and efficiency, respectively.

Fans were modeled with SUPPLY-KW and SUPPLY-DELTA-T. However, SUPPLY-STATIC and SUPPLY-EFF could also have been used. Sensitivity test between these options using equivalent inputs o f HVAC BESTEST indicated no discernible variation in fan energy or compressor energy use, and < 0.01% effect on total coil load.

Disaggregation of indoor and outdoor fans.

The test cases indicate that compressor and fans all cycle on and off together. In developing the model with DOE-2, it is possible for the modeler to disaggregate the compressor, outdoor fan, and indoor fan, or aggregate the fans with the compressor model. When components are disaggregated, the outdoor fan sees the exact same part load adjustment as the compressor, using the COOL-CLOSS-FPLR curve to apply the COP Degradation Factor (CDF) indicated by the test specification. However, part load adjustment for the indoor fan does not include the CDF adjustment and is just a straight PLR multiplier. For stricter adherence to the requirement that compressor and fans cycles on and off together, indoor and outdoor fans could have been modeled as aggregated with the compressor. However, the DOE-2 documentation indicates that there is a better latent simulation with the indoor fan modeled separately from the compressor (2.1A User's Manual, p. IV.241). Additionally, disaggregation is better for diagnostic comparisons with the other results. Therefore, we decided to disaggregate the fans in the model. The effect on total energy use of the indoor fan not exactly cycling on/off with the compressor and outdoor fan is examined in Section 5.

4. Modeling Difficulties

Indoor fan is not exactly cycling on and off with compressor and outdoor fan

Indoor fan is not exactly cycling on and off with compressor and outdoor fan - see above. Also see Section 5 for more detail on the relatively minor energy consumption inaccuracy caused by this.

Apparent minor inconsistency with specific heat used for calculating SUPPLY-DELTA-T and COIL-BF inputs

There is an apparent minor inconsistency with specific heat of air used for calculating COIL-BF inputs versus SUPPLY-AIR-DT. Both equations utilize the equation:

$q = m\ (cp)\ \Delta T$, where since the flow rate is given

$m = \rho * Q * 60$, where Q is volumetric fan air flow rate in cfm.

So that:

$q = (\rho\ (cp)\ 60) * Q * \Delta T$.

For developing inputs the term K = "ρ (cp) 60" is used. In general for standard air $\rho = 0.075$ lb/ft3. For dry air cp = 0.24 Btu/lb°F resulting in K = 1.08 and for moist air w $\approx$ 0.01 so that cp = 0.244 Btu/lb°F resulting in K = 1.10, where K has units of (Btu*min)/(ft^3*F*h). Also note that some references indicate that standard air is dry (e.g. ASHRAE Terminology, Howell et al) while others only specify the density but indicate the possibility that standard air can be moist (e.g. ANSI/ASHRAE 51-1985).

For initially calculating SUPPLY-DELTA-T at standard air conditions, the DOE-2.1A Engineers Manual (p. IV.29) uses moist (w=0.01) air (K = 1.10). However, for calculating the COIL-BF input (and data points for COIL-BF-FT), the DOE-2.1A User's Manual, p. IV.247 indicates K = 1.08. Note that an HVAC text published by ASHRAE (Howell et al, p. 3.5) uses K = 1.10 for calculating leaving air conditions using volumetric air flow rates.

This is a minor inconsistency within DOE-2 that is not surprising given ambiguities in the general literature. However, it seems that if K = 1.10 is used for SUPPLY-DELTA-T, then also assuming moist air cp for calculating leaving air conditions in determining bypass factors should be used, when giving advice for user inputs. (Note the DOE-2 Engineer's Manual indicates that DOE-2 actually adjusts ρ*cp for actual entering conditions in the hourly calculations; so perhaps it would be appropriate to advise users to base BF inputs on actual entering ρ*cp in the DOE-2 User's Manual?)

It is recommended that all these DOE-2 input requirements be made consistent with each other such that the methods for calculating the inputs are clearer to the user. An example of a good instruction *format* is the COIL-BF input discussion in the DOE-2 User's Manual (2.1A, p. IV.247).

5. Software Errors Discovered and/or Comparison Between Different Versions of the Same Software

This section documents one major problem and several minor issues. Regarding minor issues, they may seem small individually, but when summed up, they can make a difference in the aggregate results. Additionally, this documents the rigor associated with developing input decks that are as consistent as possible with the test specification, and therefore appropriate for BESTEST reference results for later (E300-series) cases when analytical verifications will not be possible.

Fixed Bug Resulting from this Work

Minimum Supply Temperature Bug in RESYS2 (36% issue)

In the earliest stage of the HVAC BESTEST test specification development and testing (ca. 8/94 prior to the beginning of IEA SHC Task 22), a problem surfaced for the RESYS2 system in a version prior to DOE-2.1E W-54. Identical input decks (for a much different preliminary version of the test specification) were used with the only difference between the input decks being the designation of SYSTEM-TYPE as PSZ versus RESYS2. The results obtained are shown in Table N1.

Table N1: RESYS2 versus PSZ for DOE-2.1E before Version W54

DOE-2 System	Compr+ODfan Elec. (kWh)	Coil Clg. (kBtu)	Latent Clg. (kBtu)	EER (Coil/Elec)
PSZ	2,587	25,700	4,100	9.8
RESYS2	1,646	26,500	5,200	16.1

In response to this 36% consumption difference and the unreasonably high EER for the RESYS2 result, one of the code authors explained that that they found a bug in the RESYS2 system model. In that model, when the indoor fan mode is set to INTERMITTENT, the capacity calculation that sets the minimum supply temperature (TCMIN) used the wrong value, resulting in the unrealistically high EER for the RESYS2. This coding error was corrected by the code authors (Hirsch 11/94).

Issues to Transmit to Code Authors

Fan heat discrepancy (2% issue at low SHR)

The current set of results for DOE-2.1E v133 indicates a discrepancy for fan heat calculated from:

Qfan = Qcoil,tot - Qcoil,lat - Qenv,sens

versus Qfanpower, where

Qcoil,tot ≡ total cooling energy extracted by the coil

Qcoil,lat ≡ latent cooling energy extracted by the coil

Qenv,sens ≡ sensible cooling energy extracted from the zone

Qfanpower ≡ fan electric energy consumption.

The differences in the fan heats for the various cases along with the percent effect on total coil load (and therefore total energy consumption) are listed in Table N2.

The following is apparent from the table:

- The greatest discrepancy occurs for the high humidity cases (E180-E195) with 40% overestimation of fan heat, which is nearly 2% of total coil load.
- For the other wet-coil cases with more commonly found SHRs, there is a 10-20% overestimation of fan heat, which is roughly 0.5% of total coil load.
- For the dry-coil cases the discrepancy seems insignificant, with only a 2% underestimation of fan heat, which is 0.1% of total coil load.

Table N2: Fan Heat Discrepancy

d:\e\iea22\hvabtest\results\doe2\21ejna2a.xls a:a63..k87; 22 jan 01

	February Totals									
Cases	Supply Fan (kWh)	Evaporator Coil Load Total (kWh)	Sensible (kWh)	Latent (kWh)	Envelope Load Sensible (kWh)	Latent (kWh)	Qfan,heat (kWh)	**Qfan,heat - Qfan (kWh)**	**(del Qfan) /Qfan (frac)**	**(del Qfan) /Qcoil (frac)**
E100	141	3794	3794	0	3655	0	139	**-2**	**-0.017**	**-0.001**
E110	122	3756	3756	0	3637	0	119	**-3**	**-0.025**	**-0.001**
E120	110	3739	3739	0	3632	0	108	**-2**	**-0.023**	**-0.001**
E130	8	215	215	0	208	0	8	**0**	**-0.050**	**-0.002**
E140	6	195	195	0	188	0	6	**0**	**0.025**	**0.001**
E150	136	4528	3786	742	3637	739	149	**13**	**0.090**	**0.003**
E160	121	4508	3769	739	3632	739	137	**16**	**0.120**	**0.004**
E165	145	4549	3809	740	3648	739	161	**16**	**0.097**	**0.003**
E170	63	2237	1498	739	1419	739	79	**16**	**0.204**	**0.007**
E180	112	4535	1607	2928	1419	2958	188	**76**	**0.405**	**0.017**
E185	137	4583	1653	2930	1437	2958	215	**78**	**0.364**	**0.017**
E190	14	579	212	366	188	370	24	**10**	**0.410**	**0.017**
E195	18	602	235	367	208	370	28	**10**	**0.347**	**0.016**
E200	151	5522	4303	1219	4122	1221	181	**30**	**0.166**	**0.005**
E100R2EP	141	3794	3794	0	3655	0	139	**-2**	**-0.015**	**-0.001**
E180R2EP	112	4535	1607	2928	1419	2958	188	**76**	**0.406**	**0.017**

Additional sensitivity tests (designated by "E???R2EP" in Table N2) indicate that this problem is likely not the result of improper entry of the fan heat temperature rise (SUPPLY-DELTA-T = 0.798), as use of equivalent inputs SUPPLY-STATIC = 1.09 and SUPPLY-EFF = 0.5 give the same results for both fan energy use and resulting coil and envelope loads. In addition a prior sensitivity test was run for Case E165 giving the same conclusion.

ID Fan does not precisely cycle on/off with compressor (2% issue at mid-PLR)

Cycling the indoor fan on/off with the compressor is activated by inputting INDOOR-FAN-MODE = INTERMITTENT under SYSTEM-FANS. Based on detailed output, although the indoor fan is cycling, the cycling is not exactly on/off with the compressor and outdoor fan. That is, the increase runtime associated with use of the COOL-CLOS-FPLR curve is not applied to the indoor fan power output (SFKW).

The significance of this discrepancy is best quantified for the mid-PLR Case E170 where:

- SFKW output: 63 kWh
- SFKW if CLOSFPLR adjustment could be applied: 73 kWh.

This is then a 14% issue for the Case E170 indoor fan energy. Since for E170 Qcoil = 2237 kWh, then if additional fan heat is also not included, the additional coil load unaccounted for (kWh) is 10/2237 = 0.45%. Since the total compressor+fans energy is 620 kWH, the missing 10/620 + 0.45% amounts to a 2.1% underestimation of total energy use. A conversation with the manufacturer that provided performance data and technical support for this project indicates that most small unitary systems do not have fan delays (Cawley). Therefore, because equipment often operates at part load, the RESYS2 indoor fan model should incorporate the part load COP degradation effect of the CLOSFPLR curve (additional operating time).

Changes from 117 to 133

The preliminary DOE-2 simulation work prior to January 10, 2000 was done with an older version (117, current version = 133). Our simulation work spotted two problems in the older version as described below. Note that although these problems were already fixed for version 133, they were still in existence through many previous versions, so that if there had been a systematic test in place sooner, we may have been able to address these problems sooner.

Minimum EWB was 60°F (3% issue for dry coil E110)

Minimum EWB was "hardwired" at EWB = 60 in Version 117, and modified in version 125 so that now EWB_{min} = (COOL-FT-MIN) - 10°F, that is ODB - 10°F. [Note: Per one of the code authors, DOE-2 default curves are only tested to COOL-FT-MIN=70 so that releasing the clamp is only a good idea for custom curves.] (hirsch email 7 Jan 00). The effect of this change is shown in Table N3 using Version 133 with minimum EWB at 60 and minimum EWB below the dry coil EWB (designated as "no limit"). Sensitivity is significant (3%) for Case E110. For the future, it would be better if the minimum EWB "clamp" could be set independently of the ODB clamp.

Table N3: Effect of Limiting EWB Extrapolation

	Compr+ODfan Elec. (kWh)	ID Fan Elec. (kWh)	Total Cap. (Btu/h)	Sens. Cap. (Btu/h)	w_{zone}
E110 EWB=60	922	137	25792	21708	0.0072
E110 no limit	948	122	24177	24177	0.0064
E150 EWB=60	1057	137	25792	21708	0.0083
E150 no limit	1051	136	25713	21846	0.0083

This issue of extrapolation effects actually arose during the preliminary testing of the E300 series cases, and may be discussed again later for those cases when that portion of the work is completed.

BF-FPLR = 0.99 at Full Load (0.4% issue for E170)

In version 117, even with the BF-FPLR curve fixed as always = 1.0, the BF-FPLR output was indicating a value of 0.99. In version 133, this now behaves appropriately. For Case E170 using version 133, the compressor + outdoor fan energy consumption sensitivity to this problem was about 0.4% based on a sensitivity tests of BF-FPLR fixed at 1.00 (566 kWh) versus 0.99 (564 kWh).

Documentation Problems

To obtain proper inputs for the items below, the modeler felt it was necessary to clarify the following definitions either with a code author or using sensitivity tests before running the model.

Capacity Input Ambiguity

The documentation (2.1A User's Manual, p. IV.241) is ambiguous about whether to input capacities as gross or net. The following wording is included with regard to total (sensible + latent) capacity:

> " ... Note: When specifying COOLING-CAPACITY for packaged DX cooling units with drawthrough fans, SUPPLY-EFF and SUPPLY-STATIC ... should be omitted and SUPPLY-DELTA-T should be set equal to zero if the COOLING-CAPACITY being defined includes cooling of the fan motor. Otherwise, double accounting for supply fan motor heat will be experienced. For better latent simulation the SUPPLY-DELTA-T should be specified and the COOLING-CAPACITY adjusted to describe the unit without the fan."

This was initially interpreted to mean if a gross capacity is used the SUPPLY-DELTA-T should be 0 (i.e. DOE2 would expand the capacity to include fan heat). So that net capacities were input with SUPPLY-DELTA-T also input and assumed to be automatically added to the fan heat. This resulted in inability to make setpoint in E200. In discussion with one of the code authors (Hirsch, 12/29/99), he thought the documentation wording could be improved, and he clarified that gross capacities should be input for both the total (COOLING-CAPACITY) and sensible (COOL-SH-CAP) capacities.

Sensitivity test results are shown in Table N4. They indicate that the main problem with a user misinterpreting the type of capacity to input, is that resulting E200 zone conditions will be 82.9°F with 0.0121 humidity ratio instead of 80.2°F with 0.0111 humidity ratio. Energy consumption effects are < 1%. For Case E170 (mid-PLR) misinterpretation of capacity inputs causes negligible (< 1%) effect on energy use and coil loads and 3 % effect on resulting humidity ratio.

Table N4: Cooling Capacity Input Sensitivity

Capacity Inputs	Temp (°F)	Humrat	Clg Elec (kWh)	Qcoil (MBtu)	Qlat (MBtu)
E200 Gross	80.2	0.0111	1290	18841	4160
E200 Net	82.9	0.0121	1293	18850	4153
E170 Gross	72.0	0.0105	566	7631	2521
E170 Net	72.0	0.0108	572	7649	2520

Details Regarding Disaggregation of Outdoor Fan Electricity

The SYSTEMS cooling electricity output (SKWQC) automatically includes the disaggregated outdoor fan energy, even when the outdoor fan energy is not included in determining COOLING-EIR. It would be useful to document this in the hourly output variable listing documentation.

Also, that COOL-CLOS-FPLR directly affects disaggregated outdoor fan electricity was not indicated in the documentation supplement and had to be established from sensitivity tests.

COOL-CLOSS-MIN clarification

CFMCYC (hourly fan cycling fraction) is not directly defined in the documentation supplement on RESYS2 (21EDOC.DOC, 10/13/96). Also, COOL-CLOSS-MIN default = 0.8 seemed high, such that our original interpretation of this input was as a limit on the output for COOL-CLOSS-FPLR (rather than as a limit to the CLPLR input to COOL-CLOSS-FPLR, which is apparent after closer scrutiny of the equations).

Therefore, it would be useful to briefly explain (if possible) why the COOL-CLOSS-MIN default is 0.8, and briefly state with text that it is a limit on "input" to (rather than output of) COOL-CLOS-FPLR.

Other Recommended Improvements

More Significant Digits for Some Variables in the .BDL Output

Some variables should have more significant digits in the BDL output. For this model it was not possible to fully verify, with BDL output, inputs of:

- SUPPLY-KW
- OUTSIDE-FAN-ELEC

It would also be helpful if the BDL default listing did not list variables or curve fits not applicable or called by a particular simulation, and indicated if certain curves are set to constants (e.g., x = 1) for a particular combination of inputs; such is the case for COOL-EIR-FPLR in our RESYS2 model.

Return Air Wet-bulb Output does not work for RESYS2.

It would be convenient for checking results if this output were operational for RESYS2.

6. Results

Comparison with Analytical Solution Results

The general level of agreement with the analytical solution results for COP and energy consumption are within 1-3% at high PLR, and within 2-5% at low PLR. This is not as good of an agreement as was achievable with the TRNSYS and CA-SIS simulation total energy consumption and COP results, which are generally within 0-2% of the analytical solution results.

The greatest differences occurred in the low PLR cases, and are likely caused primarily by the fan energy not being adjusted for cycling inefficiency (COP degradation factor) when the fans and compressor should be cycling on/off together. The errors in high-latent-load Case E180 caused by disappearing latent coil load, and sensible coil load minus sensible envelope load being greater than fan energy (heat), only result in < 1% difference in energy consumption versus the analytical solution results; since the difference is greater for more typical cases with lower latent loads (E150-E165), compensating errors are suspected in E180.

Comparison with CIEMAT's DOE-2 Results

The CIEMAT participants used a PTAC with blow-through fan for modeling the cases. The PTAC is not listed as using the new part load cycling curve used in RESYS2, so that it likely uses the original EIR-FPLR curve and related algorithm. Since CIEMAT has done a comparison of the input decks, remaining results differences may be attributable to differences between DOE-2's RESYS2 system model with draw-through fan (used by NREL), versus CIEMAT's PTAC system model, and because NREL and CIEMAT used different versions (from different suppliers) of the software.

For the results of Part IV, CIEMAT's DOE-2 results are generally about the same as NREL's: sometimes a bit better, sometimes a bit worse. CIEMAT's energy consumption inaccuracies at lower PLR are about the same because COP degradation at low PLR is not accounted for in fan energy use for either NREL's RESYS2 model or CIEMAT's PTAC model. For cases with mild/moderate latent load (E150, E160, E165, E170), CIEMAT's results have 1-3% better accuracy versus the analytical solutions than NREL's. Because of this difference, it is difficult to tell in Case E170 if possible differences in compressor COP=f(PLR) from different COP=f(PLR) models actually occurred. CIEMAT's COP results tend to be 1%–2% less accurate than NREL's because of differences in the net refrigeration effect calculation discussed below. The greatest results differences between NREL and CIEMAT were for:

- "fan heat" (the difference between sensible coil load and sensible zone load), see especially cases E100, E110, E180 and E185; software errors are suspected to be the cause

- fan electricity consumption in E110; the CIEMAT result seems inconsistent with other CIEMAT results
- latent coil load for cases E180 and E185 where the PTAC looks ok, but the RESYS2 (NREL) result is different from the corresponding latent zone load; software error suspected
- COP sensitivities for cases E180-E150 and E180-E170; see below.

One reason for the difference in COP sensitivities for E180-E150 and E180-E170 are differences in COP calculation technique, which magnify COP difference for Case E180. Three things here: 1) NREL calculates net refrigeration effect by summing sensible zone load and latent *coil* load, while CIEMAT calculates net refrigeration effect by subtracting fan energy use (heat) from total coil load. In theory these methods should give identical results; however, 2) for DOE-2/NREL the latent coil load is 1% less than the latent zone load (a probable software error), and 3) for CIEMAT the difference between sensible coil load and sensible zone load is 22% greater than the fan energy consumption (another probable software error). This results in a 24% difference in COP sensitivity between the NREL and CIEMAT results for E180-E150.

7. Other

<u>Odd Results</u>

Latent Coil Load Not Precisely Equal to Latent Envelope Load (1% issue at low SHR)

There is a very small disagreement between the latent coil load and the latent internal gains, as shown in Table N5. Since the ambient humidity ratio is 0.01 then it seems this difference could be caused by a small amount of moisture leaking in or out of the zone according to the humidity ratio gradient between the zone air and ambient air. In the worst case (E180, high latent load) this is about a 1% difference in latent coil load versus latent internal gains. Note, this is likely not an issue related to zone initial conditions because these outputs are for February - the simulation had all of January to get to steady state in the zone.

Table N5: Latent Coil Load Discrepancies

	$Q_{lat,coil}$ (MBtu)	$Q_{lat,env}$ (MBtu)	w_{zone}	$w_{coil,sat}$
E150	3764	3754	0.0083	0.00725
E160	3755	3754	0.0099	0.00875
E180	14866	15017	0.0164	0.01130
E200	6190	6201	0.0111	0.00955

Coil Energy Exceeds Total Capacity (1.4% issue at ARI rating conditions)

In the full load ARI conditions test (E200), the results indicate that DOE-2's output for gross energy removed by the cooling coil (QC) exceeds the total capacity (QCT = COOLING-CAP f(EWB,ODB)) by 400 Btu/h (1.4%). DOE-2 still makes setpoint for this, so that the total capacity provides no limit to operation. (The documentation clearly states that capacity limits are set by the sensible capacity unless there is a dry coil, so this is not a problem.) However, the documentation states that:

QCKW = (Qcap(EWB,ODB)) * EIR(EWB,ODB) * PCTON

where PCTON is a part loading multiplier. Since PCTON $\leq$ 1, then QCKW would not be precisely corresponding to the full Qcoil when Qcoil > Qcap in the DOE-2 outputs.

In general the DOE-2 outputs for total and sensible capacity are agreeing with linear interpolations/ extrapolations of the performance data within 1% for dry coils and within 2%-3% for wet coils. (This was checked for Cases E100, E110, E170, E180, E185, and E200.) In E200 the total capacity appears to be about 1% lower than the performance data, and since there also seems to be some additional heat coming to the coil from an unknown source (see Table N2), the combination of those issues may explain this discrepancy.

COIL-BF-FT Multiplier Curve Insensitivity (1% issue for E185)

As shown in Table N6, the COIL-BF-FT curve appears to have been disabled for the purpose of calculating energy consumption, although it appears active for the purpose of calculating coil surface temperature.

Table N6: COIL-BF-FT Observations

E185 Parametrics	Avg Total Electric (W)	Bypass Factor from output	COIL-BF-FT from output	Coil Surface Temp (°F)
COIL-BF= 0.049 COIL-BF-FT = enabled	2099	0.065	1.331.	62.3
COIL-BF = 0.049 COIL-BF-FT = 1.000	2099	0.049	1.000	62.4
BF = 0.065 COIL-BF-FT = 1.000	2079	0.065	1.000	62.1

If the software were working as expected, average total electric values in the second row should be those that are shown in the third row, and values of the third row should be those currently shown in the first row.

Other Sensitivity Test Results

Effect of Ambient Humidity Ratio on Case E100 (Dry Coil) Results (4% issue)

Outdoor humidity ratio (w_{amb}) was varied in weather data to check the effect of initial zone relative humidity on dry coil results. (Recall DOE-2 initializes zone conditions with ambient conditions.) The following variations were observed in the DOE-2 outputs for Case E100 as described in the Table N7.

Table N7: Effect of Ambient Humidity Ratio on Case E100 (Dry Coil) Results

w_{amb} (kg/kg)	w_{zone} (kg/kg)	Compr+ odf elec (kWh)	EIR-FT (mult.)	TotCap= SensCap (Btu/h)	CAP-FT (mult.)	Qcoil (Btu/h)
0.0102	0.0074	1378	0.3392	21119	0.757	12945
0.0053	0.0053	1431	0.3562	19806	0.710	12984

These variations are caused by the EWB being lower when the zone humidity ratio is lower throughout the simulations. However, according to one reference, once a dry coil condition is reached, the capacity should not change with decreasing wet bulb (Brandemuehl 1993, p. 4-82.). So, this result may be indicative of a bug in the DOE-2 algorithm. [Intuitively, when EWB is decreasing below the wet-bulb temperature corresponding to the initial onset of dry coil conditions, there should not be a change in the compressor power either, since there is no change to latent load, and no change to coil surface conditions either (there is no moisture on the coil). However, the Brandemuehl reference only explicitly mentions the capacity.]

The effect on ambient humidity ratio for the wet coil case E185 is negligible (<0.1%), as expected.

It may be useful to add a low ambient humidity ratio test to the current test suite to check this issue in the other programs - as dry coils are more likely to occur in actual operation with low ambient humidity.

Sensitivity to Constant Atmospheric Pressure

The weather data allows atmospheric pressure to vary in accordance with Miami, FL weather. Use of constant pressure weather data had an insignificant effect on results as shown in Table N8. The results are from an earlier sensitivity test during November 1999; so the "Varying" results do not match the final results.

Table N8: Sensitivity to Constant Atmospheric Pressure

Patm	Clg. Elec. (kWh)	Sens. Cap. (Btu/h)	Qcoil (Btu/h)
Varying	1285.707	20104	19338
Constant = 1 atm.	1285.729	20097	19338

Use of Default Curves

Although the directions to the participants are explicit about using the most detailed level of modeling possible, DOE-2 includes default curves for various types of equipment. It was interesting to compare the results from curves based on the given performance data to the RESYS2 defaults. A summary of the comparison for selected cases is included in Table N9.

Table N9: Comparison of Results for RESYS2 Default versus HVAC BESTEST Performance Curves

Descrip.	Comp+odfan elec (kWh)	del %	EIR-FT	CAP-FT	SH-FT	BF-FT	CLOS-FPLR
E100	1378		1.494	0.757	1.311	1.041	0.98
E100dflt	1310	-4.9%	1.412	0.745	1.404	0.159	0.98
E110	943		0.987	0.867	1.462	1.312	0.95
E110dflt	935	-0.8%	0.954	0.907	1.577	0.393	0.93
E170	557		0.909	0.984	1.196	0.929	0.86
E170dflt	592	+6.3%	0.925	0.988	1.205	0.899	0.82
E185	1410		1.263	0.944	0.750	1.331	0.97
E185dflt	1340	-5.0%	1.222	0.947	0.768	0.928	0.95
E195	228		1.280	0.928	0.800	1.233	0.80
E195dflt	231	+1.3%	1.241	0.929	0.817	0.863	0.73

In general, differences in cooling energy can be up to 5-6% as a result of specific curve effects, and probably up to 10% for part load energy use if E150 results are consistent with E185 results. It is interesting to note the compensating disagreement in Case E195 that combines the effect of low SHR and low PLR that is disaggregated in Cases E170 and E185. Also interesting to observe is how disagreements can be more significant in comparing differences between results. For example using E185-E170, the different performance curve choice results in a 12.3% disagreement in the difference comparison.

For the curves the differences between custom values versus default values are generally consistent, indicating custom curve fits were developed properly. However, for bypass factor f(T) curves, the default DOE-2 curve is quite a bit more sensitive than the custom curve. This is most apparent for the dry coil cases, and the DOE-2 authors indicate that the default curve fits have not been tested below EWB = 60°F. (Also, DOE-2 default curve limit settings do not allow the level of extrapolation applied here; see Section 5 for discussion of extrapolation effects.) Performance data for the HVAC BESTEST equipment does not indicate as much variation in bypass factor as is present in the DOE-2 default curve, so it would be interesting to learn more about how the default BF-FT curve was developed

Other Recommended Improvements to DOE-2

Should Bypass Factor User Input Be Necessary?

It seems that since performance curves are supplied for sensible and total capacity as a function of EWB and ODB, that a detailed simulation tool using a bypass factor model should be able to automatically calculate hourly full-load bypass factor based on the capacities. For example the ASHRAE HVAC 2 Toolkit (Brandemuehl 1993) has an iterative routine for determining apparatus dew point and therefore bypass factor when given entering and leaving air conditions. Use of such a routine avoids potential for the user to be providing potentially conflicting input information, and would save the user some time in generating custom input data. The DOE-2 code authors should consider adding such a feature as a convenience to users. Such a feature has been implemented in a custom version of DOE-2 for Florida Solar Energy Center (Henderson).

8. Conclusions and Recommendations

Regarding the DOE-2.1E Results

Working with DOE-2.1E during the development of HVAC BESTEST cases E100-E200 allowed a thorough examination of the DOE-2 outputs and identified the following issues relating to accuracy of the software. The list includes significance of the problem, and related actions.

- Minimum supply temperature calculation bug fixed in RESYS2 (36% issue)
- Minimum EWB was 60°F (3% issue for dry coil E110), already fixed in version 125
- Fan heat discrepancy (2% issue at low SHR), authors notified
- Indoor fan does not precisely cycle on/off with compressor (2% issue at mid PLR), authors notified
- BF-FPLR = 0.99 at full load (0.4% issue for E170), already fixed by version 133
- Documentation ambiguities:
    - Capacity definitions (affects ability to make set point at ARI conditions), authors notified
    - Disaggregated outdoor fan power is a function of COOL-CLOSS-FPLR and is included in cooling electric output (SKWQC), (would save the user some time), authors notified
    - COOL-CLOSS-MIN clarification, (would save the user some time), authors notified
- Odd Results (notify authors, but some reasonable explanation is possible)
    - Latent coil load not precisely equal to latent envelope load (1% issue at low SHR)
    - Coil energy exceeds total capacity (1.4% issue at ARI conditions)
- Other sensitivity test results and recommended improvements
    - Ambient humidity ratio effect on dry coil results (4% issue for E100), authors notified
    - Is bypass factor input by user really necessary?, (automating would save the user time and possibly improve accuracy), authors notified

The general level of agreement with the analytical solution results for COP and energy consumption are within 1-3% at high PLR, and within 2-5% at low PLR. This is about the same level of agreement as was achieved with CIEMAT's DOE-2 results, but is not as good of an agreement as was achievable with the

TRNSYS and CASIS simulation results. For TRNSYS and CASIS, total energy consumption and COP results are generally within 0-2% of the analytical solution results. For DOE-2, the greatest differences versus analytical solutions occurred in the low PLR cases, and are likely caused primarily by the fan energy not being adjusted for cycling inefficiency (COP degradation factor) when the fans and compressor should be cycling on/off together. The errors in high-latent-load Case E180 caused by disappearing latent coil load, and sensible coil load minus sensible envelope load being greater than fan energy (heat), only result in < 1% difference in energy consumption versus the analytical solution results; since the difference is greater for more typical cases with lower latent loads (E150-E165), compensating errors are suspected in E180.

Although the biggest error in DOE-2 was found and fixed in the early going (August 1994) when DOE-2.1E was relatively new, it looks like there may still be some remaining discrepancies that can affect energy consumption results for specific cases by up to 4%. These issues have been reported to the code authors, and to the developers of the U.S. Department of Energy's next generation simulation program, EnergyPlus that incorporates elements of DOE-2 and BLAST.

Regarding HVAC BESTEST

HVAC BESTEST has helped isolate a number of errors in DOE-2. These errors were isolated using both the diagnostic logic applied to comparative test results, as well as by checks for internal consistency among the outputs required by BESTEST. Some problems in DOE-2, such as with the hardwired minimum EWB, may have been addressed sooner if this BESTEST comparative test method had been in place.

Other software (see Section 3.4) also had some significant problems that were identified from comparison with other programs. Therefore, HVAC BESTEST has proven its usefulness by providing useful analytical verification tests that identify problems in a variety of software, and by providing an environment congenial to thorough checking of detailed output with the specific intention of looking for discrepancies.

HVAC BESTEST is just in its developmental stages and needs additional cases. From one of the additional sensitivity tests with DOE-2, it is apparent that the following case could be added:

- Low ambient humidity ratio for dry coil

Some additional mechanical equipment cases to consider are:

- Realistic controls (using E140, (low-PLR), E170 (mid-PLR), E200? (full load))
- f(PLR) based on more detailed data
- VAV with real fan curves
- Fan heat test with fan always on and low load
- E300 series (Neymark and Judkoff 2001): complete the existing cases and perhaps add more cases (highest priority)
- E100-E200 type cases over an expanded range of ODB, EWB, and EDB
- ARI conditions at 1/4 to 1/2 PLR, i.e. no f(EWB, ODB, EDB) interactions at low PLR.

Additionally, the robustness of the cases could be improved by making available expanded performance data or requiring greater extrapolation of existing performance data. The feasibility of increasing robustness in this manner should be studied.

A fundamental issue regarding the cases would be to include sufficient detail in the test specification to compare results with more detailed models that predict the behavior of individual components within the unitary system. Gathering the necessary data required for this is expected to require considerable effort. However, such an effort would allow comparison of performance map models with "first principles" models that get into the equipment physical behavior at a more detailed level.

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Program name (please include version number)

DOE-2.1E version 133

Your name, organisation, and country

Joel Neymark, National Renewable Energy Laboratory/J. Neymark & Associates, United States

Program status

	Public domain
x	Commercial: *the PC version used is commercial, and was purchased from J.J. Hirsch & Associates, Camarillo, CA, USA.*
	Research
	Other (please specify)

Solution method for unitary space cooling equipment

x	Overall Performance Maps
	Individual Component Models
	Constant Performance (no possible variation with entering or ambient conditions)
	Other (please specify)

Interaction between loads and systems calculations

	Both are calculated during the same timestep
x	First, loads are calculated for the entire simulation period, then equipment performance is calculated separately
	Other (please specify)

Time step

x	Fixed within code (please specify time step): *one hour*
	User-specified (please specify time step)
	Other (please specify)

Timing convention for meteorological data : sampling interval

x	Fixed within code (please specify interval): *one hour*
	User-specified

Timing convention for meteorological data: period covered by first record

x	Fixed within code (please specify period or time which meteorological record covers): *0:00 - 1:00*
	User-specified

Meteorological data reconstitution scheme

x	Climate assumed stepwise constant over sampling interval
	Linear interpolation used over climate sampling interval
	Other (please specify)

Output timing conventions

	Produces spot predictions at the end of each time step
	Produces spot output at end of each hour
x	Produces average outputs for each hour (please specify period to which value relates): *same as time step*

Treatment of zone air

x	Single temperature (i.e. good mixing assumed)
	Stratified model
	Simplified distribution model
	Full CFD model
	Other (please specify)

Zone air initial conditions

x	Same as outside air
	Other (please specify)

Internal gains output characteristics

	Purely convective
	Radiative/Convective split fixed within code
x	Radiative/Convective split specified by user
	Detailed modeling of source output

Mechanical systems output characteristics

x	Purely convective
	Radiative/Convective split fixed within code
	Radiative/Convective split specified by user
	Detailed modeling of source output

Control temperature

x	Air temperature
	Combination of air and radiant temperatures fixed within the code
	User-specified combination of air and radiant temperatures
	User-specified construction surface temperatures
	User-specified temperatures within construction
	Other (please specify)

Control properties

	Ideal control as specified in the user's manual
	On/Off thermostat control
	On/Off thermostat control with hysteresis
	On/Off thermostat control with minimum equipment on and/or off durations
x	Proportional control: *a throttling range setting of 0.1°F was input along with a "TWO-POSITION" thermostat type.*
	More comprehensive controls (please specify)

Performance Map: characteristics

a	Default curves
x	Custom curve fitting
	Detailed mapping not available
	Other (please specify)

Performance Map: independent variables

x	Entering Dry-bulb Temperature: *The effect of EDB is "hardwired" in DOE-2, and only affects sensible capacity.*
x	Entering Wet-bulb Temperature
x	Outdoor Dry-bulb Temperature
x	Part Load Ratio
a	Indoor Fan Air Flow Rate: *did not use; fan air flow was always at rated conditions when the fan was operating.*
	Other (please specify)

Performance Map: dependent variables

x	Coefficient of Performance (or other ratio of load to electricity consumption)
x	Total Capacity
x	Sensible Capacity
x	Bypass Factor
	Other (please specify)

Performance Map: available curve fit techniques

x	Linear, f(one independent variable): *BF-FPLR (always = 1)*
a	Quadratic, f(one independent variable)
x	Cubic, f(one independent variable): *CLOSS-FPLR*
a	Bi-Linear, f(two independent variables)
x	Bi-Quadratic, f(two independent variables): *SCAP-FT, CAP-FT, BF-FT, EIR-FT*
	Other (please specify)

Performance Map: extrapolation limits

a	Limits independent variables: *ODB; EWBmin = ODBmin - 10*
a	Limits dependent variables: *all curves*
x	No extrapolation limits
	Extrapolation not allowed
	Other (please specify)

Cooling coil and supply air conditions model

	Supply air temperature = apparatus dew point (ADP); supply air humidity ratio = humidity ratio of saturated air at ADP
a	Bypass factor model using listed ADP data
x	Bypass factor model with ADP calculated from extending condition line
x	Fan heat included
	More comprehensive model (please specify)

Disaggregation of fans' electricity use directly in the simulation and output

a	Indoor fan only
a	Outdoor fan only
x	Both indoor and outdoor fans disaggregated in the output
a	None - disaggregation of fan outputs with separate calculations by the user

Economizer settings available (for E400 series)

x	Temperature
x	Enthalpy
x	Compressor Lockout
	Other (please specify)

Appendix III-B

Dresden University of Technology

Faculty for Mechanical Engineering

Department of Technical Building

HVAC BESTEST Modeler Report

TRNSYS 14.2 (A Transient System Simulation Program)

By

H.-T. Le

G. Knabe

September 2000

Commentary on the using terms:

$P_{envelope,lat}$	latent heat flow through building envelope
$P_{envelope,sen}$	sensible heat flow through building envelope
$P_{gain,lat}$	internal latent gain
$P_{gain,sen}$	internal sensible gain
$P_{zone,lat}$	total latent load of the room
$P_{zone,sen}$	total sensible load of the room
$P_{zone,tot}$	total load of the room
$P_{Adj.\_Net\_Lat}$	Adj. Net Latent Capacity
$P_{Adj.\_Net\_Sen}$	Adj. Net Sensible Capacity
$P_{Adj.\_Net\_Tot}$	Adj. Net Total Capacity
$P_{ECL\_Capacity}$	Capacity of Evaporator Coil
P_{ODfan}	heat of condenser fan
P_{IDfan}	heat of indoor fan (supply fan)
$P_{Compressor}$	compressor power
$Q_{zone,lat}$	latent roomload (zone load) for a time period
$Q_{zone,sen}$	sensible roomload (zone load) for a time period
$Q_{zone,tot}$	total roomload (zone load) for a time period
$Q_{Adj.\_Net\_Lat}$	Adj. latent Coil Load for a time period
$Q_{Adj.\_Net\_Sen}$	Adj. sensible Coil Load for a time period
$Q_{Adj.\_Net\_Tot}$	Adj. total Coil Load for a time period
Q_{ECL_Lat}	latent Evaporator Coil Load for a time period
Q_{ECL_Sen}	sensible Evaporator Coil Load for a time period
Q_{ECL_Tot}	total Evaporator Coil Load for a time period
Q_{ODfan}	power of condenser fan for a time period
Q_{IDfan}	power of indoor fan for a time period
$Q_{Compressor}$	power of compressor for a time period
COP_{full}	Coefficient of Performance at full load operation
COP_{part}	Coefficient of Performance at part load operation
EWB	Entering WetBulb temperature
EDB	Entering DryBulb temperature
EHR	Entering Humidity Ratio
ZHR	Zone Humidity Ratio
DPT	Dew Point Temperature
PLR	Part Load Ratio
SHR	Sensible Heat Ratio
$x_{sat.}$	humidity ratio of saturated air at zone EWB
$h_{sat.}$	enthalpy of saturated air at zone EWB
x_{zone}	Zone Humidity Ratio (ZHR)
$p_{sat.}$	pressure of saturated air at zone EWB
p_{tot}	pressure of the zone air
h_{zone}	enthalpy of zone air
t_B	operating time for a time period

1. Introduction

TRNSYS is a Program for solar simulation written by University of Wisconsin, USA. Since applying a license of this program the Dresden University of Technology has changed the program codes and has written additional modules, so the TUD has a new program for the simulation of heating system and air conditioning. It is designated TRNSYS TUD. Physical and empirical models for each component of a system are at TU Dresden available. The loads and the system can be calculated in the same time step. The time step used for the simulation is a hundredth of an hour.

In order to generally run a simulation with TRNSYS, at first, the building and the HVAC system have to be modeled as precisely as possible. For that, one needs the building construction and the building location as well as the physical sizes e.g. surface coefficients, thermal and moisture capacitance of the walls, considering the storage of the zone air and of objects inside of this building etc. The control strategy is defined either in the *.dek-file or in a user specified type. The *.dek-file contains all required data for the simulation e.g. time-step, simulation period, tolerances, user specified equations, applied types, defined outputs etc. Thus, it is named as input-file or management-file for the TRNSYS simulation.

2. Modeling Assumptions

TRNSYS doesn't allow directly any input as latent gains or latent capacity. Instead of this an input of mass flow of water vapor is available, so a conversion from latent capacity into mass flow of water vapor needs to be done. The formula for this conversion is as follows:

$$m_{\mathrm{w\_flow}} = \frac{\mathrm{P}_{\mathrm{latent}}}{\Delta h_v} \quad ,$$

where:

$\mathrm{m}_{\mathrm{w\_flow}}$ = mass flow of water vapor $\left(\frac{\mathrm{kg}}{\mathrm{s}}\right)$

$\mathrm{P}_{\mathrm{latent}}$ = latent capacity (gains) (kW)

$\Delta \mathrm{h}_{\mathrm{v}}$ = water heat of vaporization $\left(\frac{\mathrm{kJ}}{\mathrm{kg}}\right)$

In the test description data points of performance map at full load operation are given. In this map, where the total capacities are greater than the sensible capacities it indicates the wet coil conditions, otherwise dry coil conditions occur. These data points are only valid for wet coil conditions, so the data points of performance map for dry coil conditions cannot be used. Therefore, an analysis of system behavior for dry coil conditions is necessary. To analyze the system behavior it is utilized the adjusted net capacity given in Table 1-6e of HVAC BESTEST [22] because the supply fan is a part of the mechanical system. Following figures show the behavior of this split system for the wet coil conditions.

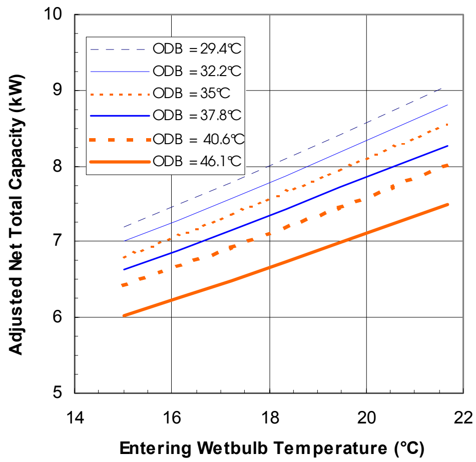


Figure 1: Behavior of Adjusted Net Total Capacity Depending on EWB and ODB

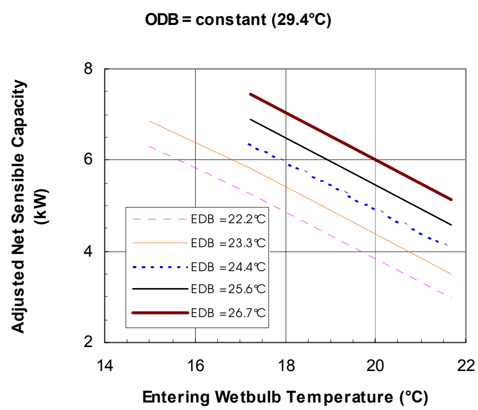


Figure 2: Behavior of Adjusted Net Sensible Capacity Depending on EWB and EDB

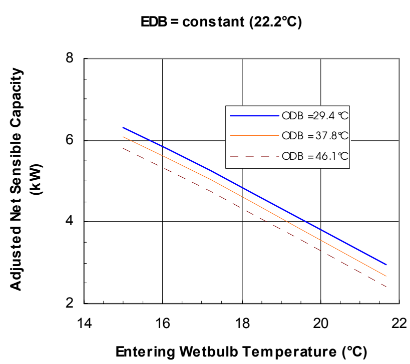


Figure 3: Behavior of Adjusted Net Sensible Capacity depending on EWB and ODB

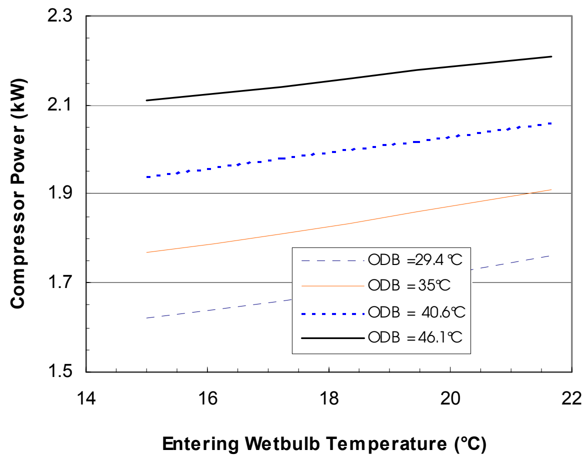


Figure 4: Behavior of Compressor Power depending on EWB and ODB

From Figures 1 to 4 it results following: In the field of wet coil conditions where the EWB greater than the intersection point (EWB_1; see figure 5) the adjusted net total capacity is proportional to the entering wet bulb temperature, whereas the adjusted net sensible capacity is inversely proportional. The coil capacities (sensible and total) do not change with varied EWB ($EWB < EWB_1$) by dry coil conditions. Figure 5 illustrates this behavior where "intersection point" indicates the initial dry coil condition (boundary of the wet coil condition).

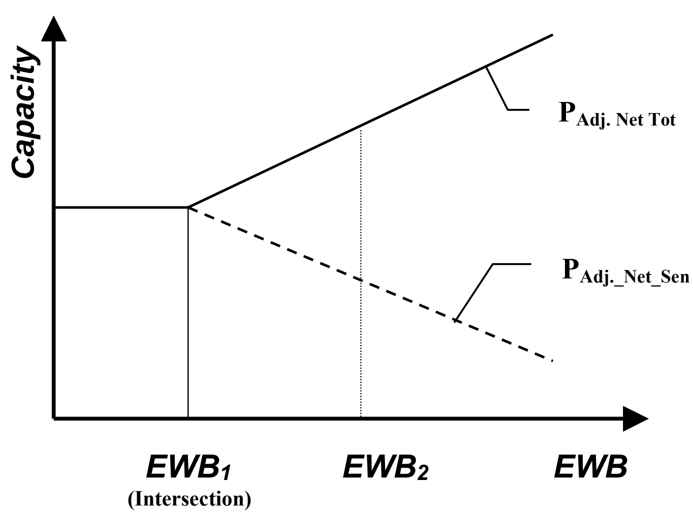


Figure 5: System behavior

These above figures 1 to 4 show that the adjusted net total capacity and the compressor power for wet coil conditions behave linear to the EWB and the ODB, whereas the adjusted net sensible capacity is a linear function of EWB, EDB and ODB. According to manufacturer, the data points of the performance map contain some uncertainties in the experimental measurements [9], therefore it is recommended to apply over full extremes valid data points for the approximation/extrapolation. That could eliminate this uncertainty. So, a fitting of custom curve of the performance map using multi-linear approximations for the wet coil performance can be done. The equations of approximation have the following formulas:

$$P_{Adj.\_Net\_Tot} = (\vartheta_{ODB} * A_1 + A_2) * \vartheta_{EWB} + (\vartheta_{ODB} * A_3 + A_4) \qquad (1)$$

$$P_{Adj.\_Net\_Sen} = (\vartheta_{ODB} * B_1 + \vartheta_{EDB} * B_2 + B_3) * \vartheta_{EWB} + (\vartheta_{ODB} * B_4 + \vartheta_{EDB} * B_5 + B_6) \qquad (2)$$

$$P_{Compressor} = (\vartheta_{ODB} * C_1 + C_2) * \vartheta_{EWB} + (\vartheta_{ODB} * C_3 + C_4) \qquad (3)$$

These equations (1), (2) and (3) are the characteristic curves of the evaporator coil identifiable from the performance map at full load operation.

The point between the dry and wet coil conditions is defined as the intersection point (EWB_1) that can be solved from equations (1) and (2):

$$\vartheta_{EWB,intersection} = \frac{(\vartheta_{ODB} \cdot B_4 + \vartheta_{EDB} \cdot B_5 + B_6) - (\vartheta_{ODB} \cdot A_3 + A_4)}{(\vartheta_{ODB} \cdot A_1 + A_2) - (\vartheta_{ODB} \cdot B_1 + \vartheta_{EDB} \cdot B_2 + B_3)} \qquad (4)$$

where:

A_1 = -0.00374128077

A_2 = 0.390148024

A_3 = -0.0135886344

A_4 = 3.3894767

B_1 = -0.000629934065

B_2 = -0.00267022306

B_3 = -0.424403961

B_4 = -0.0199848128

B_5 = 0.535732506

B_6 = 2.57416166

C_1 = -0.00040500166

C_2 = 0.0345047542

C_3 = 0.0357013738

C_4 = 0.219759426

For determination of the coil capacities and the compressor power by the points where EWB is less than EWB_1, the EWB is replaced by EWB_1. That means if EWB < EWB_1 then the coil capacities and the compressor power are a function of EWB_1 by given ODB and EDB.

Note: The replacement is only for calculation of the coil capacities and the compressor power, because they are constant in the field of dry coil conditions. But for computation of zone humidity ratio from EWB and Set point EWB and EWB_1 must not replaced.

The set point of indoor dry bulb temperature is given. Hysteresis at the set point should be set to zero. There is only the control of zone temperature. There is no zone humidity control. The zone humidity ratio will float in accordance with zone latent loads and moisture removal by the split system. The type of temperature control was initially not given. So it is suggested to use the ideal as well as the real control. (NREL later revised the test specification (May 99) to indicate ideal control.)

Using ideal control (see subsection 3.2) will give reliable results for comparison with the realistic control and checks the energy balance over zone boundary. With this controller the time step has no consequences at all.

The cases were also run using a more realistic control method (see subection 3.3). The indoor dry bulb temperature is changed continuously for using realistic control. The magnitude of such change depends on part load ratio. In order to keep this change as small as possible the time step should be short. In that case the value of time step is 36 s.

3. Modeling Options

A review of the capabilities of program TRNSYS TUD is given in the pro-forma. There are many areas where this program allows choosing the modeling techniques. Following is a short description of some cases why various techniques were chosen:

Zone initial conditions

Regarding the initial zone conditions equals to the given outdoor conditions, the analysis shows that the initial value of the entering wet bulb temperature is always greater than the intersection point. So the evaporator always begins operation under wet coil conditions. For the cases where the latent zone loads do not occur the system operates at the crossing point (see figure 5 above and subsection 3.1 below). This is an operating point where the coil just becomes dry. On the other hand, the analysis also shows a slight change of bypass factor (Table 1) as calculated from performance map data using the listed ADP values, because the bypass factors at the outdoor dry bulb temperature of 29.4 °C at the height are nearly the bypass factors at ODB of 35 °C. In [9] it shows that the bypass factor for wet coil operation is only a function of airflow rate and not dependent on either indoor or outdoor air conditions. So it is assumed to set an arithmetical mean value of bypass factor for all test cases.

Table 1: Cooling Grade (Bypass Factor) is calculated from performance map

ODB (°C)	EWB (°C)	Cooling Grade	Bypass Factor	ODB (°C)	EWB (°C)	Cooling Grade	Bypass Factor
29.4	15.0	0.9579	0.0421	37.8	15.0	0.9575	0.0425
	17.2	0.9456	0.0544		17.2	0.9423	0.0577
	19.4	0.9471	0.0529		19.4	0.9474	0.0526
	21.6	0.9544	0.0456		21.6	0.9531	0.0469
32.2	15.0	0.9594	0.0406	40.6	15.0	0.9553	0.0447
	17.2	0.9449	0.0551		17.2	0.9380	0.0620
	19.4	0.9523	0.0477		19.4	0.9519	0.0481
	21.6	0.9503	0.0497		21.6	0.9608	0.0392
35.0	15.0	0.9571	0.0429	46.1	15.0	0.9599	0.0401
	17.2	0.9428	0.0572		17.2	0.9418	0.0582
	19.4	0.9506	0.0494		19.4	0.9583	0.0417
	21.6	0.9602	0.0398		21.6	0.9645	0.0355

$0.9380.1 \leq \text{Cooling Grade} \leq 0.9645$; $\quad \text{mean}_{\text{Cooling Grade}} = 0.9522$

$0.0355 \leq \text{Bypass Factor} \leq 0.0620$; $\quad \text{mean}_{\text{Bypass Factor}} = 0.0478$

Evaluation of zone air conditions

If the bypass factor is applied, the supply air conditions can be determined. It follows a calculation of the zone air. There is an idealization that the zone air is evenly distributed in the zone, because the TRNSYS TUD full CFD model is not applied. The reasons for not using the CFD model are the unknown locations of supply air as well as of the thermostat, and a long time for computing the zone conditions.

Another method (not using the bypass factor) is with assumption that the room temperature, the zone loads and the cooling coil capacities distribute evenly in the zone. The above-mentioned approximation equations are used. This is so-called technique energy balance over zone boundary.

Both methods have the same effect. However the second one is simpler, so it is applied.

For every time step the zone loads and the cooling coil capacities are updated. It follows the calculation of zone temperature. For using the realistic control the cooling coil capacities depend on the actual zone temperature. Therefore, many iterations need to be done. Thus, it lasts several hours for computation. Because the time step is very small and the change of zone temperature is not very high, so it utilizes the zone temperature of last output for the calculation of cooling coil capacities for the actual time step.

Application of three solution techniques

To solve this problem the following methods are used:

- Analytical solution
- Simulation with an ideal control
- Simulation with a realistic control.

3.1 Analytical solution

See report of analytical solution done by TUD, located in Part II.

3.2 Simulation with an ideal control

An ideal control with a computer simulation is used to see how much its results deviate from the results of the analytical solution. There is no deviation between the set point and the actual values of zone air temperature. The program automatically predicts the zone air humidity ratio. The equipment is always in operation. The coil capacities are adjusted to the zone loads.

When comparing simulated ideal control to analytical solution results, there is in some cases very small differences caused by iterative closure tolerance differences. So the both methods (analytical solution and simulation with an ideal control) represent a suitable basis for the comparison with results of simulation with a realistic control.

3.3 Simulation with a realistic control

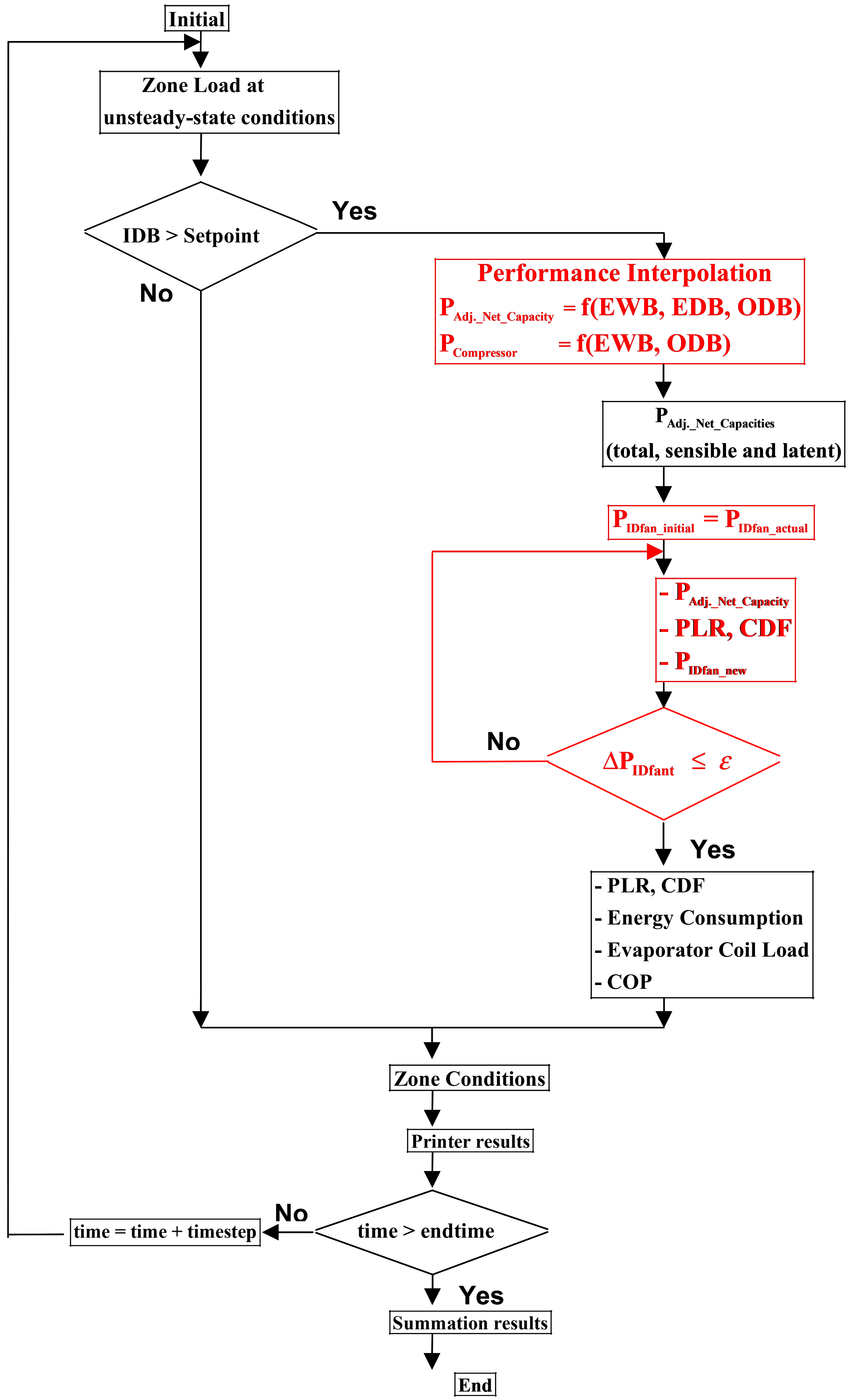
As saying above, by a using the realistic control the first step to do is comparing the actual zone temperature with the set point. If the zone temperature is less than the set point, then the compressor is shut off. Otherwise, the equipment is put into operation. That means the system must either be always on or always off for one full time step. The equipment capacities and the zone loads will be updated at beginning of the time step. The equipment capacities will be computed with the above-saying multi-linear approximation equations. Afterwards, it follows the calculation of sensible as well as latent zone balances. The zone humidity ratio will float in depending on the zone moisture balance. There is no limitation for ON/OFF operating time. That means the equipment can often be switched on/off. The hysteresis round the set point is put to 0 K.

The fan heat of supply fan is dependent on CDF factor. This is because at part loads the system run time is extended. So, there is some additional fan heat that should be accounted for that is not included in User‘s Manual Table 1-6e (which gives adjusted net capacities for full load operation). Therefore, in order to determine the indoor fan heat a few iterations are required.

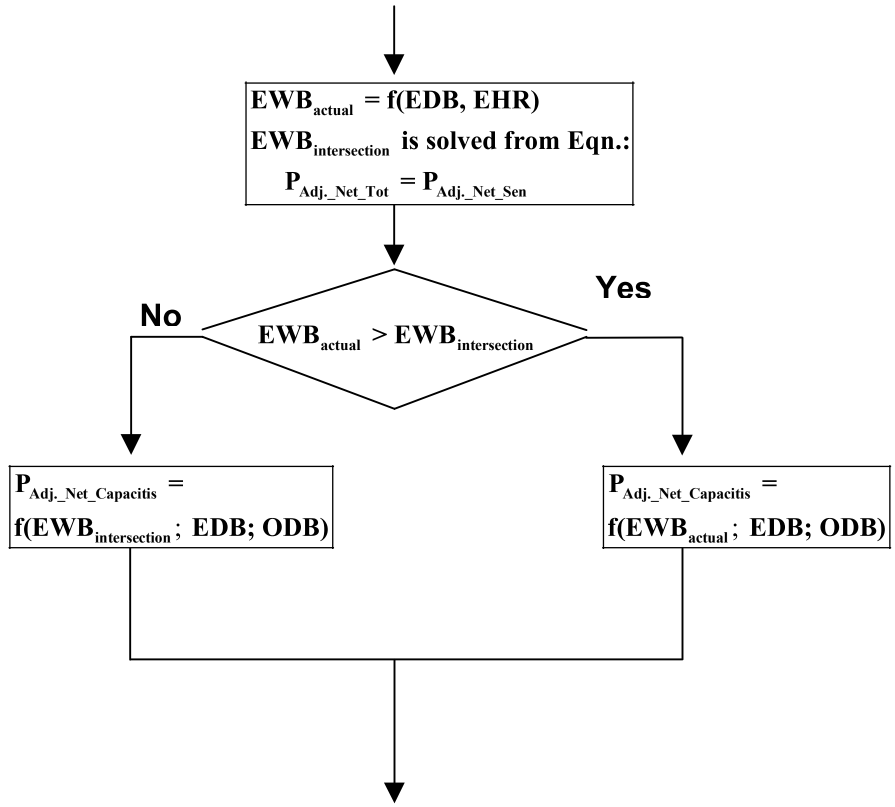
There is a marked deviation between set point and actual zone air temperature. This deviation depends on the part load ratio, so it changes from case to case. The time step for all cases amounts to 36 seconds. For cases E140 and E165 an additional calculation with a time step of 10.8 seconds is carried out to show, how much the mean COP and mean IDB are dependent on time step.

The algorithm for simulation of HVAC BESTEST with a realistic controller can be expressed in a flowchart below.

Flowchart for simulation with a realistic controller:



Calculation $P_{Adj.\_Net\_Capacity}$ by simulation model:



3.4 Methodical comparison between the analytical solution (subsection 3.1) and simulation with a realistic controller (subsection 3.3)

This subsection gives a summary of similarities and differences between analytical solution and simulation with a realistic controller.

3.4.1 Similarities

#1. Interpolation/extrapolation of performance map with multi-linear equation:

$$P_{Adj.\_Net\_Capacities} = f(EWB, EDB, ODB)$$

$$P_{Compressor} = f(EWB, ODB)$$

#2. Iteration loop for supply fan heat

#3. Consideration CDF = f(PLR) for calculation of energy consumption

3.4.2 Differences

	Analytical Solution	**Simulation**
#**1**.	Ideal controller	Real controller
#**2**.	IDB = Setpoint	IDB $\neq$ Setpoint
#**3**.	No time-step is required	Time-step is required
#**4**.	Zone Load at steady-state conditions	Zone Load at unsteady-state conditions
#5.	Steady-state operating point for Dry Coil Wet Coil Conditions is separately **Dry Coil:** (SHR =1) **IF** ($EWB_{initial} < EWB_{intersection}$) $EWB_{steady\text{-}state} = EWB_{intersection}$ **ELSE** $EWB_{steady\text{-}state} = EWB_{initial}$ **Wet Coil:** (0 < SHR < 1) $EWB_{steady\text{-}state}$ is solved from equation.: $SHR_{zone} = SHR_{coil}$	Steady-state operating point is reached and after a few hours of simulation. determined. $EWB_{steady\text{-}state} = f(EDB, EHR)$ EHR is calculated from heat- and moisture balances.
#6.	Determination $EWB_{steady\text{-}state}$ is inside of iteration loop for supply fan heat	Determination $EWB_{steady\text{-}state}$ is situated outside of iteration loop for supply fan heat.
#7.	Calculation operating time for time period results	Summation results of every time step

4. Modeling Difficulties

Nothing.

5. Software Errors Discovered and/or Comparison Between Different Versions of the Same Software (errors discussed below is only occurred by simulation model with a realistic control)

Two errors were identified in TRNSYS-TUD using HVAC BESTEST Cases E100-E200. The first error involved insufficient precision in some variables, and the second error involved inconsistent data transfer. How the test procedure helped to isolate those errors is discussed below.

At the first run, big errors for cases with small part load operation (e.g. E130, E140, E170, E190, E195) occurred . The total energy removal by the equipment is much less than the total zone internal gains loads. The higher part load ratio the better the agreement of the energy balance over the zone boundary is. That means there are either the consideration errors of modeling with the part load ratio or the errors of the program codes. These errors could not be occurred in the modeling of the building, because the case for full load test shows a closed agreement.

The checking began with case where the big errors are occurred e.g. case E130. For this case, the sum for a whole month February of sensible zone load amounted to 209 kWh and of electrical consumption of the supply fan 5 kWh, while the sum of removal energy by equipment was 116 kWh. The total capacity of the equipment at the steady state operating point is about 6100 W. The sensible zone load is 270 W. The ratio of the equipment capacity to the zone load is about 22.6. That means after about 23 time steps the equipment has to run for a one time-step. In the fact, the evaporator started again after circa 36 time steps. So this is an explanation why the sum of removal energy was much less than the sum of the zone loads. The zone dry bulb temperature is output by the building module TYPE56. This temperature essentially increased slower as expected. This fact does not happen to the case with full load operation. This lead to discovery of a problem with use of single precision variables in the subroutine that has made a big deviation caused by round off for cases with small part load ratio. After changing to double precision variables in that subroutine, the new results show more consistent.

After several checking of the modeling of the equipment, for safety there is also checked the building modeling, a small error was found in the computing technique. A type (module) is used to save the entering dry bulb temperature (EDB) and the entering humidity ratio (EHR) for calculation the equipment performance. This type was called at the end of every time step but before the printer. By calling of this type all the defined equations in DECK are updated. After that the printer prints the just updated values. So the results were inconsistent. This type has been set now after printer and it works well.

Tables 2 through 5 give a summary of results before and after both improvements were made.

Table 2: Energy Consumption of Equipment Before and After Improvement (realistic control)

Cases	February Cooling Energy Consumption								
	Before Improvement				After Improvement				(Total_before-Total_after)/ Total_after*100 [%]
	Total (kWh)	Com-pressor (kWh)	Supply Fan (kWh)	Conden-ser Fan (kWh)	Total (kWh)	Com-pressor (kWh)	Supply Fan (kWh)	Conden-ser Fan (kWh)	
E100	1519	1308	143	67	1512	1303	142	67	**0.5**
E110	1096	903	131	62	1062	876	127	59	**3.2**
E120	1007	836	117	55	1002	832	115	54	**0.6**
E130	58	50	5	3	110	95	10	5	**-47.1**
E140	36	29	4	2	69	57	8	4	**-47.9**
E150	1200	993	141	66	1192	987	139	65	**0.7**
E160	1151	959	131	61	1133	944	128	60	**1.6**
E165	1505	1285	150	70	1490	1272	148	69	**1.0**
E170	544	452	63	29	636	529	73	34	**-14.5**
E180	1136	953	124	58	1080	906	118	55	**5.2**
E185	1543	1338	139	65	1538	1334	139	65	**0.3**
E190	91	76	10	5	165	138	18	9	**-44.8**
E195	143	124	13	6	252	218	23	11	**-43.3**
E200	1479	1252	154	73	1480	1253	155	73	**-0.1**

Table 3: Evaporator Capacities Before and After Program Improvement (realistic control)

Cases	February Evaporator Coil Load								
	Before Improvement			After Improvement			(Tot_before-Tot_after)/ Tot_after*100 [%]	(Sen_be-Sen_after)/Sen_ after*100	(Lat_be-Lat_after)/Lat_ after*100
	Total (kWh)	Sensible (kWh)	Latent (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)			
E100	3799	3799	0	3798	3798	0	**0.0**	**0.0**	**---**
E110	3866	3866	0	3763	3763	0	**2.7**	**2.7**	**---**
E120	3748	3748	0	3747	3747	0	**0.0**	**0.0**	**---**
E130	116	116	0	220	220	0	**-47.3**	**-47.3**	**---**
E140	103	103	0	199	199	0	**-48.0**	**-48.0**	**---**
E150	4517	3778	739	4515	3776	739	**0.0**	**0.0**	**0.0**
E160	4552	3805	747	4499	3760	739	**1.2**	**1.2**	**1.0**
E165	4565	3818	747	4535	3796	739	**0.7**	**0.6**	**1.0**
E170	1897	1269	628	2232	1492	739	**-15.0**	**-15.0**	**-15.0**
E180	4713	1612	3101	4494	1537	2957	**4.9**	**4.8**	**4.9**
E185	4535	1577	2958	4534	1577	2957	**0.0**	**0.0**	**0.1**
E190	318	115	204	578	208	370	**-44.9**	**-44.9**	**-44.9**
E195	341	131	209	601	232	370	**-43.3**	**-43.3**	**-43.3**
E200	5504	4281	1223	5498	4277	1221	**0.1**	**0.1**	**0.1**

Table 4: Zone Load before and after Program Improvement (realistic control)

Cases	February Zone Load								
	Before Improvement			After Improvement			(Tot_be-Tot_after)/ Tot_after*100 [%]	(Sen_be-Sen_after) /Sen_ after*100	(Lat_be-Lat_after)/ Lat_ after*100
	Total (kWh)	Sensible (kWh)	Latent (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)			
E100	3656	3656	0	3656	3656	0	**0.0**	**0.0**	**---**
E110	3637	3637	0	3637	3637	0	**0.0**	**0.0**	**---**
E120	3632	3632	0	3632	3632	0	**0.0**	**0.0**	**---**
E130	209	209	0	209	209	0	**0.0**	**0.0**	**---**
E140	190	190	0	190	190	0	**0.0**	**0.0**	**---**
E150	4376	3637	739	4376	3637	739	**0.0**	**0.0**	**0.0**
E160	4371	3632	739	4371	3632	739	**0.0**	**0.0**	**0.0**
E165	4388	3648	739	4388	3648	739	**0.0**	**0.0**	**0.0**
E170	2159	1419	739	2159	1419	739	**0.0**	**0.0**	**0.0**
E180	4376	1419	2957	4376	1419	2957	**0.0**	**0.0**	**0.0**
E185	4395	1438	2957	4395	1438	2957	**0.0**	**0.0**	**0.0**
E190	559	190	370	559	190	370	**0.0**	**0.0**	**0.0**
E195	578	209	370	578	209	370	**0.0**	**0.0**	**0.0**
E200	5343	4122	1221	5343	4122	1221	**0.0**	**0.0**	**0.0**

Table 5: Mean Zone Conditions before and after Program Improvement (realistic control)

Cases	February Mean Zone Conditions								
	Before Improvement			After Improvement			(COP_old-COP_new)/ COP_new* 100	(IDB_old-IDB_new)/ IDB_new* 100	(IHR_old-IHR_new)/ IHR_new* 100
	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)			
E100	2.41	22.3	0.0076	2.42	22.6	0.0075	**-0.5**	**-1.4**	**0.8**
E110	3.41	22.3	0.0067	3.43	22.5	0.0066	**-0.5**	**-1.0**	**0.3**
E120	3.60	26.7	0.0081	3.63	27.1	0.0080	**-0.7**	**-1.3**	**1.7**
E130	1.89	22.1	0.0076	1.92	21.6	0.0075	**-1.1**	**2.1**	**0.6**
E140	2.76	22.1	0.0067	2.80	21.5	0.0066	**-1.3**	**2.6**	**0.9**
E150	3.65	22.3	0.0084	3.67	22.7	0.0085	**-0.8**	**-1.6**	**-1.2**
E160	3.84	26.7	0.0102	3.86	27.0	0.0102	**-0.5**	**-0.9**	**-0.1**
E165	2.93	23.4	0.0095	2.94	23.8	0.0095	**-0.4**	**-1.5**	**-0.6**
E170	3.38	22.2	0.0105	3.40	22.1	0.0105	**-0.8**	**0.4**	**-0.1**
E180	4.04	22.2	0.0163	4.06	22.3	0.0164	**-0.4**	**-0.4**	**-0.4**
E185	2.85	22.3	0.0162	2.86	22.4	0.0163	**-0.3**	**-0.5**	**-0.3**
E190	3.38	22.2	0.0159	3.40	21.9	0.0157	**-0.7**	**1.0**	**1.3**
E195	2.30	22.2	0.0155	2.31	22.0	0.0153	**-0.4**	**0.7**	**1.1**
E200	3.62	26.7	0.0114	3.61	26.7	0.0113	**0.2**	**0.0**	**0.9**

6. Results

Finally, three sets of HVAC BESTEST results have been submitted. The results of analytical solution and of the simulation with an ideal control show slight differences. This is possibly caused by iterative closure tolerance differences. The simulation with an ideal control has considered the dynamic behavior of the building, while the analytical solution is based on steady state building. However, for the BESTEST E100 series the dynamic behavior of the building does not make a big difference. A comparison with other participants' final results shows a close agreement. So the methods of analytical solution and the ideal control give a suitable basis for testing validity of realistic control simulations. The simulations with a realistic control and no limitation for ON/OFF run-time have a very close agreement with both above-mentioned methods.

The following tables show a summary of results of the three solution ways.

Table 6: Evaporator Coil Loads

Cases	February Totals of Evaporator Coil Load											
	Total [kWh]			(Real-Ideal)/ Ideal [%]	Sensible [kWh]			(Real-Ideal)/ Ideal [%]	Latent [kWh]			(Real-Ideal)/ Ideal [%]
	Anal. Sol.	Ideal Control	Real Control		Anal. Sol.	Ideal Control	Real Control		Anal. Sol.	Ideal Control	Real Control	
E100	3800	3800	3798	**0.0**	3800	3800	3798	**0.0**	0	0	0	**---**
E110	3765	3765	3763	**0.0**	3765	3765	3763	**0.0**	0	0	0	**---**
E120	3749	3748	3747	**0.0**	3749	3748	3747	**0.0**	0	0	0	**---**
E130	219	219	220	**0.4**	219	219	220	**0.4**	0	0	0	**---**
E140	197	198	199	**0.4**	197	198	199	**0.4**	0	0	0	**---**
E150	4518	4517	4515	**0.0**	3778	3777	3776	**-0.1**	739	739	739	**0.0**
E160	4501	4500	4499	**0.0**	3761	3761	3760	**0.0**	739	739	739	**0.0**
E165	4537	4537	4535	**0.0**	3798	3798	3796	**-0.1**	739	739	739	**0.0**
E170	2232	2231	2232	**0.0**	1493	1492	1492	**0.0**	739	739	739	**0.0**
E180	4495	4494	4494	**0.0**	1538	1538	1537	**0.0**	2957	2957	2957	**0.0**
E185	4535	4535	4534	**0.0**	1578	1578	1577	**0.0**	2957	2957	2957	**0.0**
E190	578	577	578	**0.1**	208	208	208	**0.2**	370	370	370	**0.0**
E195	601	601	601	**0.1**	232	231	232	**0.2**	370	370	370	**0.0**
E200	5498	5498	5498	**0.0**	4277	4277	4277	**0.0**	1221	1221	1221	**0.0**

Table 7: Zone Loads

Cases	February Totals of Zone Load											
	Total [kWh]			(Real-Ideal)/ Ideal [%]	Sensible [kWh]			(Real-Ideal)/ Ideal [%]	Latent [kWh]			(Real-Ideal)/ Ideal [%]
	Anal. Sol.	Ideal Control	Real Control		Anal. Sol.	Ideal Control	Real Control		Anal. Sol.	Ideal Control	Real Control	
E100	3656	3656	3656	**0.0**	3656	3656	3656	**0.0**	0	0	0	**---**
E110	3637	3637	3637	**0.0**	3637	3637	3637	**0.0**	0	0	0	**---**
E120	3632	3632	3631	**0.0**	3632	3632	3631	**0.0**	0	0	0	**---**
E130	209	209	209	**0.0**	209	209	209	**0.0**	0	0	0	**---**
E140	190	190	190	**0.0**	190	190	190	**0.0**	0	0	0	**---**
E150	4376	4376	4376	**0.0**	3637	3637	3636	**0.0**	739	739	739	**0. 0**
E160	4371	4371	4370	**0.0**	3632	3632	3631	**0.0**	739	739	739	**0.0**
E165	4388	4388	4387	**0.0**	3649	3649	3648	**0.0**	739	739	739	**0.0**
E170	2159	2159	2159	**0.0**	1420	1419	1419	**0.0**	739	739	739	**0.0**
E180	4376	4376	4376	**0.0**	1420	1419	1419	**0.0**	2957	2957	2957	**0.0**
E185	4396	4395	4395	**0.0**	1439	1438	1438	**0.0**	2957	2957	2957	**0.0**
E190	559	559	559	**0.0**	190	190	190	**0.0**	370	370	370	**0.0**
E195	579	578	579	**0.0**	209	209	209	**0.0**	370	370	370	**0.0**
E200	5343	5343	5343	**0.0**	4122	4122	4122	**0.0**	1221	1221	1221	**0.0**

Table 8: Mean Zone Conditions of the simulation period

Cases	February Mean											
	COP			(Real-Ideal)/ Ideal [%]	Air Temperature [°C]			(Real-Ideal)/ Ideal [%]	Humidity Ratio [kg/kg]			(Real-Ideal)/ Ideal [%]
	Anal. Sol.	Ideal Control	Real Control		Anal. Sol.	Ideal Control	Real Control		Anal. Sol.	Ideal Control	Real Control	
E100	2.39	2.40	2.42	**0.7**	22.2	22.2	22.6	**2.0**	0.0075	0.0075	0.0075	**0.1**
E110	3.38	3.41	3.43	**0.5**	22.2	22.2	22.5	**1.5**	0.0066	0.0066	0.0066	**0.6**
E120	3.59	3.61	3.63	**0.7**	26.7	26.7	27.1	**1.4**	0.0080	0.0080	0.0080	**0.0**
E130	1.89	1.92	1.92	**-0.2**	22.2	22.2	21.6	**-2.5**	0.0075	0.0075	0.0075	**0.2**
E140	2.75	2.80	2.80	**0.1**	22.2	22.2	21.5	**-3.1**	0.0066	0.0066	0.0066	**0.0**
E150	3.63	3.65	3.67	**0.7**	22.2	22.2	22.7	**2.1**	0.0083	0.0083	0.0085	**2.2**
E160	3.83	3.85	3.86	**0.4**	26.7	26.7	27.0	**1.1**	0.0101	0.0101	0.0102	**1.7**
E165	2.93	2.93	2.94	**0.7**	23.3	23.3	23.8	**2.1**	0.0094	0.0093	0.0095	**2.2**
E170	3.37	3.39	3.40	**0.3**	22.2	22.2	22.1	**-0.4**	0.0105	0.0105	0.0105	**0.5**
E180	4.04	4.05	4.06	**0.2**	22.2	22.2	22.3	**0.6**	0.0162	0.0163	0.0164	**0.6**
E185	2.85	2.85	2.86	**0.2**	22.2	22.2	22.4	**0.8**	0.0161	0.0162	0.0163	**0.8**
E190	3.39	3.41	3.40	**-0.1**	22.2	22.2	21.9	**-1.1**	0.0159	0.0159	0.0157	**-1.3**
E195	2.29	2.32	2.31	**-0.5**	22.2	22.2	22.0	**-0.9**	0.0154	0.0155	0.0153	**-0.9**
E200	3.62	3.61	3.61	**0.0**	26.7	26.7	26.7	**0.0**	0.0113	0.0113	0.0113	**0.0**

Table 9: Energy Consumption of Equipment and of Compressor

Cases	February Totals of Cooling Energy Consumption							
	Total [kWh]			(Real-Ideal)/ Ideal [%]	Compressor [kWh]			(Real-Ideal)/ Ideal [%]
	Anal. Solution	Ideal Control	Real Control		Anal. Sol.	Ideal Control	Real Control	
E100	1531	1522	1512	**-0.7**	1319	1311	1303	**-0.6**
E110	1076	1067	1062	**-0.5**	888	879	876	**-0.4**
E120	1013	1007	1002	**-0.6**	841	836	832	**-0.5**
E130	111	109	110	**1.3**	95	94	95	**1.3**
E140	69	68	69	**1.3**	57	56	57	**1.3**
E150	1207	1199	1192	**-0.6**	999	992	987	**-0.5**
E160	1140	1137	1133	**-0.3**	950	947	944	**-0.3**
E165	1498	1499	1490	**-0.7**	1279	1280	1272	**-0.6**
E170	641	636	636	**0.0**	533	528	529	**0.0**
E180	1083	1081	1080	**-0.1**	908	907	906	**-0.1**
E185	1545	1542	1538	**-0.2**	1340	1337	1334	**-0.2**
E190	165	164	165	**0.7**	138	138	138	**0.7**
E195	252	250	252	**0.7**	219	216	218	**0.7**
E200	1476	1480	1480	**0.0**	1249	1253	1253	**0.0**

Table 10: Energy Consumption of Fans

Cases	February Totals of Cooling Energy Consumption							
	Supply Fan [kWh]			(Real-Ideal)/ Ideal [%]	Condenser Fan [kWh]			(Real-Ideal)/ Ideal [%]
	Anal. Solution	Ideal Control	Real Control		Anal. Sol.	Ideal Control	Real Control	
E100	144.1	143.6	142.4	**-0.9**	67.6	67.5	66.8	**-0.9**
E110	128.1	127.7	126.7	**-0.8**	60.2	60.0	59.5	**-0.8**
E120	116.9	116.7	115.5	**-1.0**	54.9	54.8	54.2	**-1.0**
E130	10.4	10.3	10.4	**1.3**	4.9	4.8	4.9	**1.3**
E140	8.2	8.1	8.2	**1.3**	3.9	3.8	3.9	**1.3**
E150	141.2	140.9	139.4	**-1.0**	66.3	66.1	65.5	**-1.0**
E160	129.3	129.1	128.3	**-0.6**	60.7	60.6	60.2	**-0.6**
E165	149.1	149.4	148.0	**-0.9**	70.0	70.1	69.5	**-0.9**
E170	73.6	73.2	73.0	**-0.2**	34.6	34.4	34.3	**-0.2**
E180	118.8	118.5	118.1	**-0.3**	55.8	55.6	55.5	**-0.3**
E185	139.4	139.2	138.8	**-0.3**	65.5	65.4	65.2	**-0.3**
E190	18.2	18.0	18.1	**0.7**	8.5	8.5	8.5	**0.7**
E195	22.8	22.6	22.8	**0.7**	10.7	10.6	10.7	**0.7**
E200	154.5	154.6	154.6	**0.0**	72.5	72.6	72.6	**0.0**

Table 11: February Energy Balance of Test Zone

Cases	Total Evaporator Coil Load – (Total Zone Load + Supply Fan) [kWh]		
	Anal. Solution	Ideal Control	Realistic Control
E100	**0.0**	**0.0**	**0.0**
E110	**0.0**	**0.0**	**0.0**
E120	**0.0**	**0.0**	**0.0**
E130	**0.0**	**0.0**	**0.0**
E140	**0.0**	**0.0**	**0.0**
E150	**0.1**	**0.0**	**0.0**
E160	**0.0**	**0.0**	**0.0**
E165	**0.1**	**0.0**	**0.0**
E170	**0.0**	**0.0**	**0.0**
E180	**0.2**	**0.0**	**0.0**
E185	**0.2**	**0.0**	**0.0**
E190	**0.0**	**-0.1**	**0.1**
E195	**0.0**	**0.0**	**0.0**
E200	**0.1**	**0.0**	**0.0**

Assessment

A review of tables 6 through 11 results following:

The zone loads (table 7) have a closest agreement between ideal and realistic control simulation.

Table 6 shows that the biggest difference of evaporator coil loads between two control methods amounts 0.4% for case E130 and case E140.

The difference for zone air temperature with realistic control versus ideal control is largest for Case E140: 3.1% (see Table 8).

The zone air humidity ratio of cases E150 and E165 is deviated 2.2% from the humidity ratio of simulation with an ideal control.

Further, the COP difference for case E100, E120, E150 and E165 at table 8 is 0.7%.

The biggest difference of energy consumption (tables 9 and 10) is 1.3% for cases E130 and E140.

Table 11 shows the energy balance of the zone boundary. There is a very good agreement.

For cases E140 and E165 there are occurring big differences for zone air temperature, indoor humidity ratio and coefficient of performance. For all above-given results in tables 6 - 11 the time step is 0.01 h (=36seconds).This is maybe big and it causes the deviation. It is assumed that there is a function of time step. Therefore, the simulation for cases E140 and E165 were also run with a time step of 0.003h (=10.8 seconds) to see whether the time step really causes these differences.

Tables 12 and 13 contain the comparison for the results of simulation with a time step of 0.01h and 0.003h. These tables show that the shorter the time step the better agreements with the results of simulation with an ideal control will be reached.

Table 12: Comparison for total energy consumption and COP

Cases	**Total of Cooling Energy Consumption** [KWh]			**COP**		
	Ideal Control	Real Control	(Real- Ideal)/ Ideal [%]	Ideal Control	Real Control	(Real-Ideal)/ Ideal [%]
E140	67.8	68.6	**1.3**	2.80	2.80	**0.1**
E140_0.003 h	67.8	68.5	**1.0**	2.80	2.78	**-0.7**
E165	1499.7	1489.9	**-0.7**	2.93	2.94	**0.7**
E165_0.003 h	1499.7	1496.6	**-0.2**	2.93	2.93	**0.2**

Table 13: Comparison for IDB and IHR

Cases	**Indoor Dry-bulb Temperature** **[°C]**			**Indoor Humidity Ratio** **[kg/kg]**		
	Ideal Control	Real Control	(Real- Ideal)/ Ideal [%]	Ideal Control	Real Control	(Real-Ideal)/ Ideal [%]
E140	22.2	21.51	**-3.1**	0.0066	0.0066	**0.0**
E140_0.003 h	22.2	21.99	**-0.9**	0.0066	0.0066	**0.0**
E165	23.3	23.78	**2.1**	0.0093	0.0095	**2.2**
E165_0.003 h	23.3	23.46	**0.7**	0.0093	0.0094	**0.8**

Figure 6 shows an example for the dependence of zone air temperature on the part load ratio. In the dry coil case, the zone air temperature has been changed continuously, while the zone air humidity ratio remains constant (Figure 7).

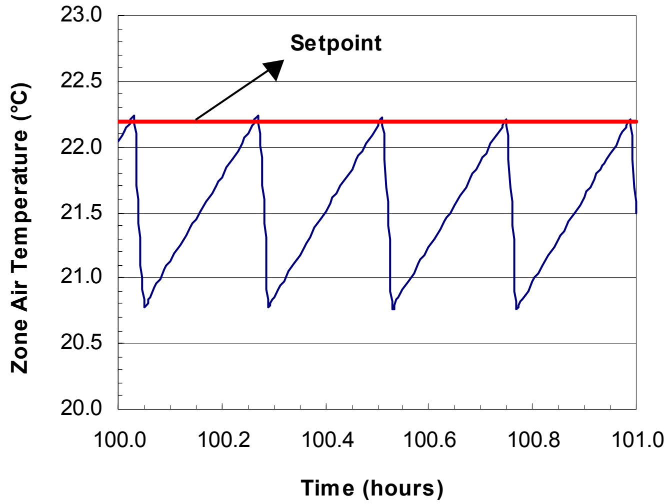


Figure 6: Curve of Zone Air Temperature for Case E140 with realistic control

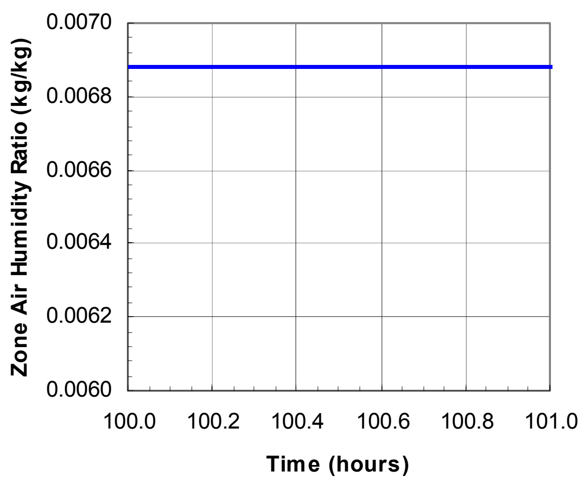


Figure 7: Curve of Zone Air Humidity Ratio for Case E140 with realistic control

Figures 8 and 9 give the curves of zone temperature and humidity ratio for case E170. Comparison with dry coil conditions, the zone temperature as well as the humidity ratio for wet coil are changed continuously.

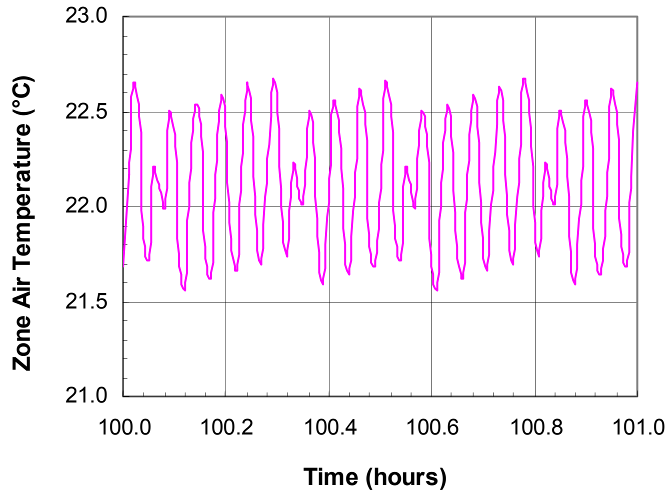


Figure 8: Curve of Zone Air Temperature for Case E170 with realistic control

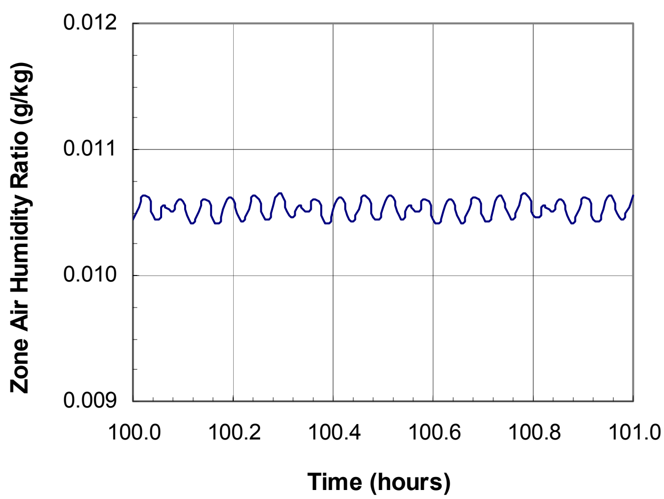


Figure 9: Curve of Zone Air Humidity Ratio for Case E170 with realistic control

7. Other (optional)

The analysis shows that the indoor air humidity ratio (IHR) for an idealized condition depends on the sensible heat ratio of zone loads, indoor dry bulb temperature and the outdoor dry bulb temperature (table 14). In reality, this is very important for planning to know how to predict the adjusted zone humidity ratio. If the IHR is above the 11.4 g/kg there is outside of thermal comfort field for human feeling (see bold values in Table 14). Table 14 shows that the less sensible heat ratio (SHR) of zone loads the high the floating IHR, although the set point temperature is set low enough. The IHR is independent on SHR of loads. This is an important behavior of split equipment.

Table 14: Indoor Humidity Ratio

SHR of Zone Loads	**Indoor Humidity Ratio** (g/kg) **At constant ODB = 32 °C and at IDB of**				
	20°C	**22°C**	**24°C**	**26°C**	**28°C**
0.20	**17.5**	**19.2**	**21.1**	**23.2**	**25.4**
0.25	**16.4**	**18.0**	**19.8**	**21.6**	**23.7**
0.30	**15.3**	**16.8**	**18.5**	**20.2**	**22.1**
0.35	**14.4**	**15.8**	**17.3**	**18.9**	**20.6**
0.40	**13.5**	**14.8**	**16.2**	**17.7**	**19.3**
0.45	**12.6**	**13.8**	**15.2**	**16.5**	**18.0**
0.50	**11.9**	**13.0**	**14.2**	**15.5**	**16.9**
0.55	11.1	**12.2**	**13.3**	**14.5**	**15.8**
0.60	10.5	11.4	**12.5**	**13.6**	**14.8**
0.65	9.8	10.7	**11.7**	**12.8**	**13.9**
0.70	9.2	10.1	11.0	**12.0**	**13.0**
0.75	8.7	9.4	10.3	11.2	**12.2**
0.80	8.1	8.9	9.6	10.5	11.4
0.85	7.6	8.3	9.0	9.8	10.7
0.90	7.1	7.8	8.5	9.2	10.0
0.95	6.7	7.3	7.9	8.6	9.3
1.00	6.3	6.8	7.4	8.0	8.7

In reality, the split equipment will be operated with limited time of ON/OFF conditions and/or a hysteresis round the set point to prevent the often switching of the equipment. So it has been suggested to simulate the BESTEST Series E100 – E200 with the following conditions:

1. Simulation with realistic control, 5 min OFF operating time and no hysteresis,
2. Simulation with realistic control, 5 min ON/OFF operating time and no hysteresis,
3. Simulation with realistic control, 5 min OFF operating time and 2K hysteresis,
4. Simulation with realistic control, 5 min ON/OFF operating time and 2K hysteresis,
5. Simulation with realistic control, no limitation of operating time and 2K hysteresis.

Following schemes make clear the above-mentioned control strategies

#1:

Compressor=ON IF IDB => Tset AND t,off => 5 min; otherwise OFF

Where: Tset = thermostat setpoint;
t,off = time the compressor has been OFF.

#2:

Compressor=ON IF IDB => Tset AND t,off => 5 min;
Compressor=OFF IF IDB <= Tset AND t,on => 5 min

Where: Tset = thermostat setpoint;
t,off = time the compressor has been OFF;
t,on = time the compressor has been ON.

#3:

Compressor=ON IF IDB => (Tset + 1 K) AND t,off => 5 min;
Compressor=OFF IF IDB <= (Tset – 1K)

Where: Tset = thermostat setpoint;
t,off = time the compressor has been OFF.

#4

Compressor=ON IF IDB => (Tset + 1 K) AND t,off => 5 min;
Compressor=OFF IF IDB <= (Tset – 1 K) AND t,on => 5 min

Where: Tset = thermostat setpoint;
t,off = time the compressor has been OFF;
t,on = time the compressor has been ON.

#5

Compressor=ON IF IDB => (Tset + 1 K);
Compressor=OFF IF IDB <= (Tset – 1 K)

Where: Tset = thermostat setpoint.

Tables 15 - 19 give the results of these modified control strategies.

Table 15: Simulation with realistic control, 5 min OFF operating time and no hysteresis

Cases	February Totals							February Mean		
	Cooling Energy Consumption				Evaporator Coil Load					
	Total [kWh]	Com-Pressor [kWh]	Supply Fan [kWh]	Con-denser Fan [kWh]	Total [kWh]	Sensible [kWh]	Latent [kWh]	COP	IDB [°C]	Humidity Ratio [kg/kg]
E100	1438	1242	133	63	3785	3785	0	2.54	26.16	0.0075
E110	1008	836	117	55	3748	3748	0	3.61	26.53	0.0067
E120	957	799	107	50	3734	3734	0	3.81	31.06	0.0081
E130	110	95	10	5	220	220	0	1.92	21.64	0.0075
E140	69	57	8	4	199	199	0	2.80	21.51	0.0066
E150	1152	958	132	62	4504	3765	739	3.80	25.65	0.0095
E160	1093	915	121	57	4487	3748	739	4.01	30.56	0.0117
E165	1450	1241	142	67	4527	3788	739	3.03	26.01	0.0104
E170	622	518	70	33	2228	1488	739	3.49	23.48	0.0112
E180	1060	892	114	54	4489	1532	2957	4.13	23.90	0.0177
E185	1505	1307	135	63	4528	1571	2957	2.92	23.94	0.0176
E190	165	138	18	9	578	208	370	3.41	21.95	0.0157
E195	251	218	23	11	601	232	370	2.31	22.06	0.0154
E200	1480	1253	155	73	5498	4277	1221	3.61	26.73	0.0113

Table 16: Simulation with realistic control, 5 min ON/OFF operating time and no hysteresis

Cases	February Totals							February Mean		
	Cooling Energy Consumption				Evaporator Coil Load					
	Total [kWh]	Com-pressor [kWh]	Supply Fan [kWh]	Con-denser Fan [kWh]	Total [kWh]	Sensible [kWh]	Latent [kWh]	COP	IDB [°C]	Humidity Ratio [kg/kg]
E100	1438	1242	133	63	3785	3785	0	2.54	26.16	0.0075
E110	1008	836	117	55	3748	3748	0	3.61	26.53	0.0067
E120	957	799	107	50	3734	3734	0	3.81	31.06	0.0081
E130	122	105	12	6	226	226	0	1.31	17.66	0.0052
E140	77	63	10	4	205	205	0	1.27	17.01	0.0046
E150	1152	958	132	62	4504	3765	739	3.80	25.65	0.0095
E160	1093	915	121	57	4487	3748	739	4.01	30.56	0.0117
E165	1450	1241	142	67	4527	3788	739	3.03	26.01	0.0104
E170	656	543	76	36	2238	1498	739	3.32	20.04	0.0096
E180	1060	892	114	54	4489	1532	2957	4.13	23.90	0.0177
E185	1505	1307	135	63	4528	1571	2957	2.92	23.94	0.0176
E190	171	143	19	9	581	211	370	3.54	19.85	0.0142
E195	260	225	24	11	604	235	369	2.46	20.21	0.0140
E200	1480	1253	155	73	5498	4277	1221	3.61	26.73	0.0113

Table 17: Simulation with realistic control, 5 min OFF operating time and 2K hysteresis

Cases	February Totals							February Mean		
	Cooling Energy Consumption				Evaporator Coil Load					
	Total [kWh]	Com-pressor [kWh]	Supply Fan [kWh]	Con-denser Fan [kWh]	Total [kWh]	Sensible [kWh]	Latent [kWh]	COP	IDB [°C]	Humidity Ratio [kg/kg]
E100	1461	1261	136	64	3789	3789	0	2.50	25.03	0.0072
E110	1023	847	119	56	3753	3753	0	3.56	25.34	0.0064
E120	969	808	109	51	3738	3738	0	3.76	29.90	0.0077
E130	110	95	10	5	220	220	0	2.07	21.33	0.0072
E140	68	56	8	4	198	198	0	2.84	21.73	0.0067
E150	1168	969	135	63	4508	3769	739	3.75	24.50	0.0091
E160	1105	924	123	58	4491	3751	739	3.96	29.50	0.0112
E165	1476	1262	146	69	4532	3793	739	2.97	24.61	0.0098
E170	631	525	72	34	2230	1491	739	3.44	22.44	0.0107
E180	1072	901	117	55	4492	1535	2957	4.09	22.86	0.0168
E185	1526	1324	137	64	4532	1575	2957	2.88	22.95	0.0168
E190	164	138	18	8	577	208	370	3.52	21.99	0.0157
E195	251	218	23	11	601	232	370	2.38	21.93	0.0153
E200	1480	1253	155	73	5498	4277	1221	3.61	26.73	0.0113

Table 18: Simulation with realistic control, 5 min ON/OFF operating time and 2K hysteresis

Cases	February Totals							February Mean		
	Cooling Energy Consumption				Evaporator Coil Load			COP	IDB [°C]	Humidity Ratio [kg/kg]
	Total [kWh]	Com-pressor [kWh]	Supply Fan [kWh]	Con-denser Fan [kWh]	Total [kWh]	Sensible [kWh]	Latent [kWh]			
E100	1461	1261	136	64	3789	3789	0	2.50	25.03	0.0072
E110	1023	847	119	56	3753	3753	0	3.56	25.34	0.0064
E120	969	808	109	51	3738	3738	0	3.76	29.90	0.0077
E130	119	102	11	5	224	224	0	1.23	18.53	0.0054
E140	75	61	9	4	204	204	0	1.20	17.87	0.0047
E150	1168	969	135	63	4508	3769	739	3.75	24.50	0.0091
E160	1105	924	123	58	4491	3751	739	3.96	29.50	0.0112
E165	1476	1262	146	69	4532	3793	739	2.97	24.61	0.0098
E170	647	537	75	35	2235	1496	739	3.35	20.83	0.0099
E180	1072	901	117	55	4492	1535	2957	4.09	22.86	0.0168
E185	1526	1324	137	64	4532	1575	2957	2.88	22.95	0.0168
E190	168	141	19	9	579	210	370	3.57	20.79	0.0149
E195	255	221	23	11	603	233	370	2.50	21.16	0.0147
E200	1480	1253	155	73	5498	4277	1221	3.61	26.73	0.0113

Table 19: Simulation with realistic control, no limitation of operating time and 2K hysteresis

Cases	February Totals							February Mean		
	Cooling Energy Consumption				Evaporator Coil Load			COP	IDB [°C]	Humidity Ratio [kg/kg]
	Total [kWh]	Com-pressor [kWh]	Supply Fan [kWh]	Con-denser Fan [kWh]	Total [kWh]	Sensible [kWh]	Latent [kWh]			
E100	1508	1300	142	67	3797	3797	0	2.42	22.77	0.0072
E110	1056	871	126	59	3762	3762	0	3.45	22.82	0.0064
E120	1001	832	115	54	3747	3747	0	3.64	26.99	0.0076
E130	110	95	10	5	220	220	0	2.07	21.33	0.0072
E140	68	56	8	4	198	198	0	2.84	21.73	0.0067
E150	1192	987	140	66	4515	3776	739	3.67	22.61	0.0085
E160	1130	942	128	60	4498	3759	739	3.87	27.15	0.0103
E165	1500	1280	149	70	4537	3798	739	2.93	23.32	0.0093
E170	634	527	73	34	2231	1492	739	3.42	22.13	0.0106
E180	1079	905	118	55	4494	1537	2957	4.06	22.37	0.0164
E185	1538	1334	139	65	4534	1577	2957	2.86	22.34	0.0163
E190	164	138	18	8	578	208	370	3.54	21.99	0.0157
E195	251	218	23	11	601	232	370	2.38	21.93	0.0153
E200	1480	1253	155	73	5498	4277	1221	3.61	26.73	0.0113

Tables 15 through 19 show a great deviation between actual zone temperature and the set point, what follows a small change of the sensible zone load caused by heat flow through the building envelop. There are also differences of humidity ratio in comparison with results of subsection 3.3. The operating time depending on PLR is longer or shorter than the operation time of simulation with an ideal control. The energy consumption is dependent on the operating time. From the tables 15–19 it could be said that COP and IDB are a function of control method, including ideal control. Tables 20 and 21 give a summary of the behavior.

Table 20: Mean COP for different control methods

Control Method / Cases	Ideal	Real	#1	#2	#3	#4	#5
From table	8	8	15	16	17	18	19
Hysterese ON OFF	- - -	- - -	- - 5 min	- 5 min 5 min	2 K - 5 min	2 K 5 min 5 min	2 K - -
E100	2.40	2.42	2.54	2.54	2.50	2.50	2.42
E110	3.41	3.43	3.61	3.61	3.56	3.56	3.45
E120	3.61	3.63	3.81	3.81	3.76	3.76	3.64
E130	1.92	1.92	1.92	1.31	2.07	1.23	2.07
E140	2.80	2.80	2.80	1.27	2.84	1.20	2.84
E150	3.65	3.67	3.80	3.80	3.75	3.75	3.67
E160	3.85	3.86	4.01	4.01	3.96	3.96	3.87
E165	2.93	2.94	3.03	3.03	2.97	2.97	2.93
E170	3.39	3.40	3.49	3.32	3.44	3.35	3.42
E180	4.05	4.06	4.13	4.13	4.09	4.09	4.06
E185	2.85	2.86	2.92	2.92	2.88	2.88	2.86
E190	3.41	3.40	3.41	3.54	3.52	3.57	3.54
E195	2.32	2.31	2.31	2.46	2.38	2.50	2.38
E200	3.61	3.61	3.61	3.61	3.61	3.61	3.61

Table 21: Mean IDB for different control methods

Control Method / Cases	Ideal	Real	#1	#2	#3	#4	#5
From table	8	8	15	16	17	18	19
Hysterese ON OFF	- - -	- - -	- - 5 min	- 5 min 5 min	2 K - 5 min	2 K 5 min 5 min	2 K - -
E100	22.2	22.6	26.2	26.2	25.0	25.0	22.8
E110	22.2	22.5	26.5	26.5	25.3	25.3	22.8
E120	26.7	27.1	31.1	31.1	29.9	29.9	27.0
E130	22.2	21.6	21.6	17.7	21.3	18.5	21.3
E140	22.2	21.5	21.5	17.0	21.7	17.9	21.7
E150	22.2	22.7	25.7	25.7	24.5	24.5	22.6
E160	26.7	27.0	30.6	30.6	29.5	29.5	27.1
E165	23.3	23.8	26.0	26.0	24.6	24.6	23.3
E170	22.2	22.1	23.5	20.0	22.4	20.8	22.1
E180	22.2	22.3	23.9	23.9	22.9	22.9	22.4
E185	22.2	22.4	23.9	23.9	22.9	22.9	22.3
E190	22.2	21.9	21.9	19.8	22.0	20.8	22.0
E195	22.2	22.0	22.1	20.2	21.9	21.2	21.9
E200	26.7	26.7	26.7	26.7	26.7	26.7	26.7

8. Conclusions and Recommendations

The BESTEST is useful to improve the program code. With the experiences of BESTEST E100-E200 series it can be suggest to check program codes step by step. Only one process will be modeled with a component considering real behavior, while other processes should be idealized. So, the component modeling the real process will be easily checked.

For this test series, the near adiabatic building is used, so the effect of its dynamic behavior has not been analyzed. There is only return air operation. There is no zone humidity control. The equipment operates for all the time. The outside conditions are constant for the whole simulation period. The purpose for an installed air conditioning is to keep to room air conditions in the field of thermal comfort. So, the BESTEST would be very useful if we expand the diagnostics by adding further test series. In the end, this will help simulation programs determine what equipment should be installed for a given real building, real weather and a realistic control, so the inside air is in the area of thermal comfort.

Regarding the control strategy e.g. realistic control, limited operating time for ON/OFF-state and hysteresis round the set point, the comparison between them shows big differences. So, it could be said what has been done up to now is a far away for the application in reality.

The table 14 shows that the indoor humidity ratio depends on the sensible heat ratio of zone load, ODB and IDB. There is a close agreement to the results of simulation with an ideal control. The comparison of two tables 15 and 16 gives a big deviation for COP of E130 and E140. The analysis is only possible for the idealized conditions. In order to diagnose the real behavior of the equipment the only one way to do is simulation with realistic conditions.

For the further works in future it could be suggested to add following:

First round: Control tests (near-adiabatic building)

- There is maybe useful to add an additional output of operating time for E100–E200 to compare participants' results.
- Limitation of operating time ON/OFF-states,
- Hysteresis round the set point,
- Combination limitation of operating time and hysteresis,
- Operating schedule like daily operation from 7am – 6pm,
- Operation with a portion of out side air,
- Combination portion of out air and operating schedule,
- Combination all above-said cases.
- Proportional control (control of the equipment capacities for matching of the given loads

Second round: Operating method tests (near-adiabatic building)

- For summer: cooling
- For winter: heating (heat pump)

Third round: Equipment and building tests (real building and real weather)

- Consideration of the thermal and moisture storage in the building envelop for a continuous and intermittent operating

Fourth round: Coupling split equipment and heat recovery like an air to air heat exchanger for operating with a portion of out side air

9. References

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[2] HVAC BESTEST Summary of 2nd Set of Results Cases E100 – E200 (March 1999). Golden, CO: National Renewable Energy Laboratory, in conjunction with IEA SHC Task 22, Subtask A2, Comparative Studies.

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[4] Neymark, J.; Judkoff, R.: Building Energy Simulation Test and Diagnostic Method for

Mechanical Equipment (HVAC BESTEST), NREL, Golden, Colorado, USA 1998.

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[8] Neymark, J.: HVAC BESTEST: E100-E200 two minor but important points, email from Neymark to Participants, Jan 10, 2000 8:10PM.

[9] Le, H.-T.: Identification Characteristic Curves of cooling coil through simulation with a physical model, project thesis at TU Dresden April 1996.

Program name (please include version number)

TRNSYS-TUD

Your name, organisation, and country

H.-T. Le, Dresden University of Technology, Germany

Program status

	Public domain
	Commercial
x	Research and tuition
	Other (please specify)

Solution method for unitary space cooling equipment

x	Overall Performance Maps
x	Individual Component Models
a	Constant Performance (no possible variation with entering or ambient conditions)
	Other (please specify)

Interaction between loads and systems calculations

x	Both are calculated during the same timestep
a	First, loads are calculated for the entire simulation period, then equipment performance is calculated separately
	Other (please specify)

Time step

	Fixed within code (please specify time step)
x	User-specified (please specify time step)
	Other (please specify)

Timing convention for meteorological data: sampling interval

a	Fixed within code (please specify interval)
x	User-specified

Timing convention for meteorological data: period covered by first record

a	Fixed within code (please specify period or time which meteorological record covers)
x	User-specified

Meteorological data reconstitution scheme

a	Climate assumed stepwise constant over sampling interval
x	Linear interpolation used over climate sampling interval
	Other (please specify)

Output timing conventions

	Produces spot predictions at the end of each time step
	Produces spot output at end of each hour
x	Produces average outputs for each hour (please specify period to which value relates)

Treatment of zone air

x	Single temperature (i.e. good mixing assumed)
a	Stratified model
a	Simplified distribution model
a	Full CFD model
	Other (please specify)

Zone air initial conditions

x	Same as outside air
a	Other (please specify) what ever you want.

Internal gains output characteristics

a	Purely convective
	Radiative/Convective split fixed within code
x	Radiative/Convective split specified by user
a	Detailed modeling of source output

Mechanical systems output characteristics

a	Purely convective
	Radiative/Convective split fixed within code
X	Radiative/Convective split specified by user
a	Detailed modeling of source output

Control temperature

x	Air temperature
	Combination of air and radiant temperatures fixed within the code
a	User-specified combination of air and radiant temperatures
a	User-specified construction surface temperatures
	User-specified temperatures within construction
	Other (please specify)

Control properties

x	Ideal control as specified in the user's manual
a	On/Off thermostat control
a	On/Off thermostat control with hysteresis
a	On/Off thermostat control with minimum equipment on and/or off durations
a	Proportional control
a	More comprehensive controls (please specify) PID

Performance Map: characteristics

	Default curves
X	Custom curve fitting
	Detailed mapping not available
	Other (please specify)

Performance Map: independent variables

x	Entering Drybulb Temperature
x	Entering Wetbulb Temperature
x	Outdoor Drybulb Temperature
a	Part Load Ratio
a	Indoor Fan Air Flow Rate
	Other (please specify)

Performance Map: dependent variables

a	Coefficient of Performance (or other ratio of load to electricity consumption)
x	Total Capacity
x	Sensible Capacity
a	Bypass Factor
	Other (please specify)

Performance Map: available curve fit techniques

x	Linear, f(one independent variable)
a	Quadratic, f(one independent variable)
a	Cubic, f(one independent variable)
a	Bi-Linear, f(two independent variables)
a	Bi-Quadratic, f(two independent variables)
a	Other (please specify) what ever you want.

Performance Map: extrapolation limits

a	Limits independent variables
a	Limits dependent variables
x	No extrapolation limits
a	Extrapolation not allowed
	Other (please specify)

Cooling coil and supply air conditions model

a	Supply air temperature = apparatus dew point (ADP); supply air humidity ratio = humidity ratio of saturated air at ADP
a	Bypass factor model using listed ADP data
a	Bypass factor model with ADP calculated from extending condition line
a	Fan heat included
x	More comprehensive model; energy balance over zone boundary

Disaggregation of fans' electricity use directly in the simulation and output

a	Indoor fan only
a	Outdoor fan only
a	Both indoor and outdoor fans disaggregated in the output
x	None - disaggregation of fan outputs with separate calculations by the user considering CDF

Economizer settings available (for E400 series)

x	Temperature
a	Enthalpy
a	Compressor Lockout
	Other (please specify)

Appendix III-C

CLIM2000 V2.1.6 & CLIM2000 V2.4

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December 2000

1 Introduction

The studies were carried out with the versions 2.1 and 2.4 of the CLIM2000 software program.

The CLIM2000 software environment was developed by Electricity Applications in Residential and Commercial Buildings Branch in Research and Development Division of the French utility company EDF (Electricité De France). This software operational since June 1989, allows the behaviour of an entire building to be simulated. Its main objective is to produce energy cost studies, pertaining to energy balances over long periods as well as more detailed physical behaviour studies including difficult non-linear problems and varied dynamics. The building is described by means of a graphics editor in the form of a set of icons representing the models chosen by the user and taken from a library containing about 150 elementary models.

For EDF, IEA Task22 is a good opportunity to compare CLIM2000 results with others building energy analysis tools available, as done in the past in the framework of IEA Annex 21.

2 Modeling

Four different modelings were used in this study:

- The first modelings used old elementary model of mechanical system available into the CLIM2000 library. As we explain after in the remainder of this paper, we knew when beginning the modelling that the results will be poor. Indeed, this elementary model was used in an EDF validation exercise and the agreement between measurements and calculated results was not excellent.
- The second modelings used new elementary model of mechanical system developed recently to be in accordance with the new commercial policy of EDF.
- The third modelings used the same new elementary model but modified further to the analysis made by NREL team (subtask leader).
- The fourth modeling used the same new elementary model but modified further to the last analysis made by NREL team (subtask leader).

The sections hereafter give a detailed description of the assumptions made and of the modeling.

2.1 Common modeling

We present in this section the common parts of the four modeling. The main differences between them are located in the elementary model of mechanical system and in the controller.

2.1.1 Building envelope

Taking into account that the constitution of the walls are identical, and that the building can be supposed sufficiently adiabatic, we considered that it is not influenced by solar radiation (very thick insulation). Only the outdoor temperature was varied for the different tests, and no other meteorological data was used.

By defining a single zone building with identical properties for all bounding surfaces, the "whole" model could be used to simulate a one dimensional single wall. We preferred to model the building by defining its six walls to allow the second series of tests.

Each wall is discretized by using 5 layers of insulation (e=0.2m ; λ=0.01 W/m.K ; ρ= 16 kg/m^3; Cp=800 J/kg.K). The internal and external surface coefficients are given in the test specification (i.e. hi=8.29 W/m².K, he=29.3 W/m².K).

2.1.2 Air volume

The air volume is represented by a specific elementary model taking into account temperature, pressure and relative humidity or humidity ratio as state variables of the system.

To allow the pressure calculations, it is necessary to use this model with a very little ventilation rate of humid air. This ventilation rate was set so that the latent and sensible loads due to this infiltration is very very low and negligible when comparing to internal latent and sensible loads as specified in the tests specification.

2.1.3 Latent and sensible loads

Sensible and latent loads were represented by using the appropriate elementary models (purely convective heat source and vapour injection respectively). The vapour rate is calculated with the following equation :

$$\boxed{m_v = \frac{P}{2500800 + (1846 * Tset)}}$$

where m_v is vapor rate in kg/s ; P is latent load in W ; Tset is set point temperature in °C.

A step function is connected to these models for not injecting loads for the numerical convergence of the steady state (each CLIM2000 simulation begins with a steady state calculation, all derivatives are set to zero) and for applying the loads for the transient state. The loads are applied at t=1s.

2.2 Mechanical system and controller

We present in this section the differences between the different modelling in terms of mechanical system and controller.

2.2.1 First modelling

As written previously, this modelling used old elementary model available into the CLIM2000 library version 2.1 **[1]**. Because we wanted not to miss the first round of simulations of HVAC BESTEST procedure, we used this model even if we knew before running the simulations that the results will be bad. Indeed, we used these models in the framework of an EDF validation exercise, and we found that the agreement between calculations and experimental data was not excellent.

This modelling is based on the test specification (September 1997).

2.2.1.1 Mechanical system and controller

The system is represented by using an old elementary model of a split system. This model represents the response of the system, for which only the evaporator side is modelled. This model is based on experimental tests and its use is very simplified. The heat exchange between evaporator and air volume is based on heat exchanger equations : an air flow rate, and a convective exchange between the tubes and air. The heat exchange coefficient varies with the air flow rate of the ventilator, that can be controlled, with the typology of fins. The apparition of dew point is taken into account by comparing the dew point of air volume and the air temperature after the evaporator.

The other side of the system (compressor, condenser) is simply described by a coefficient of performance (COP) varying with the outdoor temperature. So that, it was very difficult to implement correctly the performance maps given in the test specification into that model.

We used a PID elementary model. We used it only as PI. The proportional band was set to a very low value (10^{-8}) to approximate a non-proportional thermostat as required in the test specification. Integration time was set to 60s to prevent numerical problems when the cooling system switches on.

2.2.1.2 Variant

These tests were performed using **two sets of models** to describe the air volume, the air circulation, the latent heat loads, and the split cooling system.

These two sets of models are described with equivalent heat balances. The differences between these models are based on the state variables of the global system:

- for the first set, the state variables are the temperature, the pressure, and the relative humidity;
- for the second set, the state variables are the temperature, the pressure, and the humidity ratio;

The second set of models (absolute humidity) is bases on the first set of models, with addition of functions to translate relative humidity into humidity ratio, and all equations of the models are homogeneous.

2.2.2 Second modeling

In the beginning of 1998, it was planned to develop this model to carry out classical economical studies to be in accordance with the new commercial policy of EDF (thermodynamical systems). Unfortunately, this model was not available when the first round of simulations was analysed on April1998. That is why we used the first modeling in the first round. For the second round of simulation, the new elementary model was available.

The modeling to describe the building envelope, the air volume, the sensible and latent loads is the same as the first modeling presented in Section 2.1. The main differences between the first and the second model lings are in the elementary models used to describe the mechanical system and the controller. Then, we just present hereafter these two elementary models.

This modeling used the last version of models library of CLIM2000 **[2] [3] [4]**. The state variables of the system to be solved are temperature, pressure and humidity ratio.

The model used here is the one described in **[5]** .

This model is based on performances map. This first version of that model did not take into account the cooling equipment part load performance (COP degradation factor). In this model, extrapolation related to EWB (entering wet bulb) are not allowed i.e. if the operation point appears to be outside the map, then the model used the performances calculated on the basis of the closest values to the operation point. But, extrapolation related to EDB (entering dry bulb) is allowed. In order to simplify the model and because the performance maps given by the manufacturers are often incomplete with few data, three values of EDB are available. For the HVAC BESTEST, we selected the three following values : 22.2, 24.4 and 26.7°C. The calculation of EWB is based on the ASHRAE formula.

We used a PID elementary model. We used it only as P. The proportional band was set to a very low value (10^{-8}) to approximate a non-proportional thermostat as required in the test specification.

2.2.3 Third modeling

The summary of 2nd set of results **[10]** pointed out the better behaviour of CLIM2000 except for COP sensitivity to PLR. Then, the elementary model used to describe the mechanical equipment was modified to take into account this COP Degradation factor. We can say that from that 2nd round, our participation to the HVAC BESTEST was very useful to improve our elementary model.

The remainder of the global model (building, loads, etc..) was not modified. We kept the modelling made in the previous stages.

We carried out two sets of simulation with no extrapolation from performance map and with extrapolation to see the impact of that. The considered extrapolation is linear. In fact, we modified by hand the performance maps by replacing the data related to EWB=15°C (Table 1-6a in **[8]**) by data calculated for EWB=12.8. We considered that the performance data (net total capacity, net sensible capacity, compressor power) follow the same slope between 12.8-15.0°C than between 15-17.2°C.

2.2.4 Fourth modeling

The base of the split system modelling is the third modelling (see above). We carried out a set of simulation with two extrapolations from performance map.

We integrated a new performance map to indicate the limits of performance of split in dry coil conditions (Table A).

First extrapolation (data related to EWB < 13°C and EWB > 21,7°C)

After finding improved results for E110 and E100, the extrapolation of EWB was automated. Results using automated extrapolation of EWB agreed with the manually extrapolated results.

The considered extrapolation is now automatic and linear. We considered that the performance data (net total capacity, net sensible capacity, compressor power) follow the same slope under EWB = 15.0°C (above EWB = 21,7°C) than between 15-17.2°C (19,4-21,7°C).

Second extrapolation (data related to EDB = 27.8°C)

We add the data related to EDB = 27.8°C. The considered extrapolation is linear. In fact, we modified the performance maps by adding the data calculated for EDB=27.8°C (Table 1-6e in **[8]**). We considered that the performance data (net sensible capacity) follow the same slope between 26.7-27.8°C than between 25.6-26.7°C.

The final performance map applying manual extrapolation of EDB is included in Table B.

Table A: Dry Coil Performance Limits

Dry outside temperature (°C)	Dry inside temperature (°C)	EWB limit (°C)	P limit (kW) *
29,4	22,2	13,82	6,87
29,4	24,4	15,34	7,28
29,4	26,7	16,75	7,66
29,4	27,8	17,41	7,84
32,2	22,2	13,9	6,73
32,2	24,4	15,47	7,13
32,2	26,7	16,96	7,51
32,2	27,8	17,7	7,71
35	22,2	14,11	6,59
35	24,4	15,67	6,98
35	26,7	17,08	7,34
35	27,8	17,77	7,52
40,6	22,2	14,38	6,29
40,6	24,4	15,88	6,63
40,6	26,7	17,41	6,98
40,6	27,8	18,13	7,15
46,1	22,2	14,7	5,96
46,1	24,4	16,36	6,31
46,1	26,7	17,69	6,6
46,1	27,8	18,32	6,74

* P = adjusted net total capacity

Table B: performance map used in the modeling

Dry outside temperature (°C)	EWB (°C)	Adjusted net total capacity	Compressor power (kW)	Adjusted net sensible capacity (kW)			
		(kW)		EDB = 22,2°C	EDB = 24,4°C	EDB = 26,7°C	EDB = 27,8°C
29,4	15	7190	1620	6310	7280	7660	7840
29,4	17,2	7780	1660	5260	6370	7450	7840
29,4	19,4	8420	1710	4110	5230	6310	6870
29,4	21,7	9060	1760	2970	4050	5140	5700
32,2	15	7010	1690	6220	7130	7510	7710
32,2	17,2	7570	1740	5200	6280	7370	7710
32,2	19,4	8190	1790	4030	5140	6220	6770
32,2	21,7	8800	1840	2850	3970	5050	5610
35	15	6810	1770	6160	6980	7340	7520
35	17,2	7370	1810	5110	6190	7280	7520
35	19,4	7950	1860	3940	5050	6130	6680
35	21,7	8570	1910	2770	3850	4960	5510
40,6	15	6430	1940	5990	6630	6980	7150
40,6	17,2	6930	1980	4930	6020	6980	7150
40,6	19,4	7450	2020	3760	4850	5930	6460
40,6	21,7	8010	2060	2590	3670	4760	5290
46,1	15	6020	2110	5810	6310	6600	6740
46,1	17,2	6490	2140	4760	5840	6600	6740
46,1	19,4	6980	2180	3590	4670	5750	6270
46,1	21,7	7480	2210	2410	3500	4580	5130

3 Results

We produced 6 sets of results: BESTEST1, BESTEST2, BESTEST3, BESTEST4, BESTEST5 and BESTEST6. They are described in the next sections.

3.1 First modeling

The results obtained with the first modeling and the two sets of models as described in subsection 2.2.1 are given in Tables 1 and 2. These results were sent on March 1998. The electronic file containing the results was named: hvbtout2.xls.

Table 1: CLIM2000 results - BESTEST1 (first modeling, first set of models)

	February totals										February mean		
		Cooling energy consumption			Evaporator coil load			Enveloppe load					
Cases	Total	Compressor	Supply fan	Cond. fan	Total	Sensible	Latent	Total	Sensible	Latent	COP	IDB	Hum. ratio
	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh		°C	kg/kg
E100	1428.54	1228.95	135.81	63.77	3657.00	3656.89	0.11	3657.14	3657.03	0.11	2.46	22.21	8.47E-03
E110	952.22	794.62	107.24	50.36	3637.57	3637.40	0.17	3637.76	3637.59	0.17	3.71	22.21	6.41E-03
E120	917.21	784.19	90.52	42.50	3632.22	3632.09	0.13	3632.35	3632.22	0.13	3.86	26.71	7.18E-03
E130	82.09	76.23	3.99	1.87	210.15	209.80	0.35	210.51	210.16	0.35	2.51	22.20	3.73E-03
E140	49.91	45.52	2.98	1.40	190.65	190.26	0.38	191.04	190.66	0.38	3.76	22.20	2.47E-03
E150	1339.36	1112.23	154.56	72.58	5116.11	4376.53	739.58	5277.68	4376.71	900.98	3.70	22.60	7.87E-03
E160	1290.71	1094.57	133.47	62.67	5111.02	4371.49	739.53	5274.61	4371.48	903.13	3.86	26.71	9.08E-03
E165	1732.21	1505.08	154.56	72.58	5127.13	4387.62	739.51	5289.76	4387.65	902.11	2.87	24.84	9.52E-03
E170	758.96	662.42	65.69	30.84	2898.92	2159.27	739.65	3060.34	2159.50	900.84	3.73	22.20	6.63E-03
E180	1920.84	1693.71	154.56	72.58	7337.00	4380.70	2956.30	7992.40	4380.61	3611.79	3.74	27.40	1.10E-02
E185													
E190	243.45	224.93	12.60	5.92	929.91	559.89	370.02	1010.76	560.17	450.59	3.77	22.20	4.58E-03
E195	370.88	347.21	16.11	7.57	949.37	579.41	369.97	1030.17	579.64	450.54	2.52	22.20	6.34E-03
E200	1859.69	1632.56	154.56	72.58	6508.67	5286.89	1221.78	6781.88	5286.82	1495.06	3.42	29.96	1.11E-02

Table 2 : CLIM2000 results - BESTEST2 (first modeling, second set of models)

	February totals										February mean		
		Cooling energy consumption			Evaporator coil load			Enveloppe load					
Cases	Total	Compressor	Supply fan	Cond. fan	Total	Sensible	Latent	Total	Sensible	Latent	COP	IDB	Hum. ratio
	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh		°C	kg/kg
E100	1315.24	1204.87	75.10	35.26	3656.45	3656.42	0.03	3656.45	3656.42	0.03	2.72	22.20	4.27E-03
E110	902.50	792.84	74.62	35.04	3637.16	3637.13	0.03	3637.20	3637.16	0.03	3.95	22.20	4.04E-03
E120	901.22	818.92	56.00	26.30	3631.99	3631.96	0.04	3632.02	3631.98	0.04	3.97	26.70	3.30E-03
E130	75.21	70.79	3.01	1.41	209.09	209.05	0.04	209.13	209.09	0.04	2.74	22.20	2.20E-03
E140	47.10	43.11	2.72	1.28	189.83	189.79	0.04	189.87	189.83	0.04	3.97	22.20	2.06E-03
E150	1266.65	1116.89	101.91	47.85	5104.50	4376.74	727.76	5116.39	4376.76	739.63	3.95	22.21	4.86E-03
E160	1264.30	1153.44	75.44	35.42	5094.96	4370.55	724.40	5109.19	4370.57	738.63	3.97	26.71	4.11E-03
E165	1608.03	1469.45	94.30	44.28	5113.37	4387.29	726.08	5125.75	4387.26	738.49	3.12	23.31	4.85E-03
E170	716.42	650.24	45.04	21.15	2886.90	2159.14	727.76	2898.80	2159.17	739.63	3.97	22.20	4.14E-03
E180	1808.02	1617.32	129.77	60.93	7285.44	4376.20	2909.24	7333.02	4376.27	2956.75	3.96	22.21	6.52E-03
E185													
E190	228.76	214.38	9.78	4.59	921.79	558.73	363.06	927.74	558.76	368.98	3.99	22.20	3.09E-03
E195	338.54	323.66	10.13	4.76	941.05	577.99	363.06	947.01	578.03	368.98	2.75	22.20	3.27E-03
E200	1772.20	1626.95	98.84	46.41	6485.98	5287.63	1198.35	6509.50	5287.64	1221.86	3.60	26.71	4.83E-03

For BESTEST1, some set points for internal temperature can't be reached. Test E185 could not be performed. For BESTEST2, all set points for internal temperature can be reached, and test E185 could not be performed.

For BESTEST1 and BESTEST2, we could not provide any results for the case E185. The simulation aborted, because of a numerical problem. For this case, the cooling system is under over-load conditions. So the internal temperature increases continuously, and the injected vapour is not sufficiently removed by condensation over the evaporator coil. These results in very non-realistic physical conditions, and trends to a non-stable numerical system.

Other cases show that the system's operation is under over-load conditions with BESTEST1 (cases E150, E165, E180, E200). For these cases, the set-point can't be reached, but the (Temperature, Pressure, Humidity) conditions of the air volume reach a liable state.

The comparison of these two sets of results with the results of IEA Task22 participants **[6]** was as we expected when beginning the modeling: very bad results for CLIM2000 for the two sets of models.

3.2 Second modeling

The results obtained with the second modeling as described in subsection 2.2.2 are given in Table 3. These results were sent on February 1999. The electronic file containing the results was named: resuEDF0299.wk3 + resuEDF0299.fm3.

Table 3: CLIM2000 results -BESTEST3 (second modeling)

	February Totals										February Mean			February Maximum			February Minimum		
	Cooling Energy Consumption				Evaporator Coil Load			Envelope Load											
Cases	Total (kWh)	Compressor (kWh)	Supply Fan (kWh)	Condenser Fan (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)
E100	1567	1351	147	69	3803	3803	0	3656	3656	0	2.33	22.2	0.00692	2.33	22.2	0.007	2.33	22.2	0.0068
E110	1147	949	135	63	3772	3772	0	3637	3637	0	3.17	22.2	0.00692	3.17	22.2	0.007	3.17	22.2	0.0068
E120	956	792	111	52	3743	3743	0	3632	3632	0	3.8	26.7	0.00692	3.8	26.7	0.007	3.8	26.7	0.0068
E130	89	77	8	4	217	217	0	209	209	0	2.35	22.2	0.00692	2.35	22.2	0.007	2.35	22.2	0.0068
E140	60	50	7	3	197	197	0	190	190	0	3.17	22.2	0.00692	3.17	22.2	0.007	3.17	22.2	0.0068
E150	1193	988	140	65	4516	3777	739	4376	3637	739	3.67	22.2	0.00851	3.67	22.2	0.0085	3.67	22.2	0.0085
E160	1100	917	125	59	4496	3757	739	4371	3632	739	3.97	26.7	0.0101	3.97	26.7	0.0101	3.97	26.7	0.0101
E165	1510	1289	150	71	4538	3799	739	4388	3649	739	2.91	23.3	0.0098	2.91	23.3	0.0098	2.91	23.3	0.0098
E170	559	465	64	30	2222	1483	739	2158	1419	739	3.76	22.2	0.0107	3.76	22.2	0.0107	3.76	22.2	0.0107
E180	1025	860	112	53	4488	1531	2957	4376	1419	2957	4.27	22.2	0.0162	4.27	22.2	0.0162	4.27	22.2	0.0162
E185	1524	1322	138	65	4534	1577	2957	4396	1419	2957	2.88	22.2	0.0169	2.88	22.2	0.0169	2.88	22.2	0.0169
E190	132	110	14	7	574	204	370	560	190	370	4.24	22.2	0.0159	4.24	22.2	0.0159	4.24	22.2	0.0159
E195	203	176	18	9	597	227	370	579	209	370	2.85	22.2	0.0163	2.85	22.2	0.0163	2.85	22.2	0.0162
E200	1478	1250	155	73	5438	4217	1221	5283	4062	1221	3.57	26.8	0.0115	3.57	26.75	0.0115	3.57	26.75	0.0115

The summary of 2nd set of results [10] pointed out the better behaviour of CLIM2000 except for COP sensitivity to PLR. All CLIM2000 results except COP sensitivity were comparable to the results of the others software programs.

The interest of a comparative study is to evaluate your own software program with others. In that round, CLIM2000 results obtained with the new elementary model developed for the EDF own needs not for HVAC BESTEST comparison, are good and comparable with the results from others software programs.

All the setpoint temperatures were matched and we obtained results for all test cases.

The Figure 1 presents the CLIM2000 results in terms of mean COP sensitivities. We can see too low COP=f(PLR) sensitivity for E130-E100, E140-E110, E170-E150, E190-E180 and E195-E185. It is also mentioned in [10] that CLIM2000 results presents a slight opposite sensitivity for E170-E150, and that this could be caused by the PLR insensitivity.

This comparative study pointed out the fact that the CLIM2000 model did not have COP sensitivity to part load ratio. This was normal because no COP degradation factor was implemented into the model. After this round, it was decided to implement it to improve the model and to rerun all the test cases, just to be sure that this modification do not impact the high part load ratio test cases.

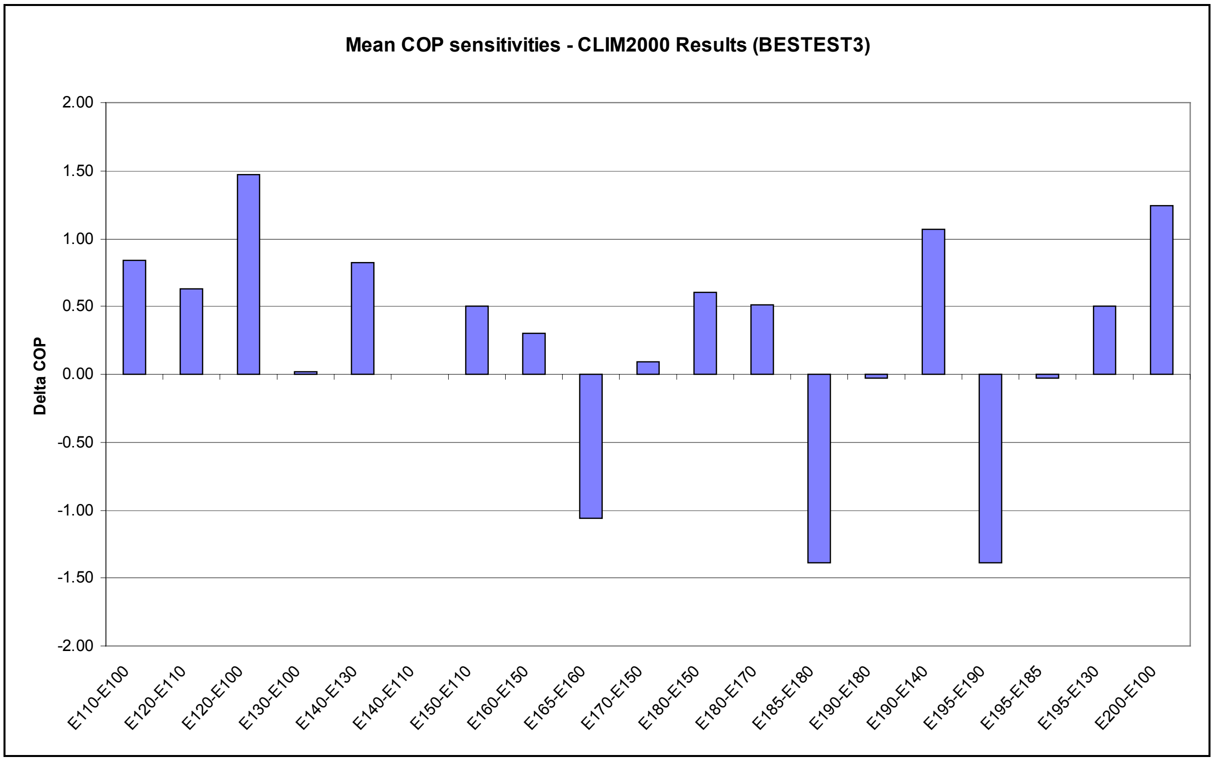

Figure 1: Mean COP sensitivities – CLIM2000 results (BESTEST3)

3.3 Third modeling

The results obtained with the third modeling as described in subsection 2.2.3 are given in Table 5 and 6. These results were sent on May 1999. The electronic files containing the results was named:

- resuEDF0599wext.wk3 + resuEDF0599wext.fm3 for with no extrapolation in performance maps.
- resuEDF0599ext.wk3 + resuEDF0599ext.fm3

As expected, the energy consumption for all test cases except E200 is higher with these modelling than with the previous one. This result is normal because the high PLR is never superior or equal to 1.

We have of course a better COP sensitivity.

These results are compared to the previous ones in the next section.

An interesting thing to note is the influence of extrapolation. EWB is the only variable that goes outside the performance map, not too far. The EWB values for the impacted cases are given in Table 4.

Then as we can see in that table, we only found few test cases impacted by the extrapolation. The cases impacted are: E100, E110, E130 and E140. For those cases, EWB goes outside the performance data (in performance maps given in [8], EWB varies from 15°C to 21.7°C).

Table 4: EWB values vs test cases

Test cases	EWB value (°C)
E100	13.6
E110	14.1
E120	15.8
E130	13.6
E140	14.1
E150	15.5
E160	18.5
E165	16.7
E170	17.4
E180	21.6
E185	21.5
E190	21.4
E195	21.1
E200	19.4

Table 5: CLIM2000 Results – COP Degradation factor added, no extrapolation – BESTEST4

	February Totals										February Mean			February Maximum			February Minimum		
	Cooling Energy Consumption				Evaporator Coil Load			Envelope Load											
Cases	Total (kWh)	Compressor (kWh)	Supply Fan (kWh)	Condenser Fan (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)
E100	1585	1366	149	70	3805	3805	0	3656	3656	0	2.31	22.2	0.00692	2.31	22.2	0.007	2.31	22.2	0.0068
E110	1181	978	139	65	3776	3776	0	3637	3637	0	3.08	22.2	0.00692	3.08	22.2	0.007	3.08	22.2	0.0068
E120	1021	846	119	56	3751	3751	0	3632	3632	0	3.56	26.7	0.00692	3.56	26.7	0.007	3.56	26.7	0.0068
E130	114	99	10	5	219	219	0	209	209	0	1.83	22.2	0.00692	1.83	22.2	0.007	1.83	22.2	0.0068
E140	77	63	9	4	199	199	0	190	190	0	2.47	22.2	0.00692	2.47	22.2	0.007	2.47	22.2	0.0068
E150	1220	1010	143	67	4519	3780	739	4376	3637	739	3.59	22.2	0.00851	3.59	22.2	0.00851	3.59	22.2	0.0085
E160	1151	959	131	61	4502	3763	739	4371	3632	739	3.8	26.7	0.0101	3.8	26.7	0.0101	3.8	26.7	0.0101
E165	1519	1297	151	71	4539	3800	739	4388	3649	739	2.89	23.3	0.0098	2.89	23.3	0.0098	2.89	23.3	0.0098
E170	646	537	74	35	2232	1493	739	2158	1419	739	3.34	22.2	0.0107	3.34	22.2	0.0107	3.34	22.2	0.01069
E180	1093	917	120	56	4496	1539	2957	4376	1419	2957	4	22.2	0.0162	4	22.2	0.0162	4	22.2	0.0162
E185	1563	1356	141	66	4537	1580	2957	4396	1419	2957	2.81	22.2	0.0169	2.81	22.2	0.0169	2.81	22.2	0.01685
E190	166	139	18	8	578	208	370	560	190	370	3.37	22.2	0.0159	3.37	22.2	0.0159	3.37	22.2	0.0159
E195	254	220	23	11	602	232	370	579	209	370	2.28	22.2	0.0163	2.28	22.2	0.0163	2.28	22.2	0.0162
E200	1478	1250	155	73	5438	4217	1221	5283	4062	1221	3.57	26.75	0.0115	3.57	26.75	0.0115	3.57	26.75	0.0115

Table 6: CLIM2000 results – COP Degradation factor included, with extrapolation – BESTEST5

Cases	February Totals										February Mean			February Maximum			February Minimum		
	Cooling Energy Consumption				Evaporator Coil Load			Envelope Load											
	Total (kWh)	Compressor (kWh)	Supply Fan (kWh)	Condenser Fan (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)
E100	1432	1233	136	64	3792	3792	0	3656	3656	0	2.55	22.2	0.00692	2.55	22.2	0.007	2.55	22.2	0.0068
E110	1182	978	139	65	3776	3776	0	3637	3637	0	3.08	22.2	0.00692	3.08	22.2	0.007	3.08	22.2	0.0068
E120	1021	846	119	56	3751	3751	0	3632	3632	0	3.56	26.7	0.00692	3.56	26.7	0.007	3.56	26.7	0.0068
E130	101	87	10	5	219	219	0	209	209	0	2.07	22.2	0.00692	2.07	22.2	0.007	2.07	22.2	0.0068
E140	70	58	8	4	198	198	0	190	190	0	2.71	22.2	0.00692	2.71	22.2	0.007	2.71	22.2	0.0068
E150	1220	1010	143	67	4519	3780	739	4376	3637	739	3.59	22.2	0.00851	3.59	22.2	0.00851	3.59	22.2	0.0085
E160	1151	959	131	61	4502	3763	739	4371	3632	739	3.8	26.7	0.0101	3.8	26.7	0.0101	3.8	26.7	0.0101
E165	1519	1297	151	71	4539	3800	739	4388	3649	739	2.89	23.3	0.0098	2.89	23.3	0.0098	2.89	23.3	0.0098
E170	646	537	74	35	2232	1493	739	2158	1419	739	3.34	22.2	0.0107	3.34	22.2	0.0107	3.34	22.2	0.01069
E180	1093	917	120	56	4496	1539	2957	4376	1419	2957	4	22.2	0.0162	4	22.2	0.0162	4	22.2	0.0162
E185	1563	1356	141	66	4537	1580	2957	4396	1419	2957	2.81	22.2	0.0169	2.81	22.2	0.0169	2.81	22.2	0.01685
E190	166	139	18	8	578	208	370	560	190	370	3.37	22.2	0.0159	3.37	22.2	0.0159	3.37	22.2	0.0159
E195	254	220	23	11	602	232	370	579	209	370	2.28	22.2	0.0163	2.28	22.2	0.0163	2.28	22.2	0.0162
E200	1478	1250	155	73	5438	4217	1221	5283	4062	1221	3.57	26.8	0.0115	3.57	26.75	0.0115	3.57	26.75	0.0115

3.4 Fourth modeling

The results obtained with the fourth modeling are given in Table 7. These results were sent on December 2000. The electronic files containing the results was named:

- resultatsBTclim2000_2.xls

The extrapolation of the values of the performance data is now automatic for variation of EWB outside the performance map.

The energy consumption for all test cases (except E200) is equal with these modeling to the previous one. Concerning this case, we add a little part on the modelization. The aim of this code revision is to limit the extrapolation of the split system power at the beginning of the simulation. These results are compared to the previous ones in the next section.

An interesting thing to note is the influence of extrapolations (the performance maps used are reported in the file resultatsBTclim2000.xls). It concerns most of the cases.

Table 7 : CLIM2000 results – COP Degradation factor included, with extrapolations – BESTEST6

cases	February Totals										February Mean			February Maximum			February Minimum		
	cooloing energy Consumption				Evaporator coil Load			envelope load											
	Total kWh	compressor kWh	supply fan kWh	condenser fan kWh	total kWh	sensible kWh	latent kWh	total kWh	sensible kWh	latent kWh	COP	IDB °C	humidity ratio kg/kg	COP	IDB °C	humidity ratio kg/kg	COP	IDB °C	humidity ratio kg/kg
E100	1530	1318	144	68	3800	3800	0	3656	3656	0	2,39	22,2	0,0069	2,39	22,2	0,0070	2,40	22,2	0,0068
E110	1089	899	129	61	3766	3766	0	3637	3637	0	3,34	22,2	0,0069	3,36	22,2	0,0070	3,32	22,2	0,0068
E120	1012	840	117	55	3749	3749	0	3632	3632	0	3,59	26,7	0,0070	3,60	26,7	0,0070	3,58	26,7	0,0070
E130	109	94	10	5	219	219	0	209	209	0	1,91	22,2	0,0069	1,92	22,2	0,0069	1,85	22,2	0,0068
E140	69	57	8	4	198	198	0	190	190	0	2,73	22,2	0,0069	2,83	22,2	0,0069	2,68	22,2	0,0068
E150	1207	999	141	66	4517	3778	739	4376	3637	739	3,63	22,2	0,0085	3,63	22,2	0,0085	3,62	22,2	0,0085
E160	1139	949	129	61	4500	3761	739	4371	3632	739	3,84	26,7	0,0101	3,85	26,7	0,0101	3,83	26,7	0,0101
E165	1501	1281	150	70	4538	3798	739	4388	3649	739	2,92	23,3	0,0099	2,93	22,2	0,0099	2,92	22,2	0,0098
E170	638	530	73	34	2232	1493	739	2159	1420	739	3,39	22,2	0,0107	3,40	22,2	0,0107	3,38	22,2	0,0107
E180	1082	908	119	56	4495	1538	2957	4376	1420	2957	4,04	22,2	0,0164	4,05	22,2	0,0164	4,04	22,2	0,0164
E185	1543	1339	139	65	4535	1578	2957	4396	1439	2957	2,85	22,2	0,0171	2,85	22,2	0,0171	2,84	22,2	0,0170
E190	164	138	18	9	577	208	370	559	190	370	3,41	22,2	0,0161	3,45	22,2	0,0161	3,37	22,2	0,0161
E195	250	217	23	11	601	232	370	579	209	370	2,31	22,2	0,0164	2,33	22,2	0,0165	2,29	22,2	0,0164
E200	1464	1239	153	72	5436	4215	1221	5283	4062	1221	3,61	26,7	0,0115	3,61	26,7	0,0115	3,61	26,7	0,0115

3.5 Comparison between different results

The comparison of Clim2000 results is just presented in terms of energy consumption and mean COP sensitivity. The BESTEST1 and BESTEST2 results are not integrated here, they were too poor

The results are presented in figures 2, 3, 4, 5, 6 and 7.

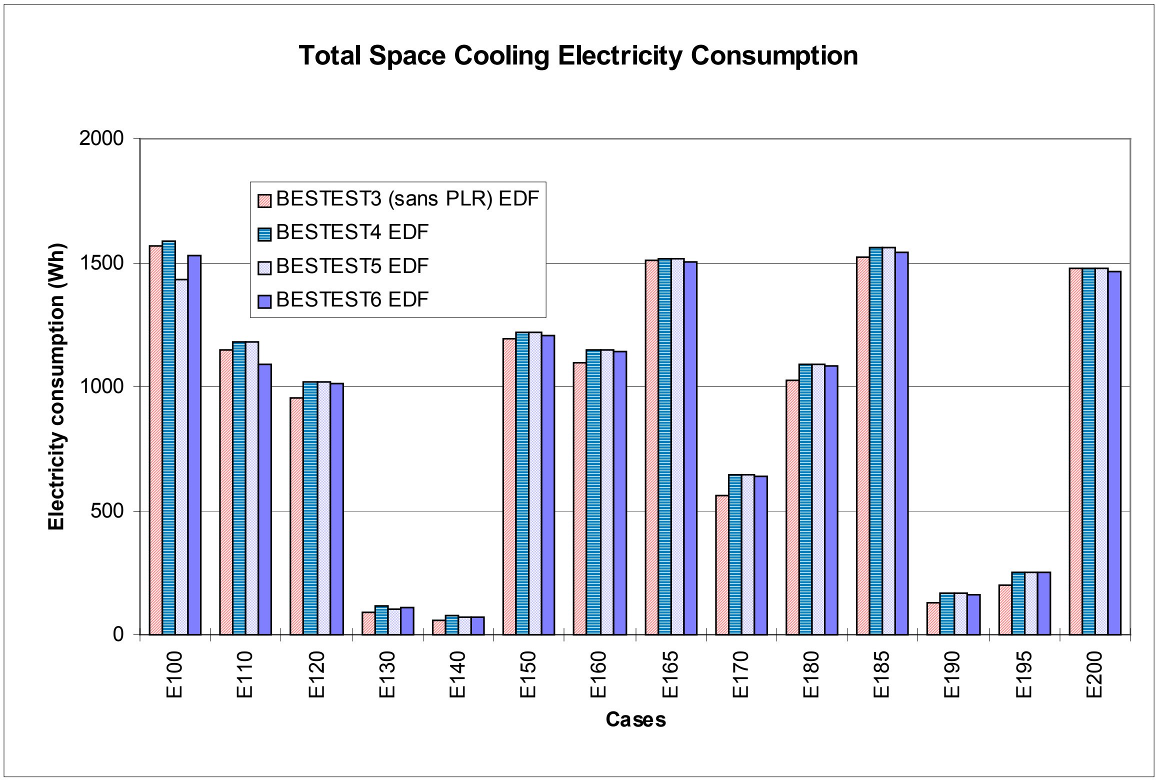

Figure 2: Total space cooling electricity consumption

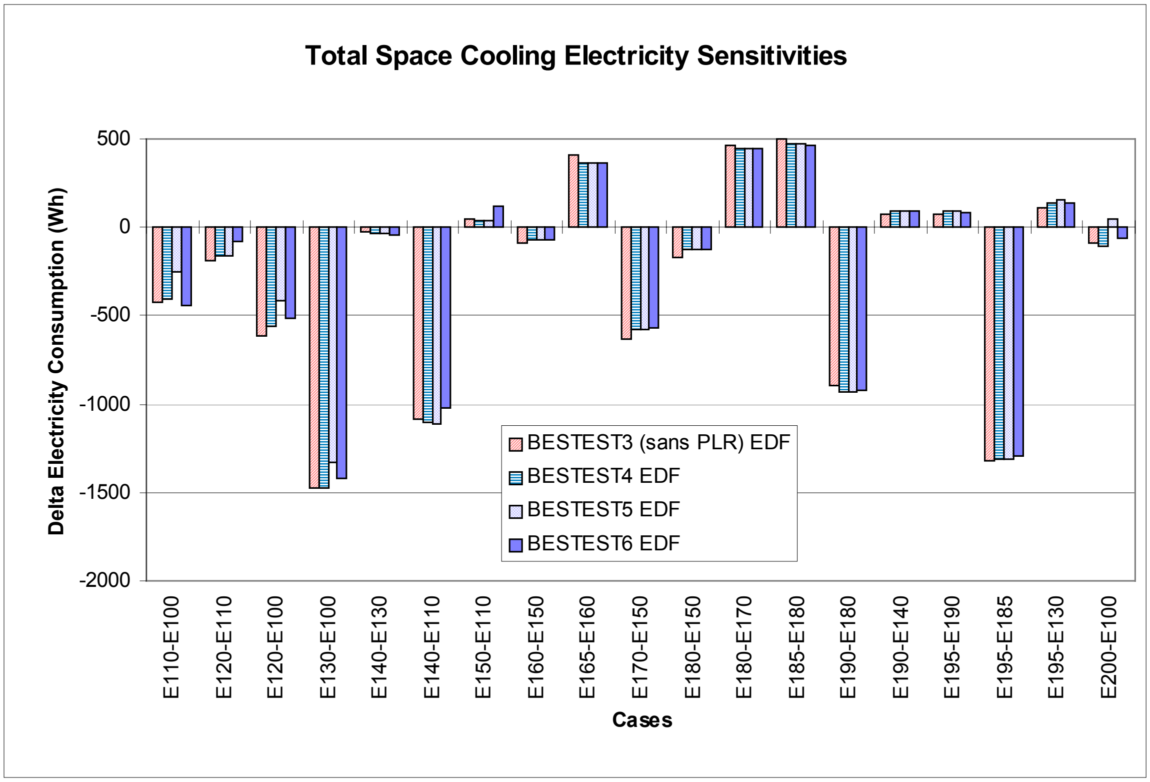

Figure 3: total space cooling electricity sensitivities

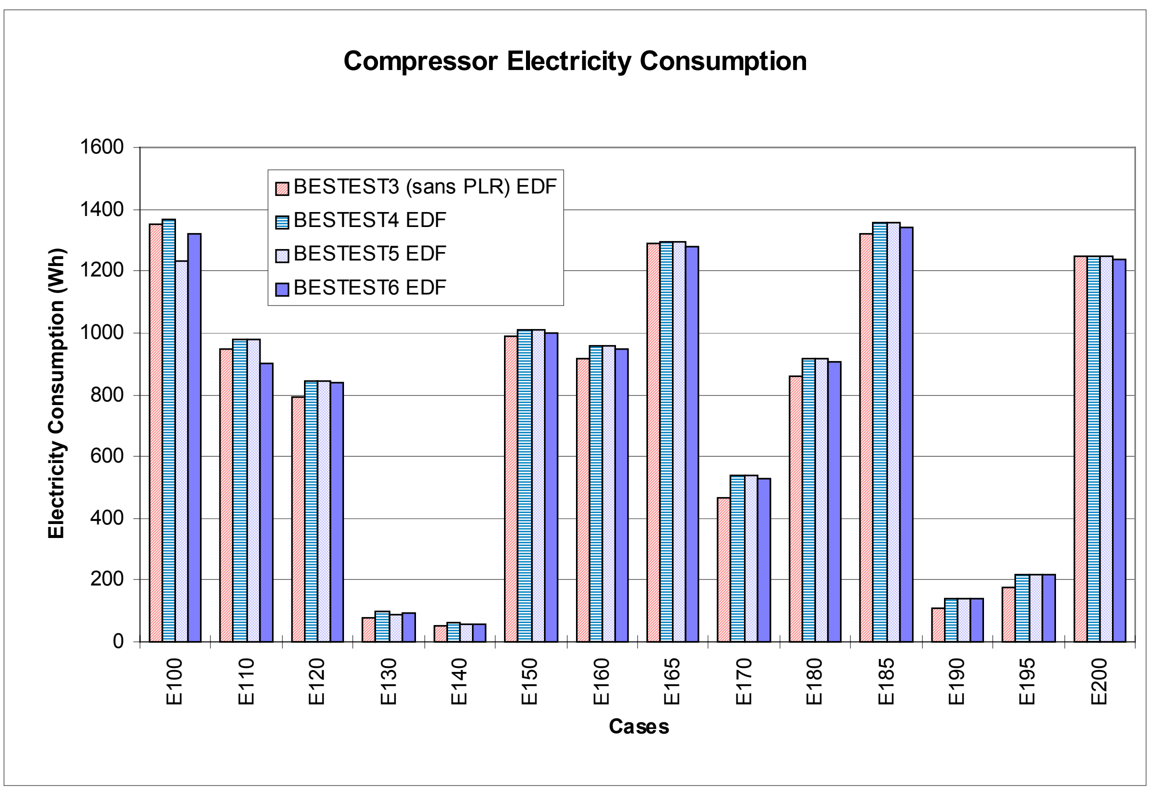

Figure 4: compressor electricity consumption

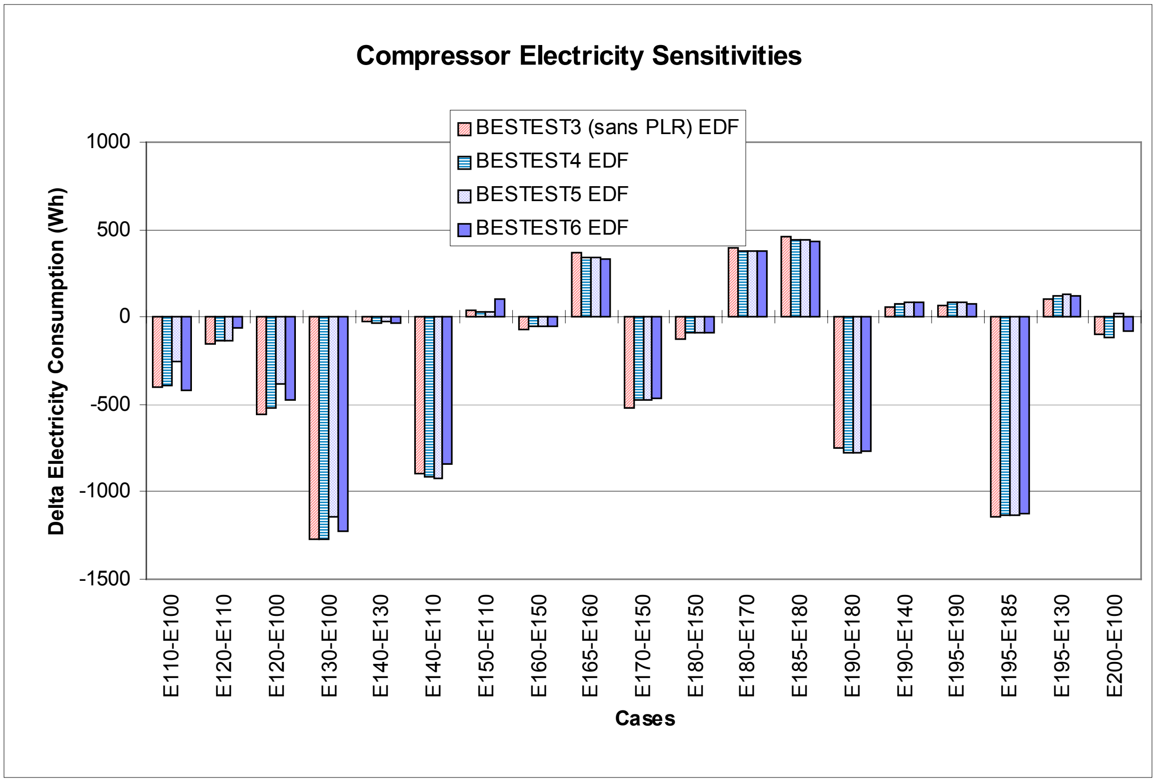

Figure 5: compressor electricity sensitivities

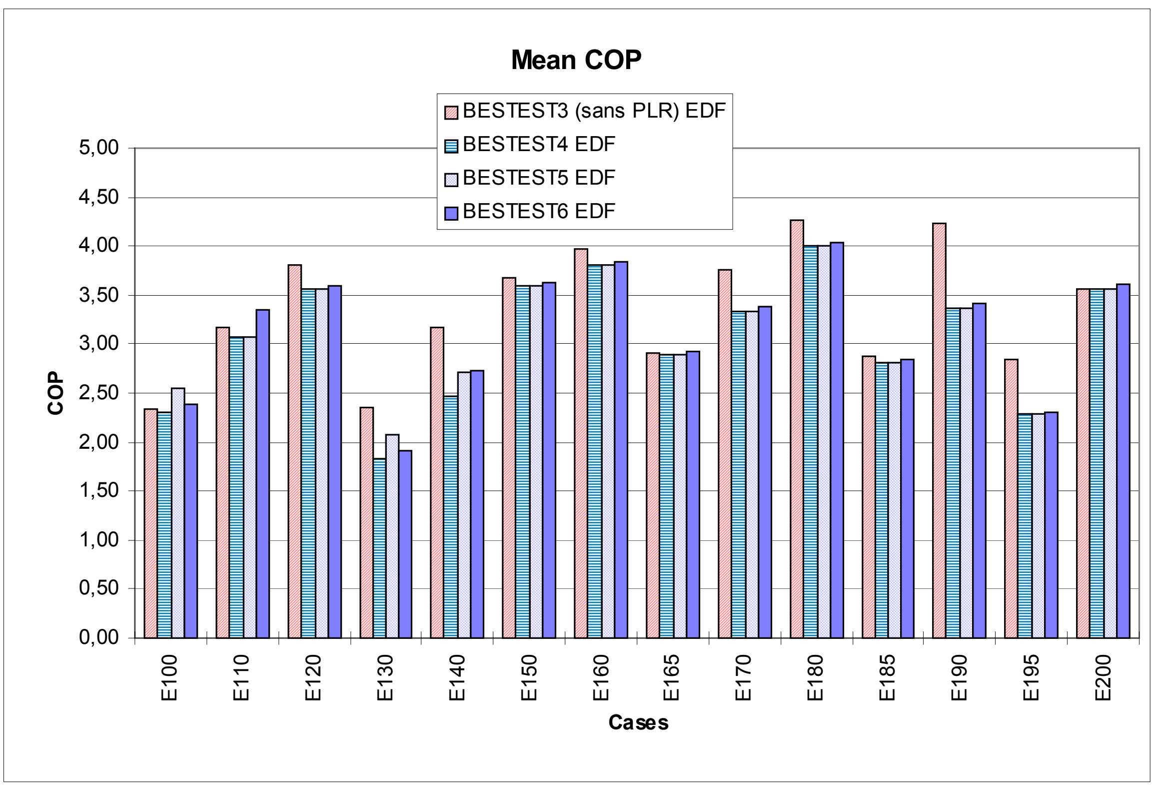

Figure 6: mean COP

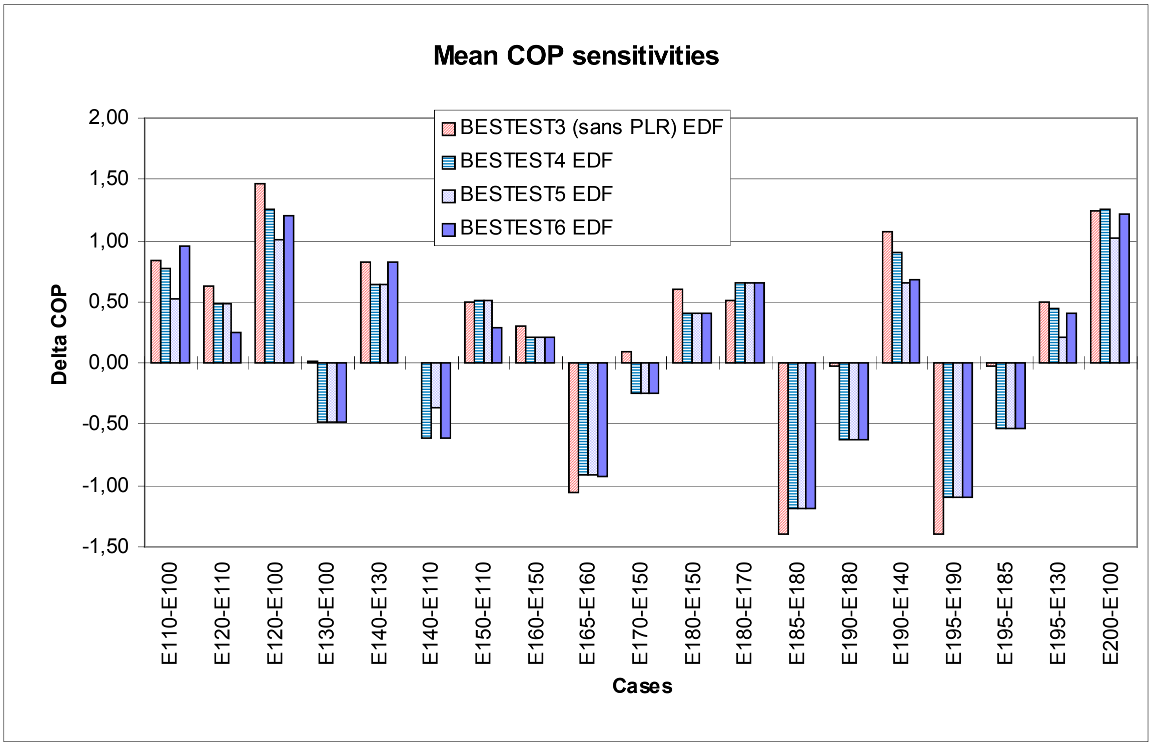

Figure 7: mean COP sensitivities

4 Conclusion

For EDF, IEA Task22 is a good opportunity to compare CLIM2000 results with others building energy analysis tools available, as done in the past in the framework of IEA Annex 21.

For the first round of simulations, we used old elementary model of mechanical system available into the CLIM2000 library. As we explain in this paper, we knew when beginning the modeling that the results will be poor. Indeed, we obtained very bad results: it was not a surprise. But we did not want to miss “the train” of HVAC BESTEST.

For the second round of simulations, the second modeling used new elementary model of mechanical system developed recently to be in accordance with the new commercial policy of EDF. This modeling gave us good results, comparable to the others participants, except for mean COP sensitivity. The summary of 2^{nd} set of results **[10]** pointed out the better behaviour of CLIM2000 except for COP sensitivity to PLR. Then, the elementary model used to describe the mechanical equipment was modified to take into account this COP Degradation factor.

For the third round of simulations, as a consequence of the results obtained in second round, we implemented the COP degradation factor into our elementary model. We also calculated the effect of extrapolation into performance maps. We found that the insensitivity to PLR disappeared totally as expected. We also found that some test cases were impacted by the extrapolation of performance maps but not too much because EWB was the only variable going out of the map and not too far.

For the fourth round of simulations, we extrapolated the performance map (using automated extrapolation of EWB and manual extrapolation of EDB) and we added a new performance map to indicate the limits of performance of the split system in dry coil conditions. These modifications permitted to reach agreement with the analytical solution results, except for the E200, which disagrees with the analytical solution results by about 1% for consumption, sensible coil load and sensible zone load.

At the end, we can say that our participation to the HVAC BESTEST was very useful to improve our elementary model. The HVAC BESTEST procedure is useful to diagnostic a software program.

5 References

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[5] FEBURIE J., " Dossier du modèle, Pompe à chaleur air/air réversible avec fichiers de performances ", EDF report HE-14/99/012, to be published.

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[7] NEYMARK J. – JUDKOFF R., " International Energy Agency, Building Energy Simulation Tests and Diagnostic Method for Mechanical Equipment (HVAC BESTEST) ", September 1997, Restricted.

[8] NEYMARK J. – JUDKOFF R., " International Energy Agency, Building Energy Simulation Tests and Diagnostic Method for Mechanical Equipment (HVAC BESTEST) ", September 1998, Restricted.

[9] NEYMARK J. – JUDKOFF R., " HVAC Bestest, Summary of 1st set of results ", IEA Task22 intermediate report, April 1998, Restricted.

[10] NEYMARK J. – JUDKOFF R., " HVAC Bestest, Summary of 2nd set of results ", IEA Task22 intermediate report, March 1999, Restricted.

Program name (please include version number)

CLIM2000

Your name, organisation, and country

EDF, France

Program status

	Public domain
a	Commercial
x	Research
	Other (please specify)

Solution method for unitary space cooling equipment

	Overall Performance Maps
x	Individual Component Models
	Constant Performance (no possible variation with entering or ambient conditions)
	Other (please specify)

Interaction between loads and systems calculations

x	Both are calculated during the same timestep
	First, loads are calculated for the entire simulation period, then equipment performance is calculated separately
	Other (please specify)

Time step

	Fixed within code (please specify time step)
	User-specified (please specify time step)
x	Other (please specify) *adapted*

Timing convention for meteorological data: sampling interval

	Fixed within code (please specify interval)
x	User-specified

Timing convention for meteorological data: period covered by first record

	Fixed within code (please specify period or time which meteorological record covers)
x	User-specified

Meteorological data reconstitution scheme

a	Climate assumed stepwise constant over sampling interval
x	Linear interpolation used over climate sampling interval
	Other (please specify)

Output timing conventions

a	Produces spot predictions at the end of each time step
x	Produces spot output at end of each hour
	Produces average outputs for each hour (please specify period to which value relates)

Treatment of zone air

x	Single temperature (i.e. good mixing assumed)
	Stratified model
	Simplified distribution model
	Full CFD model
	Other (please specify)

Zone air initial conditions

	Same as outside air
x	Other (please specify)

Internal gains output characteristics

x	Purely convective
a	Radiative/Convective split fixed within code
	Radiative/Convective split specified by user
	Detailed modeling of source output

Mechanical systems output characteristics

x	Purely convective
a	Radiative/Convective split fixed within code
	Radiative/Convective split specified by user
	Detailed modeling of source output

Control temperature

a	Air temperature
x	Combination of air and radiant temperatures fixed within the code
	User-specified combination of air and radiant temperatures
	User-specified construction surface temperatures
	User-specified temperatures within construction
	Other (please specify)

Control properties

	Ideal control as specified in the user's manual
	On/Off thermostat control
	On/Off thermostat control with hysteresis
	On/Off thermostat control with minimum equipment on and/or off durations
x	Proportional control
	More comprehensive controls (please specify)

Performance Map: characteristics

	Default curves
x	Custom curve fitting
	Detailed mapping not available
	Other (please specify)

Performance Map: independent variables

x	Entering Drybulb Temperature
x	Entering Wetbulb Temperature
x	Outdoor Drybulb Temperature
x	Part Load Ratio
x	Indoor Fan Air Flow Rate
	Other (please specify)

Performance Map: dependent variables

x	Coefficient of Performance (or other ratio of load to electricity consumption)
x	Total Capacity
x	Sensible Capacity
	Bypass Factor
	Other (please specify)

Performance Map: available curve fit techniques

x	Linear, f(one independent variable)
	Quadratic, f(one independent variable)
a	Cubic, f(one independent variable)
	Bi-Linear, f(two independent variables)
	Bi-Quadratic, f(two independent variables)
	Other (please specify)

Performance Map: extrapolation limits

	Limits independent variables
	Limits dependent variables
x	No extrapolation limits
	Extrapolation not allowed
	Other (please specify)

Cooling coil and supply air conditions model

x	Supply air temperature = apparatus dew point (ADP); supply air humidity ratio = humidity ratio of saturated air at ADP
	Bypass factor model using listed ADP data
	Bypass factor model with ADP calculated from extending condition line
x	Fan heat included
	More comprehensive model (please specify)

Disaggregation of fans' electricity use directly in the simulation and output

	Indoor fan only
	Outdoor fan only
x	Both indoor and outdoor fans disaggregated in the output
	None - disaggregation of fan outputs with separate calculations by the user

Economizer settings available (for E400 series)

	Temperature
	Enthalpy
	Compressor Lockout
	Other (please specify)

Appendix III-D

Modeler's Report for HVAC BESTEST Simulations Run on CA-SIS V1

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Octobre 2000

1 Introduction

The studies were carried out with version 1 of CA-SIS software.

The CA-SIS software environment was developed by the Electricity Application in Residential and Commercial Buildings Branch in the Research and Development Division of the French utility company EDF (Electricité de France). EDF develops the program code for the HVAC systems.

CA-SIS simulation program was developed for engineering offices studies. Its main objective is to forecast the consumption and the operational costs in order to choose and optimize the appropriate HVAC equipment. The CA-SIS calculation engine is the TRNSYS solver, property of the University of Wisconsin's Solar Energy Laboratory (USA), marketed in France by CSTB. The software calculates in a dynamic regimen. The time step is one hour.

A precise building description is given by the use of a graphical interface. A complete catalogue of HVAC system models is available (CA-SIS elementary models are based on "technology"). In addition, the software package has a library of solution types including building types.

2 Modeling

Three models are presented in this study:

- The first is made without taking into account the COP degradation factor (CDF) and without heat contribution fan. Also, there was no extrapolation of the performance maps.
- The second uses the same model, modified following the analysis by the NREL team. The model was change to take into account the COP degradation factor and the fan heat was considered. The performance tables are extrapolated.
- The third is a new model which takes into account the limits of equipment performance for a dry coil condition and a better extrapolation of performance map. We include also the CDF on the OD and ID fans.

2.1 Common Model

We have presented here the part common to the two simulations. The differences occur only at the system level (split system).

2.1.1 Building Envelope

The description of the building is simple. The building is rectangular as shown in figure 1. Its surface area is 48 m^2 and its volume 129.6 m^3.

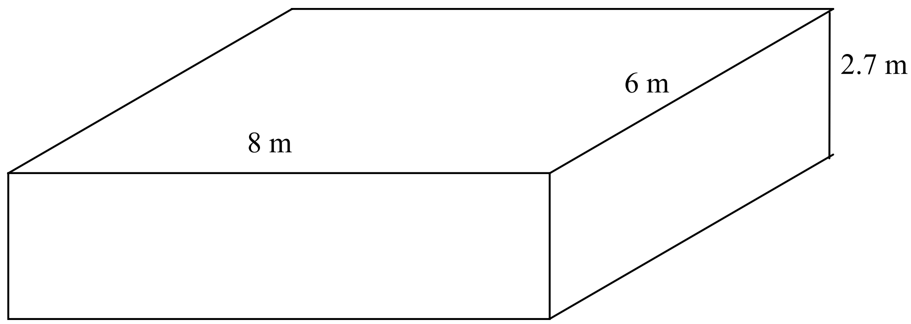

Figure 1 : Building Geometry

Outside climatic data are constant with the exception of outside dry temperature. Outside temperature is the only factor that varies.

A single thermal zone is modelled, with wall properties identical ($h_{int} = 8.29$ [$W.m^{-2}.K^{-1}$], $\lambda = 0.01$ [$W.m^{-1}.K^{-1}$], e = 1 [m], $h_{ext} = 29.3$ [$W.m^{-2}.K^{-1}$], $S_{wall} = 75$ [m²], $S_{ceiling} = 48$ [m²], $S_{floor} = 48$ [m²]).

2.1.2 Air Volume

The volume of air is represented by a dry temperature and a specific humidity.

There is no infiltration and no renewal of air.

2.1.3 Internal heat gains

Internal heat gains are described in the building. Sensible internal heat gains are purely convective and are given in the form of a sensible contribution (en [W]). Latent internal heat is given in the form of a latent contribution (en [W]).

2.2 Split System

The split system was a unit developed at EDF. So we modified the code in order to improve it capacity, so we did two models.

The inputs required by the model are the external temperature and humidity, the internal conditions and the zone load. And with this point, the evaporator data was read in the performance map.

We calculate intermittently part R which take the real part of working of split on one hour into account

$$R = \mathrm{Min}\,(\text{ load envelope}, P_{cold}) / P_{cold}$$

P_{cold} is the evaporator power available for one hour of working.

So we applied R on consumption.

The evaporator coil latent load is transmitted by a mass flow of water vapor. The conversion is :

$m_v = 3600*Q_{latent}/H_{vapor}$

m_v : mass flow of water vapor (kg/h)

Q_{latent} : latent capacity (kW)

H_{vapor} : water heat of vaporization (kJ/kg)

There are three models of split system:

-the first is simple

-the second was modified in order to obtain better result and to take all that was specified in user's manual into account (CDF, fan heat).

-the third is like the second in which the limits of equipment performance for a dry coil condition were added (with performance map). Also the performance map was modified in order to ameliorate the extrapolation.

3 Modeling Options

In this paragraph, the different models of split system are presented.

3.1 .1 The First Model

- The model is based on a performance map.

Extrapolation of the table is not possible and was not done beforehand. The table given is that of TASK22. Thus, performance parameters are given for the points considered insofar as we remain within the table given in TASK 22 (linear interpolation within the table limits) but once we are at a point outside of the table, the values given are the limit values of the table.

- Furthermore, the heat contribution of the fan was not taken into account.
- As previously indicated, the first model does not take into account the COP degradation factor.
- A relaxation on humidity is carried out in order to obtain convergence; see subsection 4.1 for more discussion about this.

3.1.2 The Second Model

This model was modified in taking into account the comment of IEA (Joel Neymark).

The model is identical to the preceding incorporating in addition extrapolation of the performance table, the COP degradation factor and heat contribution of fan.

- Performance map extrapolation was carried out by extending it in the following way (linear extrapolation):

  - we added the data calculated for EWB = 10 °C,

  - we replaced the data for EWB = 21.7 °C by the data calculated for EWB = 25 °C

  - we replaced the data for EDB = 26.7 °C by the data calculated for EDB = 30 °C

We did bigger extrapolations in order to be completed.

- The COP degradation factor was integrated to calculate the power emitted by the fans as well as the consumption.

In the second model the CDF was applied to compressor and indoor fan. (We change the code of unit split system.)

- In addition, we added the heat contribution of the fan at the level of the load to be supplied by the evaporator. (The code of unit split system was not changed. The heat contribution change only with the results.)
- Also a relaxation on humidity, temperature and envelope load is carried out in order to obtain convergence for case E200. See subsection 4.1 for more discussion of this.

3.1.3 The Third Model

This model was approximately the same as the second model. This model was modified in taking into account the comments of IEA (Joel Neymark).

- We add the limits of equipment performance in another performance map. We changed the performance map. The extrapolation was recalculated to extend the original map by linear extrapolation in the following way:
    - we added the data calculated for EWB = 13 °C
    - we replaced the data for EWB = 21.7 by the data calculated for EWB = 25 °C
    - we added the data for EDB = 27.8 °C
- the relaxation on entrance data of split (zone humidity ratio and zone temperature) is carried to accelerate the convergence for all case. (See subsection 4.1 for more discussion of this).
- The CDF adjustment was applied on OD fan and ID fan. In the others models we didn't apply it.

4 Modelling difficulties

4.1 Relaxation on data

In order to obtain the convergence, we had to do relaxation on data. The relaxation is mathematical operation in order to accelerate the convergence of a calculation. The relaxation is to take a fraction of data at previous iteration and the rest of the data at the current iteration, in the same time step:

$$X_n (t) = a * X_{n-1}(t) + (1-a) * X_n (t) .$$

It allows a stable part and another which changes with the iteration. This method improves the convergence of the data.

In the first split system model we did relaxation with a = 0.5 on zone humidity ratio.

In the second model a relaxation on zone humidity ratio, temperature and envelope load is carried out in order to obtain convergence for case E200 (relaxation is done with a coefficient a of 0.7). For the others cases we did relaxation on zone humidity ratio with a = 0.5.

In the third model a relaxation on zone humidity ratio and temperature is carried out in order to obtain convergence for all cases (relaxation for case E200 is done with a coefficient a of 0.7, for the others cases we did relaxation with a = 0.5). The other cases were repeated with a = 0.7 and gave the same results. As a result of that, the CASIS default coefficient has been changed to a = 0.7.

5 Software errors discovered and/or comparison between different versions of same software

- With the first model, we detected that:

  - we had to do relaxation on humidity in order to obtain convergence when the building have important latent load. The internal condition change considerably.

  - the extrapolation of performance map was important

  - the fan heat and CDF was forgotten

The case E200 doesn't converge because the internal condition change considerably

- With the second model, we detected that:

  - we had to do relaxation on entrance data of split system in order to obtain convergence for test case E200. The internal load is important and this case allows many iterations to converge.

For do relaxation we have change the code of UNIT split system that we have developed.

- With the third model, we detected that:

  - we had to do relaxation on entrance data (zone humidity ratio and zone temperature) of split system in order to obtain convergence for Case E200.

  - the previous model did not take into account correctly the limits of equipment performance in dry coil conditions and so that the extrapolation of performance map was not always correct.

  - fans' consumption results improved when CDF was applied to the indoor and outdoor fans.

6 Results

The results are presented for each model.

6.1 First model

These first results were given in February 2000.

cases	February Totals										February Mean			February Maximum			February Minimum		
	Cooling energy Consumption				Evaporator coil Load			Envelope load			Envelope load			Envelope load			Envelope load		
	Total	comp.	supply	cond.	total	sensible	latent	total	sensible	latent	COP	IDB	hum.	COP	IDB	hum.	COP	IDB	hum.
	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh		°C	kg/kg		°C	kg/kg		°C	kg/kg
E100	1478	1263	146	69	3656	3656	0	3656	3656	0	2.47	22.2	0.0070	2.47	22.2	0.0070	2.47	22.2	0.0070
E110	1089	907	123	58	3635	3635	0	3635	3635	0	3.34	22.2	0.0067	3.34	22.2	0.0067	3.34	22.2	0.0067
E120	1042	874	114	53	3629	3629	0	3629	3629	0	3.48	26.7	0.0070	3.48	26.7	0.0070	3.48	26.7	0.0070
E130	108	95	8	4	208	208	0	208	208	0	1.94	22.2	0.0070	1.94	22.2	0.0070	1.94	22.2	0.0070
E140	74	64	7	3	194.7	188	6.7	188	188	0	2.55	22.2	0.0070	2.55	22.2	0.0070	2.55	22.2	0.0070
E150	1220	1015	139	65	4373	3636	737	4375	3636	739	3.59	22.2	0.0083	3.60	22.2	0.0084	3.58	22.2	0.0083
E160	1152	969	125	59	4368	3629	739	4368	3629	739	3.79	26.7	0.0101	3.80	26.7	0.0102	3.78	26.7	0.0101
E165	1520	1299	150	71	4389	3649	740	4388	3649	739	2.89	23.3	0.0093	2.89	23.3	0.0094	2.88	23.3	0.0092
E170	645	551	64	30	2157	1418	739	2157	1418	739	3.34	22.2	0.0105	3.34	22.2	0.0105	3.34	22.2	0.0105
E180	1099	933	113	53	4392	1418	2974	4375	1418	2957	3.98	22.2	0.0162	4.02	22.2	0.0162	3.95	22.2	0.0162
E185	1564	1362	138	65	4397	1438	2959	4395	1438	2957	2.81	22.2	0.0160	2.83	22.2	0.0162	2.78	22.2	0.0159
E190	168	147	14	7	558	188	370	555	188	367	3.32	22.2	0.0159	3.32	22.2	0.0159	3.32	22.2	0.0159
E195	255	228	18	9	578	208	370	575	208	367	2.26	22.2	0.0154	2.26	22.2	0.0154	2.26	22.2	0.0154
E200																			

For this model, the E200 case does not converge. There is an extrapolation problem as well as a problem with the interior conditions which vary considerably at each iteration.

The results obtained are not correct as the COP degradation factor is absent and the performance map has not been extrapolated, but also the fan contribution at the level of the evaporator load is lacking.

6.2 Second simulation

These results come from modifications made following the comments on the first results.

cases	February Totals										February Mean			February Maximum			February Minimum		
	Cooling energy Consumption				Evaporator coil Load			Envelope load											
	Total	comp.	supply	cond.	total	sensible	latent	total	sensible	latent	COP	IDB	hum.	COP	IDB	hum.	COP	IDB	hum.
	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh		°C	kg/kg		°C	kg/kg		°C	kg/kg
E100	1569	1354	146	69	3802	3802	0	3656	3656	0	2.33	22.2	0.0070	2.33	22.2	0.0070	2.33	22.2	0.0070
E110	1090	908	124	58	3761	3761	0	3637	3637	0	3.34	22.2	0.0066	3.34	22.2	0.0066	3.34	22.2	0.0066
E120	1042	874	114	53	3746	3746	0	3632	3632	0	3.48	26.7	0.0070	3.48	26.7	0.0070	3.48	26.7	0.0070
E130	113	101	8	4	217	217	0	209	209	0	1.85	22.2	0.0070	1.85	22.2	0.0070	1.85	22.2	0.0070
E140	69	60	6	3	196	196	0	190	190	0	2.73	22.2	0.0066	2.73	22.2	0.0066	2.73	22.2	0.0066
E150	1222	1017	140	65	4519	3777	742	4376	3637	739	3.58	22.2	0.0083	3.59	22.2	0.0084	3.57	22.2	0.0083
E160	1152	969	125	59	4496	3757	739	4371	3632	739	3.79	26.7	0.0101	3.80	26.7	0.0102	3.78	26.7	0.0101
E165	1520	1299	150	71	4539	3799	740	4388	3649	739	2.89	23.3	0.0093	2.89	23.3	0.0093	2.88	23.3	0.0093
E170	646	552	64	30	2224	1484	740	2158	1419	739	3.34	22.2	0.0105	3.34	22.2	0.0105	3.34	22.2	0.0105
E180	1095	929	113	53	4490	1532	2958	4377	1420	2957	4.00	22.2	0.0162	4.03	22.2	0.0164	3.97	22.2	0.0160
E185	1613	1410	138	65	4533	1576	2957	4396	1439	2957	2.73	22.2	0.0161	2.75	22.2	0.0162	2.70	22.2	0.0159
E190	166	145	15	7	574	204	370	557	190	367	3.35	22.2	0.0159	3.35	22.2	0.0159	3.35	22.2	0.0159
E195	260	233	18	9	597	227	370	576	209	367	2.21	22.2	0.0154	2.21	22.2	0.0154	2.21	22.2	0.0154
E200	1483	1256	155	73	5686	4270	1416	5341	4120	1221	3.60	27.0	0.0116	3.61	27.0	0.0116	3.59	27.0	0.0115

The E200 case converges with this type of model but the results are not very satisfactory. It is by increasing the number of iterations and doing relaxation on humidity, temperature and envelope loads that CA-SIS converges.

The number of iterations is a parameter that each user can change but to do relaxation we have change the code of unit split system

The other results are better than they were with preceding simulations. They are comparable to those obtained by other codes.

6.3 Third simulation

These results come from modifications made following the comments on the first and second results.

The results obtained are better. They are comparable to analytical result.

We can see that the extrapolation of performance map and limits of performance are very important, they have a bigger influence on result.

	February Totals										February Mean			February Maximum			February Minimum		
	Cooling energy Consumption				Evaporator coil Load			Envelope load			Envelope load			Envelope load			Envelope load		
cases	Total	comp.	supply	cond.	total	sensible	latent	total	sensible	latent	COP	IDB	hum.	COP	IDB	hum.	COP	IDB	hum.
	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh		°C	kg/kg		°C	kg/kg		°C	kg/kg
E100	1531	1319	144	68	3800	3800	0	3656	3656	0	2,39	22,2	0,0075	2,39	22,2	0,0075	2,39	22,2	0,0075
E110	1077	889	128	60	3765	3765	0	3637	3637	0	3,38	22,2	0,0066	3,38	22,2	0,0066	3,38	22,2	0,0066
E120	1012	840	117	55	3749	3749	0	3632	3632	0	3,59	26,7	0,0080	3,59	26,7	0,0080	3,59	26,7	0,0080
E130	110	95	10	5	219	219	0	209	209	0	1,91	22,2	0,0075	1,91	22,2	0,0075	1,91	22,2	0,0075
E140	68	57	8	4	198	198	0	190	190	0	2,77	22,2	0,0065	2,77	22,2	0,0065	2,77	22,2	0,0065
E150	1208	1000	141	66	4517	3778	739	4376	3637	739	3,62	22,2	0,0083	3,63	22,2	0,0084	3,62	22,2	0,0083
E160	1140	950	129	61	4501	3761	740	4371	3632	739	3,84	26,7	0,0102	3,84	26,7	0,0103	3,83	26,7	0,0101
E165	1502	1283	149	70	4538	3798	740	4388	3649	739	2,92	23,3	0,0093	2,94	23,3	0,0094	2,91	23,3	0,0093
E170	638	531	73	34	2233	1493	740	2159	1420	739	3,38	22,2	0,0106	3,38	22,2	0,1060	3,38	22,2	0,1060
E180	1083	909	118	56	4495	1537	2958	4376	1420	2957	4,04	22,2	0,0164	4,05	22,2	0,0165	4,03	22,2	0,0162
E185	1544	1340	139	65	4507	1548	2959	4396	1439	2957	2,85	22,2	0,0162	2,86	22,2	0,0163	2,84	22,2	0,0161
E190	164	138	18	8	578	208	370	557	190	367	3,41	22,2	0,0160	3,41	22,2	0,0160	3,41	22,2	0,0160
E195	250	217	23	11	602	232	370	576	209	367	2,31	22,2	0,0156	2,31	22,2	0,0156	2,31	22,2	0,0156
E200	1477	1250	154	73	5498	4276	1222	5343	4122	1221	3,62	26,7	0,0114	3,63	26,7	0,0115	3,61	26,7	0,0113

7 Other

Nothing

8 Conclusion

For EDF, IEA Task22 is a good opportunity to compare CA-SIS result with other software and analytical solution results. For the first result, we used a model in which it had no heat contribution of fan and no CDF. Also the performance map wasn't extrapolated. So the result was not very good.

For the second model, we took into account this disagreements and the result was better except for test case E200. But we can see that it stay disagreement.

In the third model we integrated all comments about disagreements. We added the CDF on ID and OD fan, we have seen the effect of extrapolation map and the importance to take into account the limits of performance in dry coil condition. So the result are better and comparable with others participant's results and analytical results.

The HVAC BESTEST is useful to improve the program code of unit developed by EDF.

9 References

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Program name (please include version number) :

CA-SIS V**

Your name, organisation, and country

Sabine HAYEZ, EDF, France

Program status

	Public domain
	Commercial
	Research
x	Other (please specify) : internal use only

Solution method for unitary space cooling equipment

x	Overall Performance Maps
x	Individual Component Models
	Constant Performance (no possible variation with entering or ambient conditions)
	Other (please specify)

Interaction between loads and systems calculations

x	Both are calculated during the same timestep
	First, loads are calculated for the entire simulation period, then equipment performance is calculated separately
	Other (please specify)

Time step

x	Fixed within code (please specify time step) time step is 1 hour
	User-specified (please specify time step)
	Other (please specify)

Timing convention for meteorological data: sampling interval

x	Fixed within code (please specify interval): 1 hour
	User-specified

Timing convention for meteorological data: period covered by first record

x	Fixed within code (please specify period or time which meteorological record covers): 1 hour
	User-specified

Meteorological data reconstitution scheme

	Climate assumed stepwise constant over sampling interval
	Linear interpolation used over climate sampling interval
w	Other (please specify): meteorological data are done hour by hour

Output timing conventions

x	Produces spot predictions at the end of each time step
	Produces spot output at end of each hour
	Produces average outputs for each hour (please specify period to which value relates)

Treatment of zone air

x	Single temperature (i.e. good mixing assumed)
	Stratified model
	Simplified distribution model
	Full CFD model
	Other (please specify)

Zone air initial conditions

x	Same as outside air
a	Other (please specify), what ever you want

Internal gains output characteristics

x	Purely convective
	Radiative/Convective split fixed within code
a	Radiative/Convective split specified by user
	Detailed modeling of source output

Mechanical systems output characteristics

x	Purely convective
	Radiative/Convective split fixed within code
a	Radiative/Convective split specified by user
	Detailed modeling of source output

Control temperature

x	Air temperature
	Combination of air and radiant temperatures fixed within the code
	User-specified combination of air and radiant temperatures
	User-specified construction surface temperatures
	User-specified temperatures within construction
	Other (please specify)

Control properties

x	Ideal control as specified in the user's manual
a	On/Off thermostat control
a	On/Off thermostat control with hysteresis
a	On/Off thermostat control with minimum equipment on and/or off durations
a	Proportional control
	More comprehensive controls (please specify)

Performance Map: characteristics

	Default curves
x	Custom curve fitting
	Detailed mapping not available
	Other (please specify)

Performance Map: independent variables

x	Entering Drybulb Temperature
x	Entering Wetbulb Temperature
x	Outdoor Drybulb Temperature
	Part Load Ratio
	Indoor Fan Air Flow Rate
	Other (please specify)

Performance Map: dependent variables

x	Coefficient of Performance (or other ratio of load to electricity consumption)
x	Total Capacity
x	Sensible Capacity
	Bypass Factor
	Other (please specify)

Performance Map: available curve fit techniques

x	Linear, f(one independent variable)
	Quadratic, f(one independent variable)
	Cubic, f(one independent variable)
	Bi-Linear, f(two independent variables)
	Bi-Quadratic, f(two independent variables)
	Other (please specify)

Performance Map: extrapolation limits

	Limits independent variables
	Limits dependent variables
	No extrapolation limits
	Extrapolation not allowed
x	Other (please specify), you need calculate the news data

Cooling coil and supply air conditions model

	Supply air temperature = apparatus dew point (ADP); supply air humidity ratio = humidity ratio of saturated air at ADP
	Bypass factor model using listed ADP data
	Bypass factor model with ADP calculated from extending condition line
x	Fan heat included
	More comprehensive model (please specify)

Disaggregation of fans' electricity use directly in the simulation and output

	Indoor fan only
	Outdoor fan only
x	Both indoor and outdoor fans disaggregated in the output
	None - disaggregation of fan outputs with separate calculations by the user

Economizer settings available (for E400 series) not realised

	Temperature
	Enthalpy
	Compressor Lockout
	Other (please specify)

Appendix III-E

DOE-2.1E ESTSC Version 088
CIEMAT- IER
SPAIN
August 2000

1. Introduction

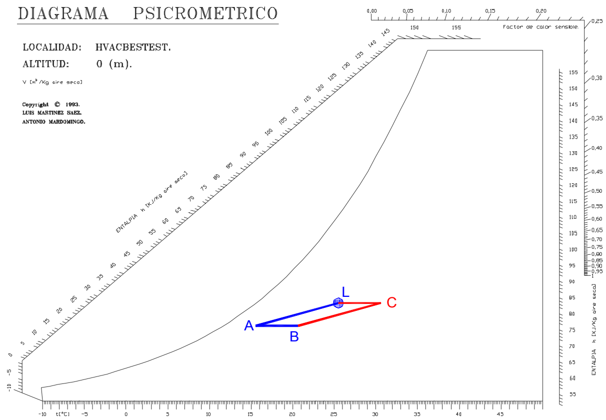
The DOE-2 program has been used by NREL to develop their model. General description of the software such as timestep or procedure of HVAC modeling can be read at NREL modeler report. However, CIEMAT used an ESTSC version of DOE-2, which is not exactly the same as the J.J. Hirsch version used by NREL.

2. Modeling Assumptions

We had few difficulties to develop our simulations and some assumptions have been done, as follows:

1. The Infrared Emittances of the opaque surfaces have not been considered.
2. INSIDE-FILM-RESISTANCE. We have considered an inside surface coefficient of 0.1206 m^2K/W for vertical surfaces, 0.108 m^2K/W for the floor and 0.1631 m^2K/W for the roof which is recommended by the program developer.
3. Internal heat gains have been simulated as all convective gains caused by equipment.
4. THERMOSTAT THROTTLING-RANGE =0.056º C which is the minimum setting in DOE-2. Exact ideal on/off control is not possible.
5. The unitary split system has been simulated using the Packaged Terminal Air Conditioned System (PTAC) model. The DOE-2 user's manual description of this model is: "*PTAC are designed primarily for commercial installations to provide the total heating and cooling function for a room or zone, and are specifically designed for through–the-wall installation. The units (which are hybrid systems/plants) are mostly used in hotel and motel guest rooms, apartments, hospitals, nursing homes, and office buildings. All PTAC units discharge air directly into the space without ductwork*".
6. No data about minimum and maximum supply temperature for the equipment were supplied. A minimum supply air temperature of 6ºC has been assumed. [*Editor's note: Based on this comment by CIEMAT, guidance regarding this input was included in the final version of Part I.*]
7. The geometry of the evaporator coil has not been considered.
8. The PTAC system is defined in DOE-2 as a blow-through system. It does not have the ability to be a DRAW-THROUGH system. The fan position affects the temperature of the air entering the cooling coil and this affects the performance maps of the equipment. The curves that define the performance map have been constructed considering this effect. The sensible capacity and specially the coil bypass factor have to be calculated considering that the fan heats the air before it passes through the

cooling coil. The final temperature (after the cooling coil and the fan) has to be the same for the real fan position and the modeled one. Next figure will explain the effect considered.



The real process in the equipment is the signed as L-A-B. The process considered in our model is L-C-B. Changes in the input data have been made for a proper consideration of this difference.

9. The sensible capacity, bypass factor and the curves fit for those parameters have been calculated considering the method used by the tool and which is documented in its manuals (2.1.A Reference manual, pIV.238).

$$SCAP_T = (COOL - SH - CAP)(COOL - SH - FT(EWB, ODB)) - [1.08 * CFM * (1 - BF)(80 - EDB)]$$

Where: $SCAP_T$; sensible capacity

COOL-SH-CAP; sensible at ARI conditions.

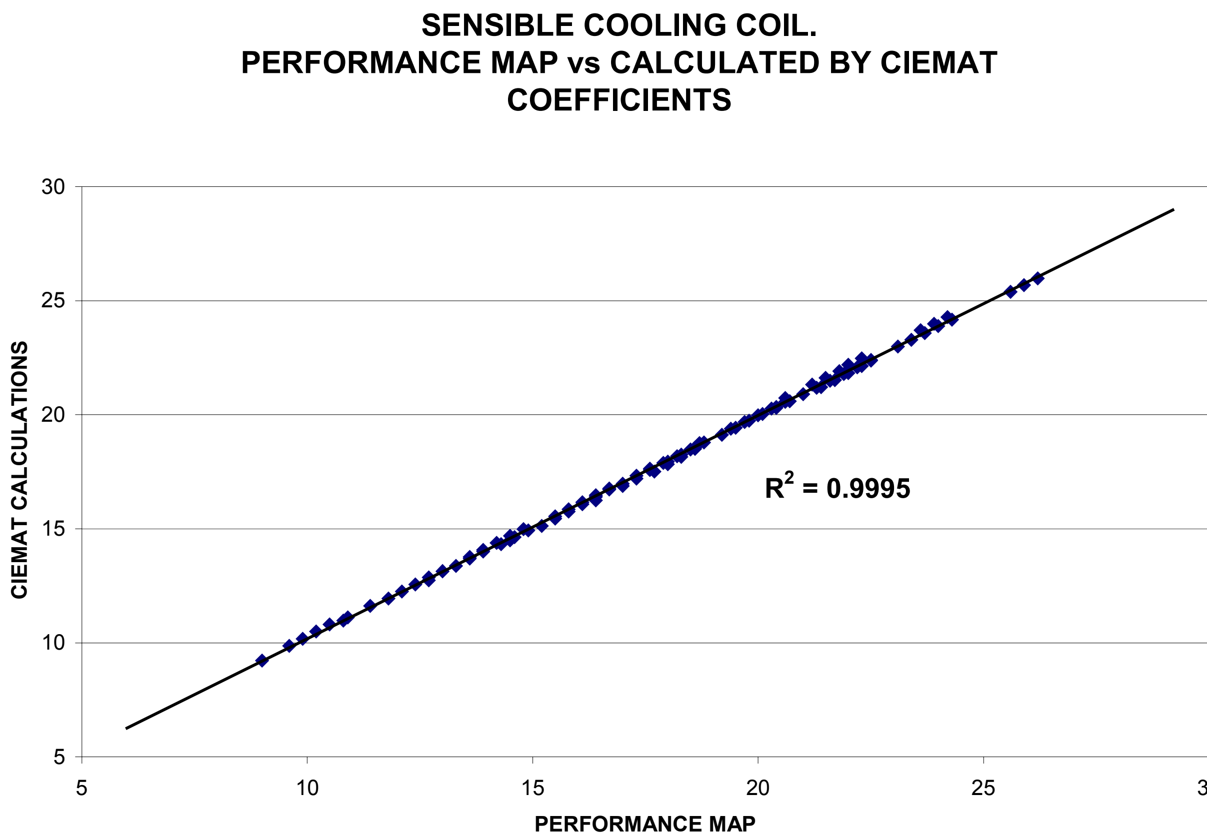
COOL-SH-FT(EWB,ODB); Curve fit as function of entering wet bulb and outside dry bulb temperatures.

CFM; Supply airflow

BF; Cooling coil bypass factor. It is a function of EWB and ODB.

$$BF = BF_{ARI} * BF\text{-}FT(EWB, ODB)$$

All the points supplied at the user manual have been considered and the parameters of this equation (curves and nominal values at ARI conditions) have been calculated according a multi-lineal regression. This caused that the values at ARI conditions might be lightly different to those provided at the user's manual. This would cause an error on the calculations at ARI conditions but optimizes the behavior in other working points of the performance map, as the next graphic shows.



3. Modeling Options

SYSTEM FANS: SUPPLY STATIC & SUPPLY EFFICIENCY

The program includes two possibilities for determining indoor distribution fan power heat (see NREL modeler report). Both methods should obtain similar results and the sensitivity tests made confirmed that results showed disagreements <0.01%.

4. Modeling Difficulties

Few outputs of the report have been calculated, because the program was not able to provide directly those results.

- COOLING ENERGY CONSUMPTION

  a) CONDENSER FAN. The program calculates together the compressor and condenser fan consumption. As this and interior fan are single speed fans and both of them cycle on and off together with compressor we have calculated the condenser fan consumption using the equation:

  COND.FAN.CONS=(108 W)x (672 hours) x (cold air flow / maximum cold air flow)

  b) SUPPLY FAN. Calculated by the program.

  c) COMPRESSOR. Cooling electrical consumption (does not include the supply fan) minus condenser fan consumption

- EVAPORATOR COIL LOAD

  a) TOTAL. Calculated by the program

  b) LATENT. Calculated by the program

  c) SENSIBLE. =Total - latent.

- ENVELOPE LOAD

  a) SENSIBLE. Calculated by the program

  b) LATENT. Calculated by the program

  c) TOTAL. = Sensible + latent.

- COEFFICIENT OF PERFORMANCE

  Calculated using the following equations:

  COP=(Net refrigerating effect) / (total energy input)

  Net Refrigerating Effect = (Total Evaporator Coil Load) - (Supply Fan Energy Cons.)

- INTERIOR DRY BULB AND HUMIDITY RATIO

  The program calculates both of them.

5. Software Errors Discovered and/or Comparison Between Different Versions of the Same Software

A few software errors have been detected. Those are:

DOCUMENTATION PROBLEMS

1. The DOE-2 has the ability to calculate the equipment operating characteristic curves using different working points. The data for those curves fit have to be provided to the program in American units, even if the Metric option of the program is being used. The program considers the data provided as English units, so if they are given in metric, they will be misinterpreted.
2. The DOE-2 user's manual (2.1-A IV.248) defines the coil bypass factor as a function of entering wet bulb and entering dry bulb. There is an error, because the program uses this curve as a function of entering wet bulb and outside dry bulb temperature.

TO TRANSMIT TO CODE AUTHORS

1. In some cases a fan heat inconsistency have been detected: sensible cooling coil load minus sensible room load should be equal to fan consumption and it is not. Those differences are variable for each case. This could be caused by a fan heat modeling error, but could be other kind of error. Next table shows those disagreements:

   sens.load error=[(sens.zone load+fan energy)–(sens.coil load)]/(sens.zone load + fan energy)*100

CASE	SENS COIL LOAD	SENS ROOM LOAD + SUPPLY FAN ENERGY	ERROR
E100	3841.47	3799.605	1.10%
E110	3803.58	3768.518	0.93%
E120	3763.48	3740.053	0.63%
E130	215.778	215.66	0.05%
E140	195.53	195.45	0.04%
E150	3803.58	3768.518	0.93%
E160	3777.178	3749.504	0.74%
E165	3828.258	3788.549	1.05%
E170	1486.857	1479.473	0.50%
E180	1553.184	1528.865	1.59%
E185	1608.094	1571.905	2.30%
E190	203.007	202.641	0.18%
E195	225.64	225.054	0.26%
E200	4313.176	4269.653	1.02%
MEAN ERROR			**0.81%**

The supply fan electricity consumption depends only of the supply airflow, which is very sensible to the supply air temperature. The differences between NREL's version and CIEMAT's could be caused by *"fan heat inconsistency with fan power"*. Besides this, there are two points that should be considered.

- The PTAC system has always a blow-through fan disposition (DOE-2 limitation), so we modified the BF and the EDB and some other Input data to obtain the same supply air temperature that in a draw through disposition. This should not affect the results (the fan consumption, and all other values look quite good), but could explain those discrepancies. A good way to check this could be to compare the supply air temperature, to explain these discrepancies.
- The performance map for CIEMAT model has been created by using the equation that mentioned by NREL in its model report introduction (2.1.A reference manual, p.IV.238). This would explain small differences between CIEMAT's sensible capacitance and others and also a different supply air temperature.

Fan Heat Comparison between DOE/CIEMAT and DOE/NREL showed *fan heat inconsistency with fan power* even when different methods to input the fan data have been used intentionally. Possible program bug. Errors around 1-2% to total coil load.

1. NREL: KW/cfm and delta-t
2. CIEMAT: static pressure and efficiency.

The fan heat is sensible coil load minus zone load. This is true theoretically, but DOE-2/CIEMAT showed an inconsistency of 1-2% as showed previous table and next equation:

SENSIBLE-COIL-LOAD=SENSIBLE-ZONE-LOAD+FAN-HEAT+ERROR ($\pm$1-2%)

The DOE-2 makes some corrections to the loads calculations at the SYSTEM subprogram. If the FAN HEAT graph is observed, both DOE-2 models are showing the largest disagreements in almost all cases. If it is assumed that this error is only caused by the fan-heat there would be an error of 37%

in the fan heat. But we tend to believe that there is some kind of inconsistency between the sensible load calculated at the LOADS subprogram and at the SYSTEM subprogram.

This would explain NREL errors at E180-E185 and elsewhere, and CIEMAT's in all the cases. If this error is not considered as part of the fan simulation, but as part of the global sensible coil load, it will be negligible. Round assumptions or differences between the temperature considered at the ZONE subprogram and the SYSTEM one could cause those 1-2% disagreements on the sensible coil loads.

2. Some discrepancies on the COP value between NREL results and CIEMAT's. In those cases where CIEMAT's coil loads are higher than zone load + fan heat, a higher COP value is calculated. As COP is given by the relation between the coil load and the energy consumed, this overestimation of the COP could be caused by an unreal overcalculation of the cooling loads

   NREL COP for E180 might be lower because its sensible coil load is higher that CIEMAT's calculations. DOE-2.1E/NREL's latent coil load was greater than latent zone load at certain times (see latent coil load vs envelope load in Part IV results tables). This higher coil load may be related to the "fan heat" discrepancy (see fan heat graph in Part IV). It could be once again a "*fan heat inconsistency with fan power".* This could be a program bug.

<u>EXPLANATION OF SOME INPUT DECK CHANGES.</u>

Some changes have been made to the input deck along the HVAC BESTEST project to solve disagreements or errors on the definition. Those changes are:

1. **Simulating January the same as February.** In previous rounds January had infiltration, no internal gains and the equipment was off. This caused that the initial conditions were roughly different that for the other models.

2. **Curve fit data was revised.** The inputs and curve fit parameters were re-defined as it has been explained in this modeler report. The new curve fit data agree with the draw through disposition of the fan and the equation used by DOE-2 to calculate the sensible capacity at each time (see previous items on this modeler report).

3. **Minimum allowable supply air temperature was reduced to 6ºC.** This temperature was not defined in the user's manual. In previous round it has been considered as 10ºC. Considering the possibility that this temperature could be too high and could be causing inaccuracy in some cases.

4. **Heating thermostat was set to 20.2ºC.** In previous cases it had the same setpoint as cooling. As DOE-2 does not allow a definition of an exact ideal on/off control is not possible, some proportionality is required and so heating setpoint must be settled. If the heating setpoint were the same that cooling, no decreasing temperature floating effect would be allowed. This effect will not occur probably, but must be allowed, just in case it can happen at some time.

5. **Fan control has been changed to "cycling".** Previously it was constant which did not agree with the user's manual.

6. **Sensible Heat Capacity at ARI conditions has been reduced by 4%.** This is the result of the calculations made using the equation recommended by the software developer (see previous items). The sensible capacity of the coils has been corrected at ARI conditions but the error along the performance map is very small (R^2=0.9995).

7. Sensible heat capacity, bypass factor and curve fit have been reviewed according to equation recommended by the software developer for COOL-SH-CAP and COOL-SH-FT by using the equation that NREL mentioned in their report (2.1.A reference manual, p.IV.238 andDOE-2.1E/NREL modeler report)

6. Results

The HVAC BESTEST have been developed along the TASK 22 while the different models and tools have submitted results and asked for more detailed or new data. This explains changes or modifications on the simulation input.

The results for CIEMAT model are analyzed together with all the other models along the final report. This analysis is considered detailed enough. The main conclusions obtained are:

- ***Fan heat is up 37% high throughout.*** *This error has been explained previously in this modeler report. Our point of view is different to this conclusion, the disagreement has to be analyzed not only as a fan heat error, but as a cooling coil error (see item 5 of this report). The error would be only 1-2%, to total coil load.*
- ***Disaggregated indoor fan does not pick up the CDF adjustment.*** *This problem have been observed also for the NREL version of the program*
- ***Outdoor fan does not pick up the CDF adjustment.*** *The outdoor fan energy heat is not a DOE-2 output for this system. It has been hand-calculated by us (see item 4 of this modeler report) using the indoor fan results. This error has to be assumed and is consistent with the previous one.*

7. Other

NOTHING

8. Conclusion

For CIEMAT, the HVACBESTEST is a very good exercise for comparative validation of the software. It is very important to have a greater confidence in HVAC simulation.

Program name (please include version number)

DOE-2.1E version 088

Your name, organisation, and country

Juan Travesí. Centro de Investigaciones Energéticas Medioambientales y Tecnológicas (CIEMAT). Spain

Program status

X	Public Domain
X	Commercial
	Research
	Other (please specify)

Solution method for unitary space cooling equipment

X	Overall Performance Maps
	Individual Component Models
	Constant Performance (no possible variation with entering or ambient conditions)
	Other (please specify)

Interaction between loads and systems calculations

	Both are calculated during the same timestep
X	First, loads are calculated for the entire simulation period, then equipment performance is calculated separately. *During the system calculations the loads results are checked and some small re-calculations are made.*
	Other (please specify)

Time step

X	Fixed within code (please specify time step). *1 hour*
	User-specified (please specify time step)
	Other (please specify)

Timing convention for meteorological data: sampling interval

X	Fixed within code (please specify interval): *1 hour*
	User-specified

Timing convention for meteorological data: period covered by first record

X	Fixed within code (please specify period or time which meteorological record covers). 0.00-1.00
	User-specified

Meteorological data reconstitution scheme

X	Climate assumed stepwise constant over sampling interval
	Linear interpolation used over climate sampling interval
	Other (please specify)

Output timing conventions

	Produces spot predictions at the end of each time step
	Produces spot output at end of each hour
X	Produces average outputs for each hour (please specify period to which value relates). *Same as time step.*

Treatment of zone air

X	Single temperature (i.e. good mixing assumed)
	Stratified model
	Simplified distribution model
	Full CFD model
	Other (please specify)

Zone air initial conditions

X	Same as outside air
	Other (please specify)

Internal gains output characteristics

	Purely convective
	Radiative/Convective split fixed within code
X	Radiative/Convective split specified by user. *Also sensible/latent and scheduled.*
	Detailed modeling of source output

Mechanical systems output characteristics

X	Purely convective
	Radiative/Convective split fixed within code
	Radiative/Convective split specified by user
	Detailed modeling of source output

Control temperature

X	Air temperature
	Combination of air and radiant temperatures fixed within the code
	User-specified combination of air and radiant temperatures
	User-specified construction surface temperatures
	User-specified temperatures within construction
	Other (please specify)

Control properties

	Ideal control as specified in the user's manual
	On/Off thermostat control
	On/Off thermostat control with hysteresis
	On/Off thermostat control with minimum equipment on and/or off durations
X	Proportional control. *A throttling range setting of 0.056°C was input along with a "TWO POSITION" thermostat type*
	More comprehensive controls (please specify)

Performance Map: characteristics

A	Default curves
X	Custom curve fitting
	Detailed mapping not available
	Other (please specify)

Performance Map: independent variables

X	Entering Drybulb Temperature. *It only affects sensible capacity in DOE-2.*
X	Entering Wetbulb Temperature
X	Outdoor Drybulb Temperature
X	Part Load Ratio
A	Indoor Fan Air Flow Rate. *Was not used. In the HVACBESTES cases the fan airflow is constant when it is operating.*
	Other (please specify)

Performance Map: dependent variables

X	Coefficient of Performance (or other ratio of load to electricity consumption)
X	Total Capacity
X	Sensible Capacity
X	Bypass Factor
	Other (please specify)

Performance Map: available curve fit techniques

X	Linear, f(one independent variable). *COIL-BF-FPLR*
A	Quadratic, f(one independent variable)
X	Cubic, f(one independent variable). *COOL-EIR-FPLR*
A	Bi-Linear, f(two independent variables)
X	Bi-Quadratic, f(two independent variables). *COOL-EIR-FT, COOL-CAP-FT, COOL-SH-FT, COOL-BF-FT*
	Other (please specify)

Performance Map: extrapolation limits

A	Limits independent variables
A	Limits dependent variables
X	No extrapolation limits
	Extrapolation not allowed
	Other (please specify)

Cooling coil and supply air conditions model

	Supply air temperature = apparatus dew point (ADP); supply air humidity ratio = humidity ratio of saturated air at ADP
A	Bypass factor model using listed ADP data
X	Bypass factor model with ADP calculated from extending condition line
X	Fan heat included
	More comprehensive model (please specify)

Disaggregation of fans' electricity use directly in the simulation and output

A	Indoor fan only. *Other system types*
X	Outdoor fan only
A	Both indoor and outdoor fans disaggregated in the output. *Other system types*
	None - disaggregation of fan outputs with separate calculations by the user

Economizer settings available (for E400 series)

X	Temperature
A	Enthalpy
A	Compressor Lockout
	Other (please specify)

Appendix III-F

Prometheus

Martin Behne, KLIMASYSTEMTECHNIK, Berlin, Germany

[Editor's Note: KST was not able to work on this project after January 2000. Therefore, they were not able to complete refinement of their simulation results, and were only able to submit this preliminary version of their modeler report.]

Introduction

The German participant KLIMASYSTEMTECHIK (KST), Berlin uses the simulation program PROMETHEUS which has been developed within the company. For more than 20 years, PROMETHEUS has been improved and adapted to the needs in modern building and system simulation. The program is used to assess building's energy demand, heating and cooling loads and temperatures. In many cases, it is the company's base for consulting architects and building owners.

For KST, the IEA Task 22 is a very comprehensive opportunity to test and compare the program's capabilities with other simulation tools available and to improve its agreement with real, i.e., measured data (validation), and to exchange knowledge and experiences with other modellers or user of models

Problems concerning the modelling and the documentation provided

No problems occurred with modelling the BESTEST 'test chamber'.

A characteristic of PROMETHEUS, the input file with the weather data has a unique format therefore, weather data in, e.g., TMY format, has to be transformed. However, this is not a special problem within the BESTEST validation test but a typical routine when working with PROMETHEUS.

The specifications provided by NREL about the BESTEST test set-up very well organized, the descriptions, Tables and Figures were clear and all information required was included.

Performance data

However, some problems occurred with the characteristics of the cooling coil capacity. In the first test runs, the cooling capacity was not sufficient to maintain the setpoint temperature with variation E200. This was caused by a misunderstanding of the full-load cooling coil capacity provided in Table 1-6a [see Part 1]. Although, it was said in the description, it was assumed to **not** include the fan heat.

A re-run after the April 1998 Meeting in Golden, CO where the first comparison of the test results were presented ended in a slightly improved performance (COP) and satisfied loads (E200). As a consequence, no differences in loads or energy consumption occurred when comparing with the results of the other models.

The new specifications sent in September 1998 included some changes which were very well explained and easy to understand. The new performance data for the cooling coil was used to re-run the tests and new results were obtained and sent to NREL. The performance data (COP) was calculated considering the new CDF curve. The COP put into the result file has been calculated from hourly values as follows:

$$\mathbf{COP = CDF * COP_{100}}$$

with : $CDF = 1 - 0.229 * (1 - PLR)$

$PLR = (Q_{Coil} - Q_{supply\ fan}) / Q_{coil,\ max}$

$COP_{100} = (Q_{Coil} - Q_{supply\ fan})/(Q_{compressor} + Q_{supply\ fan} + Q_{outdoor\ fan})$

The time (solar time/time zone) had been adjusted within the model according to the equation given.

[Editor's Note: Per communications with KST (Behne, 24 Sep 1998), the CDF effect was calculated externally after the PROMETHEUS simulation, and KST was planning to incorporate a COP=f(PLR) algorithm directly into PROMETHEUS.]

Hotline

A hotline was not needed.

Bugs

There were no bugs detected in PROMETHEUS.

Appendix III-G

HVAC BESTEST MODELER REPORT
ENERGYPLUS VERSION 1.0.0.023

Prepared by
R. Henninger & M. Witte, GARD Analytics, Inc.

July 2001

1. *Introduction*

Software: EnergyPlus Version 1.0.0.023
Authoring Organizations: Ernest Orlando Lawrence Berkeley National Laboratory
University of Illinois
U.S. Army Corps of Engineers,
Construction Engineering Research Laboratories
Oklahoma State University,
GARD Analytics, Inc.
University of Central Florida,
Florida Solar Energy Center
U.S. Department of Energy,
Office of Building Technology, State and Community Programs, Energy Efficiency and Renewable Energy
Authoring Country: USA

HVAC BESTEST was used very effectively during the development of EnergyPlus to identify inconsistencies and errors in results. Included in this report are discussions about changes and results during four different rounds of testing using Beta Version 5-07, Beta Version 5-14, Version 1.0.0.011 (the initial public release, April 2001) and Version 1.0.0.023 (a maintenance release, June 2001).

Related validation work for EnergyPlus has included comparative loads tests using ASHRAE Standard 140 (ASHRAE 2001) which is based on the loads BESTEST suite (Judkoff and Neymark 1995) and analytical building fabric tests using the results of ASHRAE research project 1052RP (Spitler et al 2001). An overview of EnergyPlus testing activities will be presented at the IBPSA Building Simulation 2001 Conference in August 2001 (Witte et al 2001). Selected testing results have been published on the EnergyPlus web site (see URL in References subsection), and additional results will be published as they become available.

2. *Modeling Methodology*

For modeling of the simple unitary vapor compression cooling system, the EnergyPlus Window Air Conditioner model was utilized. No other DX coil cooling system was available at the time that this work began, but others have been added since then. The Window Air Conditioner model consists of three modules for which specifications can be entered: DX cooling coil, indoor fan and outside air mixer. The outside air quantity was set to 0.0. The DX coil model is based upon the DOE-2.1E DX coil simulation algorithms with modifications to the coil bypass factor calculations.

The building envelope loads and internal loads are calculated each hour to determine the zone load that the mechanical HVAC system must satisfy. The DX coil model then uses performance information at rated conditions along with curve fits for variations in total capacity, energy input ratio and part load fraction to determine performance at part load conditions. Sensible/latent capacity splits are determined by the rated sensible heat ratio (SHR) and the apparatus dewpoint/bypass factor approach.

Five performance curves are required:

1) The total cooling capacity modifier curve (function of temperature) is a bi-quadratic curve with two independent variables: wet bulb temperature of the air entering the cooling coil, and dry bulb temperature of the air entering the air-cooled condenser. The output of this curve is multiplied by the rated total cooling capacity to give the total cooling capacity at specific temperature operating conditions (i.e., at temperatures different from the rating point temperatures).

2) The total cooling capacity modifier curve (function of flow fraction) is a quadratic curve with the independent variable being the ratio of the actual air flow rate across the cooling coil to the rated air flow rate (i.e., fraction of full load flow). The output of this curve is multiplied by the rated total cooling capacity and the total cooling capacity modifier curve (function of temperature) to give the total cooling capacity at the specific temperature and air flow conditions at which the coil is operating.

3) The energy input ratio (EIR) modifier curve (function of temperature) is a bi-quadratic curve with two independent variables: wet bulb temperature of the air entering the cooling coil, and dry bulb temperature of the air entering the air-cooled condenser. The output of this curve is multiplied by the rated EIR (inverse of the rated COP) to give the EIR at specific temperature operating conditions (i.e., at temperatures different from the rating point temperatures).

4) The energy input ratio (EIR) modifier curve (function of flow fraction) is a quadratic curve with the independent variable being the ratio of the actual air flow rate across the cooling coil to the rated air flow rate (i.e., fraction of full load flow). The output of this curve is multiplied by the rated EIR (inverse of the rated COP) and the EIR modifier curve (function of temperature) to give the EIR at the specific temperature and airflow conditions at which the coil is operating.

5) The part load fraction correlation (function of part load ratio) is a quadratic curve with the independent variable being part load ratio (sensible cooling load / steady-state sensible cooling capacity). The output of this curve is used in combination with the rated EIR and EIR modifier curves to give the "effective" EIR for a given simulation time step. The part load fraction correlation accounts for efficiency losses due to compressor cycling. In the earlier versions of EnergyPlus, this correction could only be applied to the condensing unit power, but a revision was made to also allow a part load correction for the indoor fan (see Round 4 discussion).

The DX coil model as implemented in EnergyPlus does not allow for simulation of the cooling coil bypass factor characteristics as called out in the specification.

3. Modeling Assumptions

Thermostat Control

Ideal thermostat control was assumed with no throttling range.

DX Coil Curve Fits

Since EnergyPlus utilizes a DX coil model very similar to that used in DOE-2, the performance curves initially used in EnergyPlus were identical to those used in DOE-2. Joel Neymark from NREL, who provided the DOE-2 modeling support for HVAC BESTEST, kindly provided us with a copy of the DOE-2 input files that he used for performing the DOE-2 analysis. Provided with the matrix of performance data in English units for each of the curves, we converted the temperature input variables to metric units and reran DOE-2 to get the curve fit coefficients. (This shortcut on the curves was done in order to save some time. New curve coefficients were developer later, see Round 4.) The resulting coefficients used for the initial runs are presented below.

1) **Total cooling capacity modifier curve** (function of temperature)
Form: Bi-quadratic curve
curve = a + b*wb + c*wb**2 + d*edb + e*edb**2 + f*wb*edb
Independent variables: wet bulb temperature of the air entering the cooling coil, and dry bulb temperature of the air entering the air-cooled condenser.

a = 0.40731210
b = 0.04517144
c = 0.00008412
d = 0.00140582
e = -0.00003830
f = -0.00046771

2) **Total cooling capacity modifier curve** (function of flow fraction)
Form: Quadratic curve
curve = a + b*ff + c*ff**2
Independent variables: ratio of the actual air flow rate across the cooling coil to the rated air flow rate (i.e., fraction of full load flow).

Since the indoor fan always operates at constant volume flow, the modifier will be 1.0, therefore:

a = 1.0
b = 0.0
c = 0.0

3) **Energy input ratio (EIR) modifier curve** (function of temperature)
Form: Bi-quadratic curve
curve = a + b*wb + c*wb**2 + d*edb + e*edb**2 + f*wb*edb
Independent variables: wet bulb temperature of the air entering the cooling coil, and dry bulb temperature of the air entering the air-cooled condenser.

a = 0.72724128
b = -0.02055985
c = 0.00075095
d = 0.01355680
e = 0.00040789
f = -0.00086178

4) **Energy input ratio (EIR) modifier curve** (function of flow fraction)
Form: Quadratic curve
curve = a + b*ff + c*ff**2
Independent variables: ratio of the actual air flow rate across the cooling coil to the rated air flow rate (i.e., fraction of full load flow).

Since the indoor fan always operates at constant volume flow, the modifier will be 1.0, therefore:

a = 1.0
b = 0.0
c = 0.0

5) **Part load fraction correlation** (function of part load ratio)
Form: Quadratic curve
curve = a + b*ff + c*ff**2
Independent variable: part load ratio (sensible cooling load/steady state sensible cooling capacity)

Part load performance specified in Figure 1-3 of Volume 1 of the HVAC BESTEST specification [Part 1], therefore:

a = 0.771
b = -0.229
c = 0.0

4. Modeling Options

Throughout the HVAC BESTEST exercise with EnergyPlus, the Window Air Conditioner model was used to simulate the HVAC system. Subsequent to the initial rounds of testing, two new DX system models have been added to EnergyPlus, Furnace:BlowThru:HeatCool and DXSystem:AirLoop. No attempt was made to utilize Furnace:BlowThru:HeatCool since it does not accommodate a draw-thru fan option. DXSystem:AirLoop is a significantly different equipment configuration which has not been tested with this suite.

5. Modeling Difficulties

Weather Data

The TMY weather files provided as part of the HVAC BESTEST package are not directly usable by EnergyPlus. In order to create an EnergyPlus compatible weather file, the TMY file was first converted to BLAST format using the BLAST weather processor (WIFE). An EnergyPlus translator was then used to convert the weather data from the BLAST format to EnergyPlus format.

Table 1-2 of HVAC BESTEST Volume 1 [Part 1] indicates that the ambient dry-bulb and relative humidity should be as follows for the various data sets:

Data Set	HVAC BESTEST Dry-Bulb Temp.	HVAC BESTEST Relative Humidity
HVBT294.TMY	29.4 C	39%
HVBT350.TMY	35.0 C	28%
HVBT406.TMY	40.6 C	21%
HVBT461.TMY	46.1 C	16%

The converted EnergyPlus weather data set contains slightly different values for ambient relative humidity as indicated below:

Data Set	EnergyPlus Dry-Bulb Temp.	EnergyPlus Relative Humidity
HVBT294.TMY	29.4 C	38.98%
HVBT350.TMY	35.0 C	28.41%
HVBT406.TMY	40.6 C	20.98%
HVBT461.TMY	46.1 C	15.76%

Building Envelope Construction

The specification for the building envelope indicates that the exterior walls, roof and floor are made up of one opaque layer of insulation (R=100) with differing radiative properties for the interior surface and exterior surface (ref. Table 1-4 of Volume 1 [Part 1]). To allow the surface radiative properties to be set at different values, the exterior wall, roof and floor had to be simulated as two insulation layers, each with an R=50. The EnergyPlus description for this construction was as follows:

```
MATERIAL:Regular-R,
        INSULATION-EXT,         ! Material Name
        VerySmooth,             ! Roughness
        50.00,                  ! Thermal Resistance {m2-K/W}
        0.9000,                 ! Thermal Absorptance
        0.1000,                 ! Solar Absorptance
        0.1000;                 ! Visible Absorptance

MATERIAL:Regular-R,
        INSULATION-INT,         ! Material Name
        VerySmooth,             ! Roughness
        50.00,                  ! Thermal Resistance {m2-K/W}
        0.9000,                 ! Thermal Absorptance
        0.6000,                 ! Solar Absorptance
        0.6000;                 ! Visible Absorptance
```

```
CONSTRUCTION,
        LTWALL,                         ! Construction Name
            ! Material layer names follow:
        INSULATION-EXT,
        INSULATION-INT;
```

Indoor Fan

The specification calls for the unitary air conditioner to have a draw-thru indoor fan. The Window Air Conditioner model in early beta versions of EnergyPlus could only model a blow-thru fan configuration. In Version 1 Build 05 and later a draw-thru configuration is also available. This limitation may have affected the latent load on the cooling coil and the compressor energy consumption in the early results (Round 1 and Round 2), but other issues were also contributing errors at that point. A draw-thru fan was modeled in Round 3 and Round 4.

Compressor and Condenser Fan Breakout

The rated COP required as input by the EnergyPlus DX coil model requires that the input power be the combined power for the compressor and condenser fans. As such, there are no separate input variables or output variables available for the compressor or condenser fan. The only output variable available for reporting in EnergyPlus is the DX coil electricity consumption which includes compressor plus condenser fan.

6. Software Errors Discovered and/or Comparison Between Different Versions of the Same Software – Round 1

During the first round of simulations several potential software errors were identified in EnergyPlus Beta Version 5-07:

- Fan electrical power and fan heat were consistently low compared to the analytical results for all tests.

- The reported cooling coil loads were consistently too high and apparently had not been adjusted for the fraction of the time step that the equipment operated, however, the DX coil electricity consumption and actual load delivered to the space were being adjusted appropriately for cycling time.

- For the dry coil cases, the reported sensible coil load was slightly higher than the reported total coil load. Latent load was not available as an output variable, but was calculated by subtracting the sensible from the total. This error caused small negative latent loads to be calculated for the dry coil cases.

- Zone relative humidity was higher for many tests compared to the analytical results, especially for the tests with wet coils. This difference was probably due to simulating a blow-thru configuration rather than the required draw-thru configuration.

Software change requests were posted. Once a new version became available, the tests were rerun.

7. *Results – Round 1*

Results from the first modeling with EnergyPlus Beta 5-07 are presented in Table 1. The evaporator total coil load was too large because cycling during the time step was not accounted for. The negative latent coil loads for cases E100 through E140 result from the reported coil sensible load being greater than the total load.

Table 1 – HVAC BESTEST Results for EnergyPlus Beta 5 Build 07

	February Totals										February Mean			February Maximum			February Minimum		
	Cooling Energy Consumption				Evaporator Coil Load			Zone Load											
Cases	Total (kWh)	Compressor (kWh)	Supply Fan (kWh)	Condenser Fan (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)
E100	1517.1		136.5		4210.0	4265.4	-55.4	3653.7	3653.7	0.0	2.41	22.2	0.0075	2.41	22.2	0.0076	2.37	22.2	0.0074
E110	1029.8		114.0		4979.6	5036.4	-56.8	3635.2	3635.2	0.0	3.53	22.2	0.0064	3.54	22.2	0.0064	3.48	22.2	0.0063
E120	988.3		107.1		5380.0	5455.8	-75.8	3630.2	3630.2	0.0	3.67	26.7	0.0080	3.68	26.7	0.0081	3.62	26.7	0.0079
E130	105.0		7.7		4210.8	4267.0	-56.2	206.3	206.3	0.0	1.96	22.2	0.0075	1.97	22.2	0.0076	1.93	22.2	0.0074
E140	63.1		5.9		4979.6	5036.9	-57.2	187.8	187.8	0.0	2.98	22.2	0.0064	2.98	22.2	0.0064	2.93	22.2	0.0063
E150	1185.1		133.9		5129.7	4328.5	801.2	4374.4	3635.2	739.2	3.69	22.2	0.0083	3.71	22.2	0.0084	3.68	22.2	0.0082
E160	1124.0		122.2		5700.6	4821.1	879.6	4369.4	3630.2	739.2	3.89	26.7	0.0101	3.91	26.7	0.0101	3.87	26.7	0.0100
E165	1495.7		144.8		4790.6	4053.8	736.8	4385.5	3646.3	739.2	2.93	23.3	0.0093	2.95	23.3	0.0093	2.92	23.3	0.0092
E170	622.0		62.0		5492.9	3688.1	1804.9	2156.8	1417.6	739.2	3.47	22.2	0.0106	3.50	22.2	0.0106	3.45	22.2	0.0105
E180	1088.1		112.1		6250.7	2138.4	4112.3	4374.4	1417.6	2956.8	4.02	22.2	0.0165	4.09	22.2	0.0165	3.96	22.2	0.0164
E185	1570.8		136.4		5182.5	1807.7	3374.8	4392.9	1436.1	2956.8	2.80	22.2	0.0164	2.85	22.2	0.0164	2.75	22.2	0.0162
E190	161.5		14.3		6250.7	2217.2	4033.5	557.4	187.8	369.6	3.45	22.2	0.0162	3.52	22.2	0.0163	3.37	22.2	0.0160
E195	247.6		17.9		5175.4	1963.8	3211.7	575.9	206.3	369.6	2.33	22.2	0.0158	2.37	22.2	0.0159	2.27	22.2	0.0156
E200	1472.6		153.7		5562.7	4380.1	1182.5	5341.1	4120.2	1221.0	3.63	26.7	0.0113	3.65	26.7	0.0114	3.61	26.7	0.0112

8. *Software Errors Discovered and/or Comparison Between Different Versions of the Same Software – Round 2*

EnergyPlus Beta 5-14 included changes to fix the following problems which were identified in HVAC BESTEST Round 1:

- Reporting of cooling coil loads were corrected to account for run time during cycling operation.
- The methods of calculating SHR and coil bypass factor were modified to eliminate the problem where the dry coil cases reported sensible coil loads which were slightly higher than the reported total coil loads. This error was causing small negative latent loads to be calculated for the dry coil cases.

During the second round of simulations with EnergyPlus Beta 5-14 the cooling coil error identified during the first round of simulations was corrected to account for cycling during each time step, and this brought the evaporator coil loads closer to the range of results for the other programs; but the loads were still higher than they should be. Another potential error was therefore identified which may have been masked by the coil problem identified in Round 1:

- Although there was excellent agreement for zone total cooling load, the evaporator cooling coil load was larger than the zone cooling load plus fan heat..
- Also, the mean indoor dry bulb for Case E200 moved from 26.7C to 27.1C.
- The other problems identified in Round 1 still remained (low fan power, poor agreement in zone humidity ratio).

9. *Results – Round 2*

Results from the second round of simulations with EnergyPlus Beta 5-14 are presented in Table 2.

Table 2 – HVAC BESTEST Results for EnergyPlus Beta 5 Build 14

	February Totals										February Mean			February Maximum			February Minimum		
	Cooling Energy Consumption				Evaporator Coil Load			Zone Load											
Cases	Total (kWh)	Compressor (kWh)	Supply Fan (kWh)	Condenser Fan (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)
E100	1535.8		138.7		3842.1	3842.1	0.0	3653.7	3653.7	0.0	2.38	22.2	0.0074	2.38	22.2	0.0074	2.38	22.2	0.0074
E110	1039.6		115.2		3792.2	3792.2	0.0	3635.2	3635.2	0.0	3.50	22.2	0.0062	3.50	22.2	0.0062	3.49	22.2	0.0062
E120	1003.0		109.2		3792.0	3792.0	0.0	3630.2	3630.2	0.0	3.62	26.7	0.0078	3.63	26.7	0.0078	3.61	26.7	0.0078
E130	106.6		7.8		216.9	216.9	0.0	206.3	206.3	0.0	1.93	22.2	0.0074	1.94	22.2	0.0074	1.93	22.2	0.0074
E140	63.8		6.0		195.9	195.9	0.0	187.8	187.8	0.0	2.94	22.2	0.0062	2.95	22.2	0.0062	2.94	22.2	0.0062
E150	1197.9		135.6		4589.9	3820.1	769.8	4374.4	3635.2	739.2	3.65	22.2	0.0084	3.67	22.2	0.0084	3.64	22.2	0.0083
E160	1139.1		124.1		4587.2	3814.9	772.3	4369.4	3630.2	739.2	3.84	26.7	0.0102	3.86	26.7	0.0102	3.82	26.7	0.0101
E165	1513.5		146.9		4620.2	3849.7	770.4	4385.5	3646.3	739.2	2.90	23.3	0.0094	2.92	23.3	0.0094	2.88	23.3	0.0093
E170	630.3		62.9		2272.2	1502.4	769.7	2156.8	1417.6	739.2	3.42	22.2	0.0107	3.45	22.2	0.0107	3.40	22.2	0.0106
E180	1104.9		114.2		4640.3	1561.5	3078.9	4374.4	1417.6	2956.8	3.96	22.2	0.0166	4.02	22.2	0.0166	3.90	22.2	0.0165
E185	1594.9		139.0		4686.1	1607.2	3078.9	4392.9	1436.1	2956.8	2.75	22.2	0.0165	2.81	22.2	0.0165	2.71	22.2	0.0163
E190	164.4		14.5		591.1	206.2	384.9	557.4	187.8	369.6	3.39	22.2	0.0163	3.45	22.2	0.0164	3.31	22.2	0.0162
E195	251.9		18.2		613.9	229.0	384.9	575.9	206.3	369.6	2.29	22.2	0.0159	2.33	22.2	0.0160	2.23	22.2	0.0157
E200	1486.6		155.2		5627.7	4351.7	1276.0	5340.7	4119.7	1221.0	3.59	27.1	0.0116	3.60	27.2	0.0117	3.59	27.0	0.0115

10. *Software Errors Discovered and/or Comparison Between Different Versions of the Same Software – Round 3*

The suite of HVAC BESTEST cases were simulated again using EnergyPlus Version 1.0.0.011 (the first public release of Version 1.0, April 2001) which included the following changes from Beta 5-14:

- Modified method for calculating coil outlet conditions.
- Changed to use of Double Precision throughout all of EnergyPlus. (This change was prompted by various issues not related to HVAC BESTEST.)
- Added two output variables for tracking run time
  Window AC Fan RunTime Fraction
  Window AC Compressor RunTime Fraction

- Added an output variable for coil latent load.
- Added Draw-Thru Fan option to Window AC.
- The name of the DX coil object was changed from COIL:DX:DOE2 to COIL:DX:BF-Empirical to better represent its algorithmic basis.

In addition, the following input file changes were made :

- Changed from blow-thru fan to draw-thru configuration.
- Updated the DX coil object name to COIL:DX:BF-Empirical.

The following changes in results were observed:

- Indoor fan power consumption and fan heat decreased significantly from Round 2, moving farther below the analytical results.
- Space cooling electricity consumption changed slightly from Round 2 and moved closer to the analytical results.
- Mean indoor humidity ratio decreased compared to Round 2, moving farther away from the analytical results for most of the dry coil cases and moving closer to the analytical results for the wet coil cases.
- Mean indoor dry bulb for Case E200 moved further out of range to 27.5C (the setpoint for this case is 26.7C).

In general, except for fan power and fan heat, the overall EnergyPlus Version 1.0.0.011 results compared much better to the HVAC BESTEST analytical results.

11. Results – Round 3

Results from the third round of simulations with EnergyPlus Version 1.0.0.011 are presented in Table 3.

Table 3 – HVAC BESTEST Results for EnergyPlus Version 1 Build 11

	February Totals										February Mean			February Maximum			February Minimum		
	Cooling Energy Consumption				Evaporator Coil Load			Zone Load											
Cases	Total (kWh)	Compressor (kWh)	Supply Fan (kWh)	Condenser Fan (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)
E100	1527.6		132.6		3834.9	3834.9	0.0	3654.1	3654.1	0.0	2.39	22.2	0.0071	2.39	22.2	0.0071	2.39	22.2	0.0071
E110	1032.2		109.2		3785.3	3785.3	0.0	3635.6	3635.6	0.0	3.52	22.2	0.0060	3.53	22.2	0.0060	3.52	22.2	0.0060
E120	1001.3		104.4		3786.5	3786.5	0.0	3630.5	3630.5	0.0	3.63	26.7	0.0077	3.63	26.7	0.0077	3.62	26.7	0.0077
E130	106.3		7.5		217.0	217.0	0.0	206.7	206.7	0.0	1.94	22.2	0.0071	1.95	22.2	0.0071	1.94	22.2	0.0071
E140	63.5		5.7		195.9	195.9	0.0	188.2	188.2	0.0	2.96	22.2	0.0060	2.96	22.2	0.0060	2.96	22.2	0.0060
E150	1197.5		130.9		4584.5	3815.0	769.5	4374.7	3635.5	739.2	3.65	22.2	0.0082	3.68	22.2	0.0082	3.64	22.2	0.0081
E160	1137.3		119.1		4581.3	3809.3	772.1	4369.7	3630.5	739.2	3.84	26.7	0.0100	3.87	26.7	0.0100	3.83	26.7	0.0099
E165	1514.9		142.2		4614.7	3844.5	770.2	4385.9	3646.7	739.2	2.90	23.3	0.0092	2.92	23.3	0.0092	2.88	23.3	0.0091
E170	631.2		61.0		2270.2	1500.7	769.5	2157.1	1417.9	739.2	3.42	22.2	0.0105	3.45	22.2	0.0105	3.40	22.2	0.0104
E180	1100.9		110.8		4636.5	1558.6	3077.9	4374.7	1418.0	2956.8	3.97	22.2	0.0163	4.04	22.2	0.0163	3.92	22.2	0.0162
E185	1590.5		135.5		4682.1	1604.2	3077.9	4393.3	1436.5	2956.8	2.76	22.2	0.0161	2.81	22.2	0.0162	2.72	22.2	0.0160
E190	164.0		14.1		591.0	206.2	384.8	557.8	188.2	369.6	3.40	22.2	0.0160	3.46	22.2	0.0161	3.33	22.2	0.0158
E195	253.3		17.9		613.9	229.2	384.8	576.3	206.7	369.6	2.28	22.2	0.0156	2.32	22.2	0.0156	2.23	22.2	0.0154
E200	1479.4		148.4		5621.0	4345.1	1276.0	5340.7	4119.7	1221.0	3.61	27.5	0.0116	3.62	27.6	0.0117	3.61	27.4	0.0115

12. Software Errors Discovered and/or Comparison Between Different Versions of the Same Software – Round 4

The suite of HVAC BESTEST cases were simulated again using EnergyPlus Version 1.0.0.023 (a maintenance release, June 2001), which included both input file and source code changes from Version 1.0.0.011.

Input file changes for Round 4:

- The equipment performance curves were refit from scratch using the Excel function LINEST. Data for the curves were taken from Table 1-6c of the HVAC BESTEST specification [Part 1]. Curve fits were developed using SI units since this is what EnergyPlus requires. Previously, the DOE-2 curve coefficients from Neymark's work had been used, but the EIR curve fit done for DOE-2 applied only to the compressor input power. The EIR curve required for the EnergyPlus DX Coil model is based on compressor input power plus outdoor condenser fan power. The resulting curves used for the latest round of EnergyPlus simulations were as follows:

  CoolCapFT = a + b*wb + c*wb**2 + d*edb + e*edb**2 + f*wb*edb
  where
  wb = wet-bulb temperature of air entering the cooling coil
  edb = dry-bulb temperature of the air entering the air-cooled condenser
  a = 0.43863482
  b = 0.04259180
  c = 0.00015024
  d = 0.00100248
  e = -0.00003314
  f = -0.00046664
  Data points were taken from first three columns of Table 1-6c of specification [Part 1]. CoolCap data was normalized to ARI rated capacity of 8,181 W, i.e. CoolCapFT = 1.0 at 19.4 C wb and 35.0 C edb.

  EIRFT = a + b*wb + c*wb**2 + d*edb + e*edb**2 + f*wb*edb
  where:

wb = wet-bulb temperature of air entering the cooling coil
edb = dry-bulb temperature of the air entering the air-cooled condenser
a = 0.77127580
b = -0.02218018
c = 0.00074086
d = 0.01306849
e = 0.00039124
f = -0.00082052

edb and wb data points were taken from the first two columns of Table 1-6c of specification [Part 1]. Energy input data points for corresponding pairs of edb and wb were taken from column labeled "Compressor Power" in Table 1-6c [Part 1] with an additional 108 W added to them for outdoor fan power. EIR is energy input ratio [(compressor + outdoor fan power)/cooling capacity] normalized to ARI rated conditions, i.e. EIRFT = 1.0 at 19.4 C wb and 35.0 C edb.

- Relaxed the min/max limits of the performance curve independent variables, wb and edb, to allow extrapolation of CoolCapFT and EIRFT outside the bounds of the equipment performance data given in the specification in accordance with comments in Section 1.3.2.2.3.2 of Part 1.

- The BESTEST CDF curve was determined based on net total capacities of the unit while the EnergyPlus DX Coil model requires that the part load curve be expressed on the basis of gross sensible capacities. A new CDF curve was developed which was intended to be on a gross capacity basis, but a later review of this curve showed an error in the derivation. Further review showed that there is really little difference between net part load and gross part load, so the revised curve was then removed and the original CDF curve was used.

- The CDF curve (part load curve) was applied to the indoor fan operation where previously there was no input available for this. This change also required using the FAN:SIMPLE:ONOFF object instead of FAN:SIMPLE:CONSTVOLUME which has been used previously.

- Added one week of infiltration to the beginning of the Case E120 run period to prevent overdrying of the zone during the simulation warmup period. (See the results discussion below for more details.)

Relevant source code changes from Version 1.0.0.011 to Version 1.0.0.023:

- Standard air conditions for converting volume flow to mass flow in the indoor fan calculations were changed. HVAC BESTEST specifies that the volume flow rate is for dry air at 20C. EnergyPlus was using a dry-bulb of 25C at the initial outdoor barometric pressure with a humidity ratio of 0.014 kg/kg, although the EnergyPlus documentation indicated 21C and 101325 Pa was being used. EnergyPlus now calculates the initial air mass flow based on dry air at 20C at the standard barometric pressure for the specified altitude, and the documentation reflects this change.

- The specific heat for air throughout the air-side HVAC simulation was changed from a dry c_p basis to a moist c_p basis. Previously, a mixture of dry and moist c_p had been used for various HVAC calculations.

- The heat of vaporization (h_{fg}) for converting a zone latent load into a load in the HVAC system was changed.

- A new input field was added to FAN:SIMPLE:ONOFF to allow a CDF curve (part load curve) to be applied to the indoor fan operation where previously part load adjustments could only be applied to the compressor and outdoor fan.

- Changed the moisture initialization to use the initial outdoor humidity ratio to initialize all HVAC air nodes.

The following changes in results were observed:

- The sensible and latent coil loads improved and now track very close to the analytical results.

- The mean indoor temperature for Case E200 improved and now, along with rest of the cases, matches exactly with the analytical results.

- The mean indoor humidity ratio tracks the analytical values better, especially for the wet coil cases. For Case E120 however, the EnergyPlus humidity ratio (0.0038) was much less than the analytical value (0.0079). Introducing infiltration for the first week of January only and then turning infiltration off, eliminates this problem and gives a mean indoor humidity ratio for the month of February of 0.0081. Even though all nodes are initialized to the outdoor humidity ratio at the beginning of the simulation, conditions during the simulation warmup days overdry the zone for this case. Without the infiltration during the first week, there is no source of moisture to overcome the overdrying and establish the desired equilibrium.

- Indoor fan power consumption and fan heat match analytical results in most cases or are slightly less than analytical results.

- COP results changed but are still mixed. One problem may have to do with the basis of the CDF curve in BESTEST versus what EnergyPlus requires. The BESTEST CDF curve was determined based on net total capacities of the unit while the EnergyPlus DX Coil model requires that the part load curve be expressed on the basis of gross sensible capacities.

13. Results – Round 4

Results from the fourth round of simulations with EnergyPlus Version 1.0.0.023 are presented in Table 4.

Table 4 – HVAC BESTEST Results for EnergyPlus Version 1 Build 23

	February Totals										February Mean			February Maximum			February Minimum		
	Cooling Energy Consumption				Evaporator Coil Load			Zone Load											
Cases	Total (kWh)	Compressor (kWh)	Supply Fan (kWh)	Condenser Fan (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	Total (kWh)	Sensible (kWh)	Latent (kWh)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)	COP	IDB (°C)	Humidity Ratio (kg/kg)
E100	1520.0		143.6		3797.6	3797.6	0.0	3654.1	3654.1	0.0	2.40	22.2	0.0075	2.41	22.2	0.0075	2.40	22.2	0.0075
E110	1069.1		127.5		3763.1	3763.1	0.0	3635.5	3635.5	0.0	3.40	22.2	0.0066	3.40	22.2	0.0066	3.40	22.2	0.0066
E120	1006.4		116.4		3746.9	3746.9	0.0	3630.5	3630.5	0.0	3.61	26.7	0.0080	3.61	26.7	0.0080	3.60	26.7	0.0080
E130	108.6		10.3		217.0	217.0	0.0	206.7	206.7	0.0	1.90	22.2	0.0075	1.91	22.2	0.0075	1.90	22.2	0.0075
E140	67.9		8.1		196.3	196.3	0.0	188.2	188.2	0.0	2.77	22.2	0.0066	2.78	22.2	0.0066	2.77	22.2	0.0066
E150	1197.1		140.2		4508.7	3776.0	732.7	4374.7	3635.6	739.2	3.65	22.2	0.0084	3.68	22.2	0.0084	3.64	22.2	0.0083
E160	1131.7		128.3		4491.0	3759.0	732.0	4369.7	3630.5	739.2	3.86	26.7	0.0103	3.88	26.7	0.0103	3.84	26.7	0.0102
E165	1491.1		148.5		4528.7	3795.5	733.2	4385.9	3646.7	739.2	2.94	23.3	0.0094	2.96	23.3	0.0094	2.93	23.3	0.0093
E170	635.4		73.0		2224.9	1491.2	733.6	2157.1	1417.9	739.2	3.40	22.2	0.0106	3.42	22.2	0.0106	3.37	22.2	0.0105
E180	1082.0		118.4		4481.2	1537.3	2943.9	4374.7	1418.0	2956.8	4.04	22.2	0.0162	4.11	22.2	0.0162	3.99	22.2	0.0161
E185	1540.4		139.1		4522.6	1576.6	2946.0	4393.3	1436.5	2956.8	2.85	22.2	0.0161	2.90	22.2	0.0161	2.80	22.2	0.0159
E190	164.3		18.0		574.3	206.4	367.9	557.8	188.2	369.6	3.39	22.2	0.0159	3.45	22.2	0.0159	3.32	22.2	0.0157
E195	250.2		22.7		597.7	229.6	368.1	576.3	206.7	369.6	2.30	22.2	0.0154	2.35	22.2	0.0155	2.25	22.2	0.0153
E200	1464.6		153.4		5484.5	4274.3	1210.2	5341.5	4120.5	1221.0	3.65	26.7	0.0115	3.67	26.7	0.0115	3.63	26.7	0.0113

14. Comparison of Changes that Occurred with Versions of EnergyPlus

This section documents the comparative changes that took place in results (see Figures 1 through 6) as modifications were made to the EnergyPlus code or changes were made in the modeling approach (see Table 5). The analytical results shown in Figures 1 –6 represent the baseline against which all EnergyPlus results were compared. Results for other intermediate versions of EnergyPlus not discussed above have been included. EnergyPlus Version 1.0.0.023 (June 2001) is the most current public release of the software.

Table 5 – Summary of EnergyPlus Changes that were Implemented

Version	Input File Changes	Code Changes
Beta 5-12 thru Beta 5-14		DX coil calculations modified to account for cycling Modified method of calculating SHR and coil bypass factor
Beta 5-15 thru Beta 5-18	Changed DX coil object names	Changed name of DX coil object from COIL:DX:DOE2 to COIL:DX:BF-Empirical to better represent its algorithmic basis (no impact on results)
Ver 1-01 thru Ver 1-11	Changed from blow-thru to draw-thru fan configuration	Changed to double precision Modified method of calculating coil outlet conditions Added draw-thru fan option to WindowAC model
Ver 1-12 thru Ver 1-14	New equipment performance curves Adjusted fan mass flow and efficiency to achieve desired mass flow and fan power	
Ver 1-15 thru Ver 1-17	Went back to specified values for fan mass flow and efficiency	Partial implementation of moist c_p Fan power calculated using a standard initial density for volume to mass flow conversion
Ver 1-18 thru Ver 1-19	Changed basis of CDF curve from net to gross Opened up min/max limits for performance curves	Complete implementation of moist c_p h_{fg} calculation modified for latent loads
Ver 1-20 thru Ver 1-23	Went back to original CDF curve (modified curve used with Ver 1-19 was incorrect) Changed from FAN:SIMPLE:CONSTVOLUME to FAN:SIMPLE:ONOFF Used CDF curve for fan power to account for cycling	Implemented optional PLR curve for fan cycling Changed moisture initializations to use outdoor humidity ratio

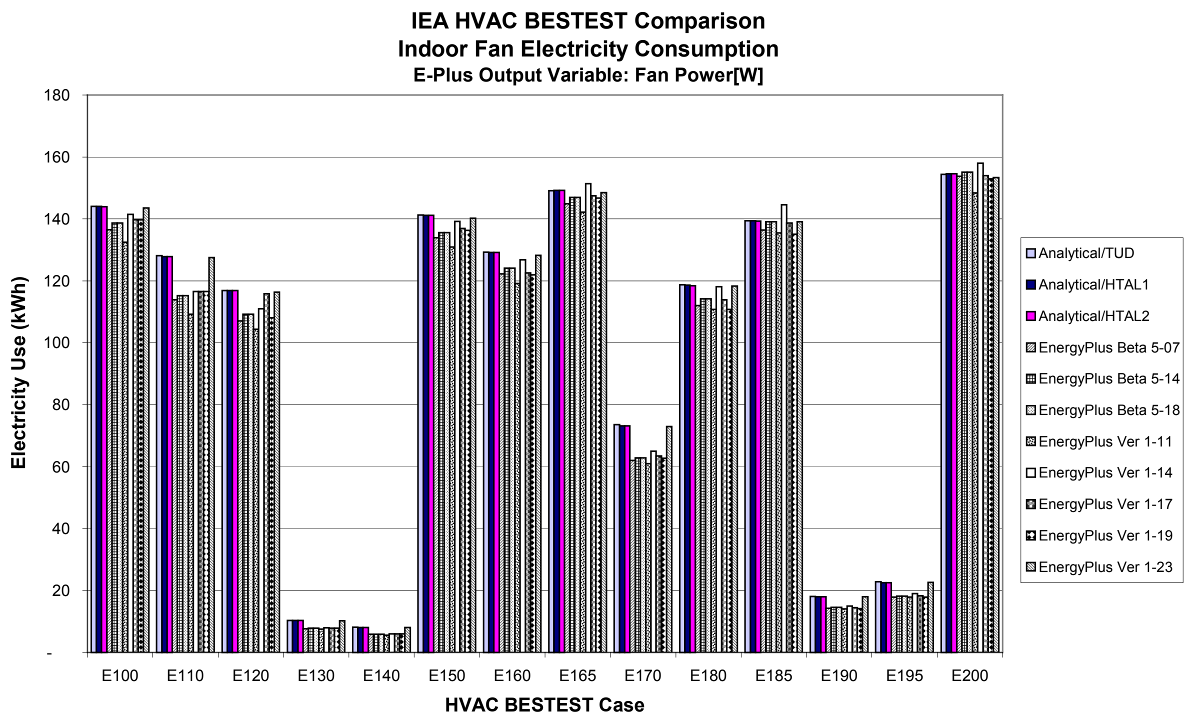


Figure 1 Indoor Fan Power Results for Versions of EnergyPlus

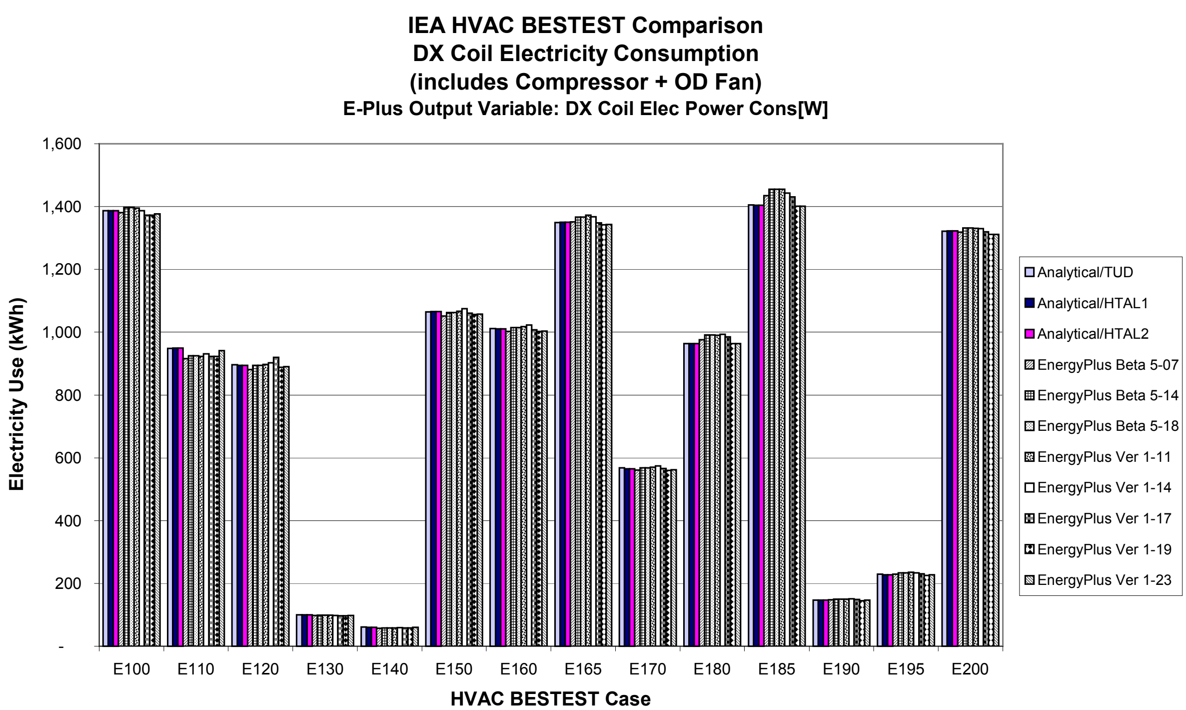


Figure 2 Compressor Plus Outdoor Fan Electricity Consumption Results for Versions of EnergyPlus

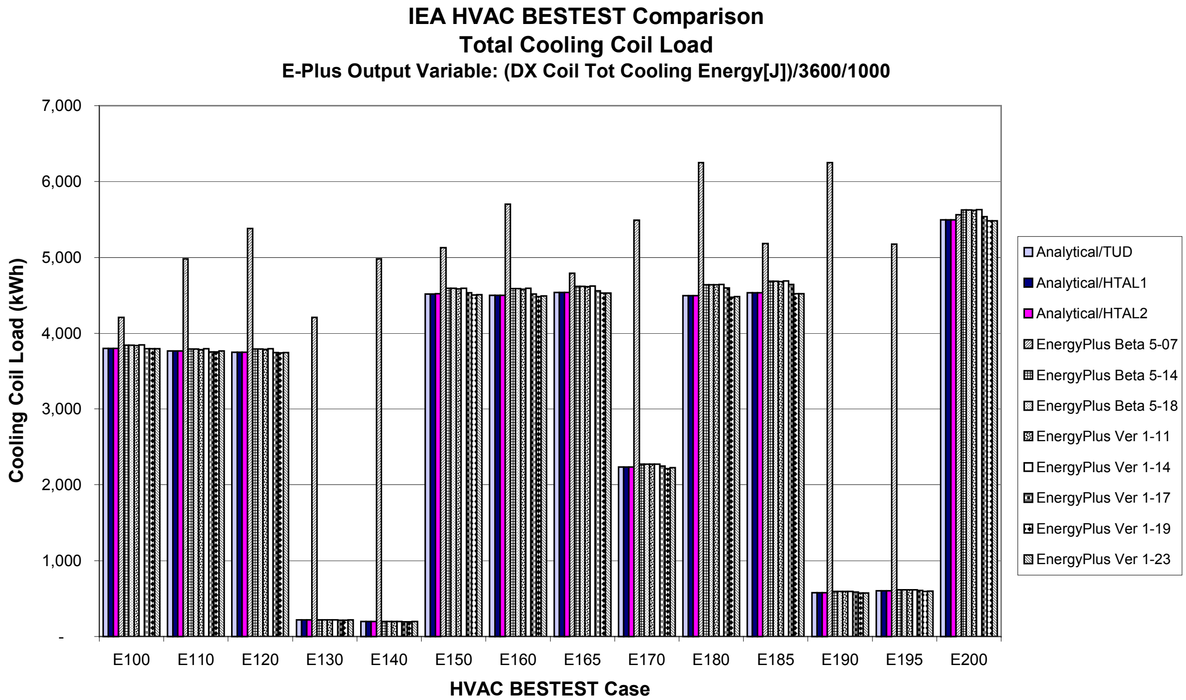


Figure 3 Total Cooling Coil Load Results for Versions of EnergyPlus

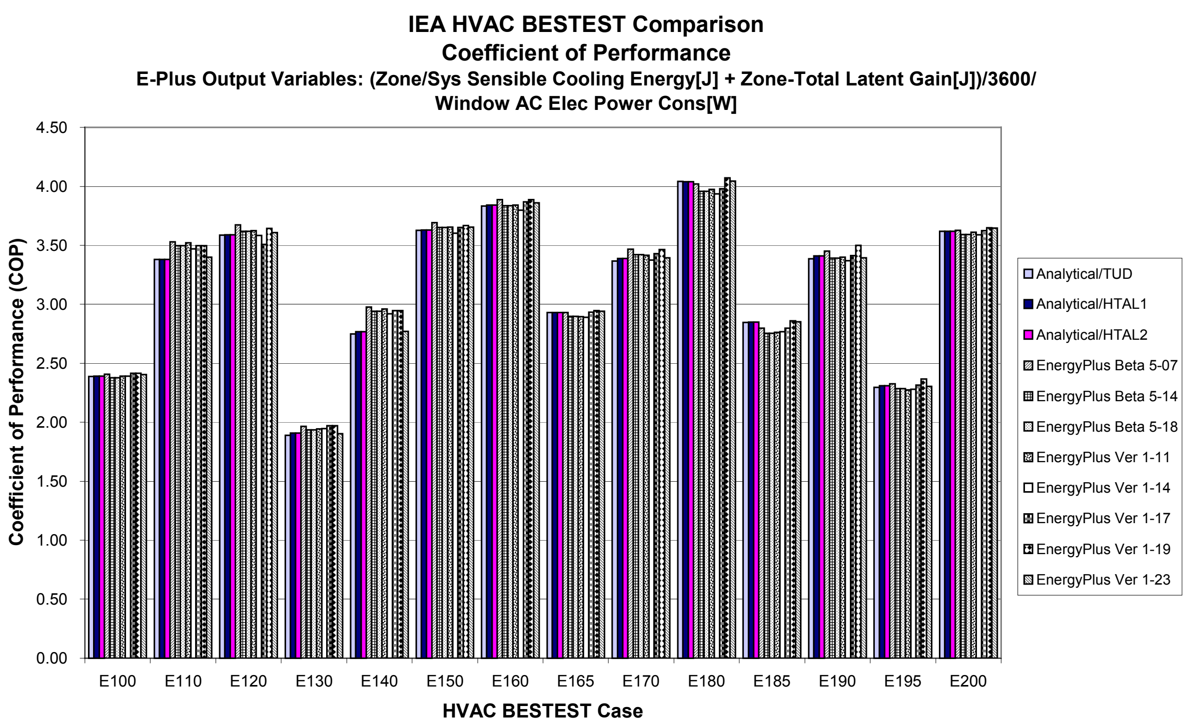


Figure 4 Coefficient of Performance Results for Versions of EnergyPlus

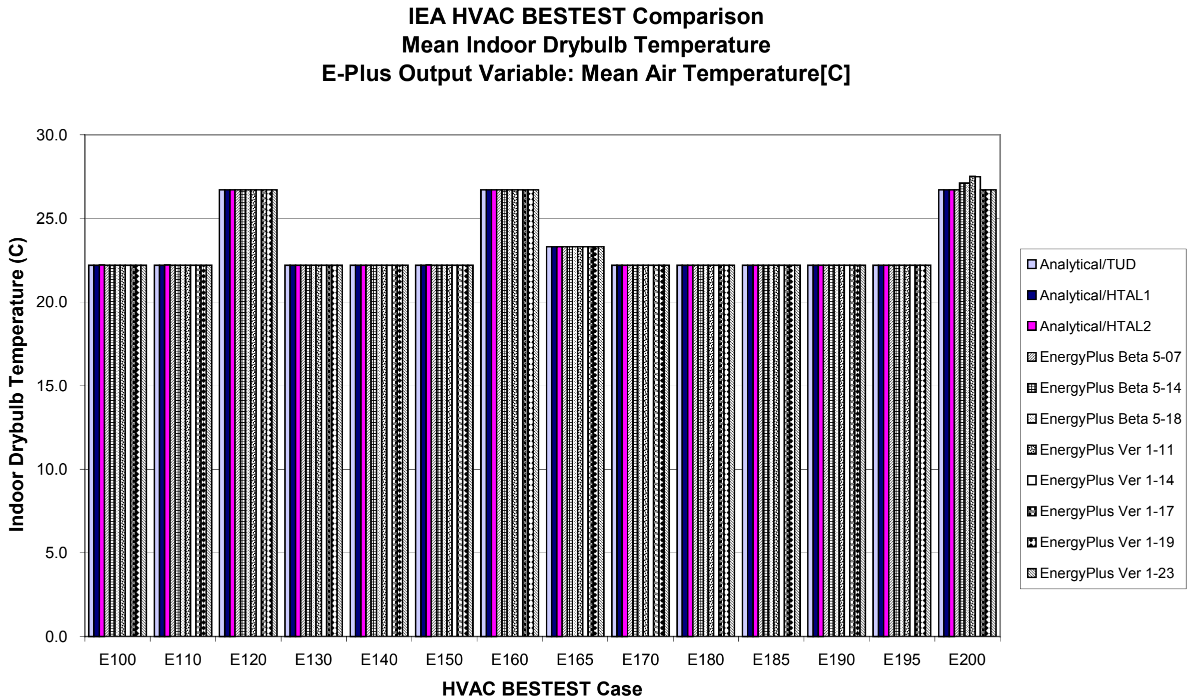


Figure 5 Indoor Dry-Bulb Temperature for Versions of EnergyPlus

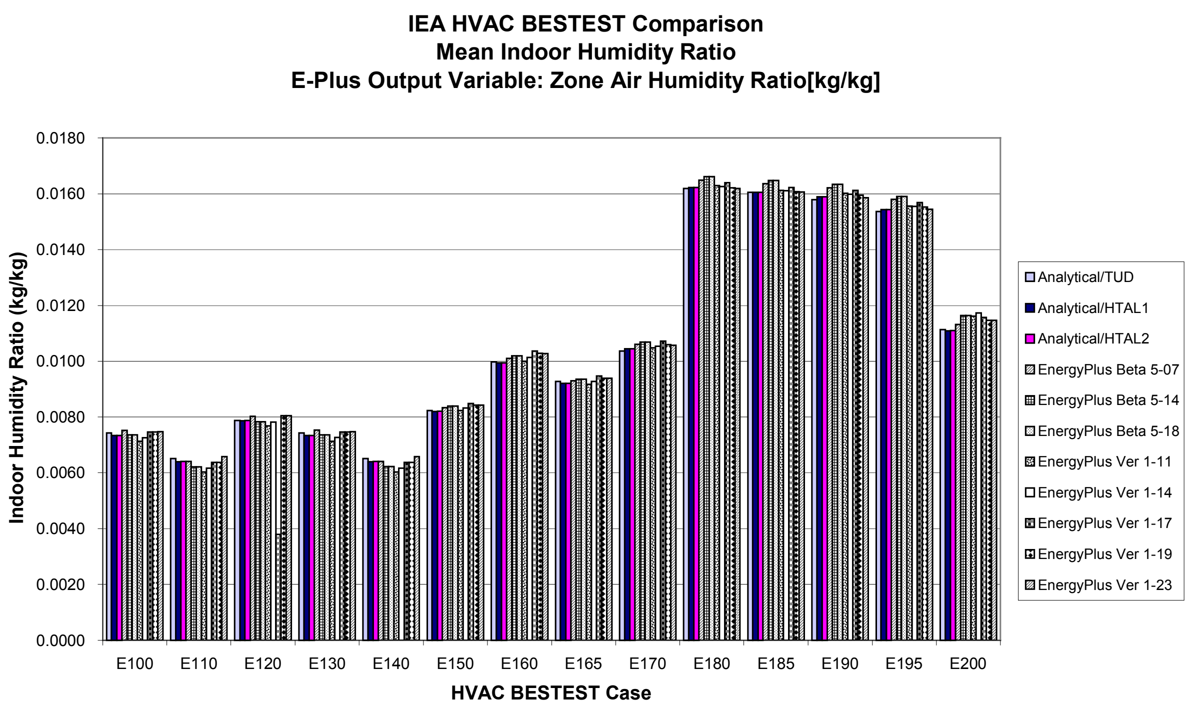


Figure 6 Indoor Humidity Ratio Results for Versions of EnergyPlus

15. Conclusions

The HVAC BESTEST suite is a very valuable testing tool which provides excellent benchmarks for testing HVAC system and equipment algorithms versus the results of other international building simulation programs. As discussed above, HVAC BESTEST allowed the developers of EnergyPlus to identify errors in algorithms and improve simulation accuracy.

16. References

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http://www.eren.doe.gov/buildings/energy_tools/energyplus/

Program name (please include version number)

EnergyPlus Version 1.0.0.023

Your name, organisation, and country

Michael J. Witte, GARD Analytics, Inc., United States

Program status

	Public domain
	Commercial:
	Research
x	Other (please specify): *Government-sponsored, end-user license is no charge, other license types have fees associated with them*

Solution method for unitary space cooling equipment

x	Overall Performance Maps
	Individual Component Models
	Constant Performance (no possible variation with entering or ambient conditions)
	Other (please specify)

Interaction between loads and systems calculations

x	Both are calculated during the same timestep
	First, loads are calculated for the entire simulation period, then equipment performance is calculated separately
	Other (please specify)

Time step

	Fixed within code (please specify time step):
x	User-specified (please specify time step): *one hour for envelope*
x	Other (please specify): *program automatically adjusts HVAC time step, <= envelope time step*

Timing convention for meteorological data : sampling interval

	Fixed within code (please specify interval):
x	User-specified: *one hour*

Timing convention for meteorological data : period covered by first record

x	Fixed within code (please specify period or time which meteorological record covers): *0:00 - 1:00*
	User-specified

Meteorological data reconstitution scheme

	Climate assumed stepwise constant over sampling interval
x	Linear interpolation used over climate sampling interval
	Other (please specify)

Output timing conventions

	Produces spot predictions at the end of each time step
	Produces spot output at end of each hour
x	Produces average outputs for each hour (please specify period to which value relates): *user-specified, hourly data is average or sum for previous hour, can specify output at each time step*

Treatment of zone air

x	Single temperature (i.e. good mixing assumed)
	Stratified model
	Simplified distribution model
	Full CFD model
	Other (please specify)

Zone air initial conditions

x	Same as outside air
	Other (please specify)

Internal gains output characteristics

	Purely convective
	Radiative/Convective split fixed within code
x	Radiative/Convective split specified by user: *100% convective for these tests*
	Detailed modeling of source output

Mechanical systems output characteristics

x	Purely convective
	Radiative/Convective split fixed within code
a	Radiative/Convective split specified by user: *for types of equipment not used in these tests*
	Detailed modeling of source output

Control temperature

x	Air temperature
	Combination of air and radiant temperatures fixed within the code
	User-specified combination of air and radiant temperatures
	User-specified construction surface temperatures
	User-specified temperatures within construction
	Other (please specify)

Control properties

x	Ideal control as specified in the user's manual
	On/Off thermostat control
	On/Off thermostat control with hysteresis
	On/Off thermostat control with minimum equipment on and/or off durations
	Proportional control
	More comprehensive controls (please specify)

Performance Map: characteristics

	Default curves
x	Custom curve fitting
	Detailed mapping not available
	Other (please specify)

Performance Map: independent variables

	Entering Drybulb Temperature: *program calculates adjustments internally*
x	Entering Wetbulb Temperature
x	Outdoor Drybulb Temperature
x	Part Load Ratio
a	Indoor Fan Air Flow Rate: *always=1, because fan always operates at rated conditions*
	Other (please specify)

Performance Map: dependent variables

x	Coefficient of Performance (or other ratio of load to electricity consumption)
x	Total Capacity
	Sensible Capacity: *program calculates internally based on user-specified nominal SHR*
	Bypass Factor: *program calculates internally based on nominal SHR and current conditions*
x	Other (please specify): *indoor fan power (function of PLR)*

Performance Map: available curve fit techniques

x	Linear, f(one independent variable): *flow fraction curves set to constant=1*
x	Quadratic, f(one independent variable) : *PLF-FPLR (cycling loss)*
a	Cubic, f(one independent variable):
a	Bi-Linear, f(two independent variables)
x	Bi-Quadratic, f(two independent variables): *CAP-FT, EIR-FT*
	Other (please specify)

Performance Map: extrapolation limits

x	Limits independent variables: *27.4 <= ODB <=48.1; 13.0 <= EWB <= 23.7, 0.0 <= PLR <= 1.0*
	Limits dependent variables
	No extrapolation limits
	Extrapolation not allowed
	Other (please specify)

Cooling coil and supply air conditions model

	Supply air temperature = apparatus dew point (ADP); supply air humidity ratio = humidity ratio of saturated air at ADP
	Bypass factor model using listed ADP data
x	Bypass factor model with ADP calculated from extending condition line: *nominal BF is calculated from user-specified nominal SHR*
x	Fan heat included
	More comprehensive model (please specify)

Disaggregation of fans' electricity use directly in the simulation and output

x	Indoor fan only
	Outdoor fan only
	Both indoor and outdoor fans disaggregated in the output
	None - disaggregation of fan outputs with separate calculations by the user

Economizer settings available (for E400 series)

a	Temperature *(E400 series not run)*
a	Enthalpy *(E400 series not run)*
a	Compressor Lockout *(E400 series not run)*
	Other (please specify)

PART IV:

Simulation Field Trial Results

4.0 Part IV: Simulation Field Trial Results

Introduction

Here we present the simulation results for the field trials of cases E100–E200. This results set is the final version after numerous iterations to incorporate clarifications to the test specification, simulation input deck corrections, and simulation software improvements. An electronic version of the final results is included on the accompanying CD in the file RESULTS.XLS, with its navigation instructions included in RESULTS.DOC. The results are presented here in the following order:

- Text summarizing the results organized by graph titles
- Graphs of results (beginning on p. IV-9)
- Tables of results (beginning on p. IV-23).

Table 4-1 summarizes the following information for the 11 models that 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 tables, graphs, and the following text.

Except for the PROMETHEUS results, these results have been updated to be consistent with all test specification revisions through May 2000. The PROMETHEUS participants (KST) were not able to work on this project after January 2000; therefore, they were unable to complete the refinement of their simulation results.

Independent simulations of the same program by separate organizations (such as has occurred with DOE-2.1E) minimized the potential for user errors for those simulations.

The text summarizes remaining disagreements of the simulations versus the analytical solutions observed in reviewing the data. Most of the discrepancies seen in previous results sets have been addressed (see Part III). However, a few disagreements remain.

Examples of Shorthand Language Used in the Graphs

Case descriptions are summarized in Tables 1-1a and 1-1b (see Part 1). We have attempted to give a brief description of the cases in the x-axis labels of the accompanying graphs. The resulting shorthand language for these labels work according to the following examples; see Section 3.7 of Part III for definition of acronyms.

"E110 as100 lo ODB" means the data being shown is for case E110 and case E110 is exactly like case E100 except the ODB (outdoor dry-bulb temperature) was reduced. Similarly for the sensitivity plots, "E165-E160 IDB+ODB @hiSH" means the data shown are for the difference between cases E165 and E160, and this difference tests sensitivity to a variation in both IDB and ODB occurring at a high sensible heat ratio.

Zone Condition and Input Checks

These are organized sequentially according to the bar charts beginning on p. IV-9.

Mean IDB

All results are within 0.2°C of the setpoint except for TRNSYS-real for cases E100 – E165, E190 (within 0.3°C–0.6°C, for realistic controller).

(Max-Min)/Mean IDB

TRNSYS-real gives a variation ranging from 2% to 8%.

Table 4-1. Participating Organizations and Computer Programs

Model	Authoring Organization	Implemented By	Abbreviation
Analytical solution	Hochschule Technik & Architektur Luzern, Switzerland (HTAL)	Hochschule Technik & Architektur Luzern, Switzerland	HTAL1
Analytical solution with realistic controller model	Hochschule Technik & Architektur Luzern, Switzerland	Hochschule Technik & Architektur Luzern, Switzerland	HTAL2
Analytical Solution	Technische Universität Dresden, Germany (TUD)	Technische Universität Dresden, Germany	TUD
CA-SIS V1	Electricité de France, France (EDF)	Electricité de France, France	CA-SIS
CLIM2000 2.1.6	Electricité de France, France	Electricité de France, France	CLM2000
DOE-2.1E-088	LANL/LBNL/ESTSC,[a,b,c] USA	CIEMAT,[d] Spain	DOE21E/CIEMAT
DOE-2.1E-133	LANL/LBNL/JJH,[a,b,e] USA	NREL/JNA,[f] USA	DOE21E/NREL
ENERGYPLUS 1.0.0.023	LBNL/UIUC/CERL/OSU/GARD Analytics/FSEC/DOE-OBT,[a,g,h,i,j,k]	GARD Analytics, USA	Energy+
PROMETHEUS	Klimasystemtechnik, Germany (KST)	Klimasystemtechnik, Germany	Prometh
TRNSYS 14.2-TUD with ideal controller model	University of Wisconsin, USA; Technische Universität Dresden, Ger.	Technische Universität Dresden, Germany	TRN-id TRNSYS-ideal
TRNSYS 14.2-TUD with real controller model	University of Wisconsin, USA; Technische Universität Dresden, Ger.	Technische Universität Dresden, Germany	TRN-re TRNSYS-real

[a]LANL: Los Alamos National Laboratory
[b]LBNL: Lawrence Berkeley National Laboratory
[c]ESTSC: Energy Science and Technology Software Center (at Oak Ridge National Laboratory, USA)
[d]CIEMAT: Centro de Investigaciones Energeticas, Medioambientales y Tecnologicas
[e]JJH: James J. Hirsch & Associates
[f]NREL/JNA: National Renewable Energy Laboratory/J. Neymark & Associates
[g]UIUC: University of Illinois Urbana/Champaign
[h]CERL: U.S. Army Corps of Engineers, Construction Engineering Research Laboratories
[i]OSU: Oklahoma State University
[j]FSEC: University of Central Florida, Florida Solar Energy Center
[k]DOE-OBT: U.S. Department of Energy, Office of Building Technology, State and Community Programs, Energy Efficiency and Renewable Energy

Mean Indoor Humidity Ratio

Range of the simulations' disagreement with the analytical solutions:

- Wet coils indicate about 0%–9% range of disagreement.
- Dry coils indicate about 0%–3% range of disagreement.
- CLIM2000 has outlying value by 11% for E120 and 6% for E185 and E195.

The greatest disagreement appears to be in case E120 for CLIM2000, and PROMETHEUS. Because CASIS previously had a similar error and fixed it by improving their interpolation method, CLIM2000 and PROMETHEUS should be checked.

(Max-Min)/Mean Humidity Ratio

Generally steady:

- CLIM2000 has 2% variation for dry coils
- CASIS, DOE21E/NREL, ENERGYPLUS, and PROMETHEUS vary 1%–2% for wet coils
- TRNSYS-real has up to 4% variation (E180).

Total Zone Load

Zone loads mostly agree very closely for this near-adiabatic test cell with only internal gains. Differences that should be checked include:

- CLIM2000 is 1% below the mean for E200.
- PROMETHEUS' total zone load varies by 2% from the other results for cases E185 and E195 (see sensible zone load below).

Sensible Zone Load

CLIM2000 is 1% below the mean for E200.

Results are similar to total zone load. However, PROMETHEUS' sensible zone load varies from other results by 8% in E185 and 7% in E195—this difference should be investigated.

Latent Zone Load

No substantial disagreements (all results well within 1%). PROMETHEUS has very slight (0.5%) disagreement with the analytical solutions. That disagreement may be related to the sensible zone load disagreement.

Output Comparisons

These are organized sequentially according to the bar charts beginning on p. IV-13.

Mean Coefficient of Performance (COP)

The greatest variation appears to be for dry-coil cases E100, E110, E130, and E140. These cases indicate about 3%–10% variation between the minimum and maximum simulation results; PROMETHEUS generally has the greatest disagreement with the analytical verification results, with the greatest disagreement seen at low part load ratio (PLR).

Notable disagreements:

- For low PLR cases, DOE21E/CIEMAT has 5%–6% disagreement for dry coils (E130 and E140) and 3% disagreement for wet coils (E190 and E195). This is because their COP calculation obtains net refrigeration effect from total coil load less fan energy (as noted later, the fan energy and "fan heat" are not consistent in DOE-2.1E); a different (and better agreeing) result is obtained if net refrigeration effect is calculated from summing sensible and latent zone loads (as NREL did). This is further discussed in the modeler reports by NREL and CIEMAT (see Part III).
- PROMETHEUS E180–E195 (cases with high latent internal gains) have generally higher COP than the other results.

(Max-Mean)/Mean COP

The TRNSYS-real transient variation increases to up to 20% as PLR decreases (caused by the realistic controller).

Mean COP Sensitivities

Disagreements:

- PROMETHEUS has a number of disagreements where sensitivity is greater than the others (up to 0.34 COP or 10%): e.g., E190–E140, E120–110, etc. These should be checked.
- Disagreements between NREL and CIEMAT's DOE-2.1E results (e.g., E180–E150) could be related to the latent load discrepancy for NREL's results as discussed in NREL's and CIEMAT's modeler reports (see Part III).

Total Space Cooling Electricity Consumption

Disagreements:

- DOE21E/NREL results are 3% low in E170; half this difference was because COP Degradation Factor as a function of PLR (CDF f(PLR)) was not applied to indoor (ID) fan electricity.
- PROMETHEUS appears 5% low in E180.

Total Space Cooling Electricity Sensitivities

Disagreements are generally consistent with the COP sensitivity disagreements.

Compressor Electricity Consumption

- These results are consistent with the total electric consumption results.
- Disaggregated results for compressor electric consumption were not provided for ENERGYPLUS.

Compressor Electricity Sensitivities

- These results are consistent with the total electric consumption sensitivity results.
- Disaggregated results for compressor electric consumption were not provided for ENERGYPLUS.

Indoor (Supply) Fan Electricity Consumption

The y-axis scale is smaller versus that for compressor consumption, so percentage differences are less significant in terms of energy costs (for those cases where the fan cycled on/off with the compressor). However, the differences can still indicate disagreements that should be explained.

For DOE-2.1E results, the occurrence of relatively less indoor fan energy versus total energy compared to the other results happens because in DOE-2.1E the disaggregated indoor fan does not pick up the COP degradation factor (CDF) adjustment at part loads; the DOE2.1E/NREL compressor and outdoor fan results do account for the CDF adjustment.

The sources of differences between CIEMAT and NREL DOE-2.1E simulations (e.g., in E110) should also be isolated.

Indoor (Supply) Fan Electricity Sensitivity

The differences in specific results among various cases for each simulation program are consistent with the indoor fan consumption results described above.

Outdoor (Condenser) Fan Electricity Consumption

Note that the y-axis scale is again small, so percentage differences are less significant in terms of energy costs.

For E170 relative differences between compressor consumption versus outdoor (OD) fan consumption for DOE21E/CIEMAT results are not as consistent as for other results (compare “crowns” [tops] of graphs). This is because CDF was not applied to CIEMAT’s OD fan “hand” calculation in the DOE21E/CIEMAT runs (see CIEMAT’s modeler report in Part III). Disaggregated results for compressor electricity consumption were not provided for ENERGYPLUS.

Outdoor (Condenser) Fan Electricity Sensitivities

The differences in specific results among various cases for each simulation program are consistent with the outdoor fan consumption results described above. Disaggregated results for compressor electricity consumption were not provided for ENERGYPLUS.

Total Coil Load

Total coil loads exhibit generally good agreement. Notable loads differences are more apparent from the other loads charts described below.

Total Coil Load Sensitivities

Total coil loads sensitivities exhibit generally good agreement. Notable loads differences are more apparent from the other loads charts described below.

Sensible Coil Load

- DOE21E/NREL E180 and E185 seem slightly high; see below for “Sensible Coil Load Versus Sensible Zone Load (Fan Heat).”
- PROMETHEUS seems slightly high in E185.

Sensible Coil Load Sensitivities

The differences in specific results among various cases for each simulation program are consistent with the sensible coil load results described above.

Latent Coil Load

DOE21E/NREL, ENERGYPLUS, and PROMETHEUS are slightly low in E180 and E185.

Latent Coil Load Sensitivities

The differences in specific results among various cases for each simulation program are consistent with the sensible coil load results described above.

Sensible Coil Load Versus Sensible Zone Load (Fan Heat)

The largest disagreements are:

- DOE21E/CIEMAT is high for E100-E120, E150-E165, E180, E185, and E200.
- DOE21E/NREL is high for E180, E185, and E200.
- PROMETHEUS is low for E170.

Latent Coil Load versus Latent Envelope Load

Because there is no moisture diffusion for the envelope and the zone is at steady state, latent coil loads should match latent zone loads very closely.

Disagreements between latent coil and latent zone loads of >10 kWh occur for:

- DOE21E/NREL: E180, E185 (-30 kWh maximum difference)
- ENERGYPLUS: E180, E185, E200 (-13 kWh maximum difference).

Summary of Remaining Disagreements among the Simulation Programs

Comments about the final results of each simulation program are listed below. The disagreements are those remaining after numerous iterations of simulations (including bug fixes and input corrections) and test specification revisions. We have reported the remaining differences that did not have legitimate reasons to the appropriate code authors.

- CA-SIS/EDF
  - The modeler report indicates that the performance map was revised (extrapolated) manually to run the cases. This makes it difficult for a typical CA-SIS user to obtain the same results as the EDF development group has obtained for these test cases. Extrapolation of the given performance data should be automated within CA-SIS.
- CLIM2000/EDF
  - The indoor humidity ratios for E185 and E195 are outlying by 5%.
  - The zone sensible and total loads are 1% below the mean for E200.
- DOE21E/CIEMAT
  - The disaggregated indoor fan does not pick up the CDF adjustment.
  - The OD fan energy does not appear to pick up the CDF adjustment.
  - The difference between sensible coil load and sensible zone load (fan heat) is up to 37% higher throughout relative to this difference for the analytical solution results (this might be a bug in the software). This output magnifies an effect that does not appear to have much impact on the energy consumption for these test cases, but could be important for other simulations.
- DOE21E/NREL

- The disaggregated indoor fan does not pick up the CDF adjustment.
- The difference between the sensible coil and zone load (fan heat) is high in E180 and E185.
- The latent coil load is slightly (1%) different from the latent zone load in E180 and E185.
- The differences in results versus DOE21E/CIEMAT are attributable to differences between DOE-2.1E's RESYS2 (used by NREL) and PTAC (used by CIEMAT) system models (e.g., PTAC requires a blow-through ID fan), and because NREL and CIEMAT used different versions (from different suppliers) of the software.

- ENERGYPLUS
  - Latent coil load is slightly (<1%) different from latent zone load in wet-coil cases.
- PROMETHEUS/KST

KST was not able to complete refinement of their simulation results for this project because they could not participate in the task beyond January 2000.

  - A number of COP sensitivity disagreements (E180–E150, E190–E140, E120–E110, etc) should be checked.
  - Total electric consumption is 5% lower than the others in E180.
  - The sensible zone load varies by 8% in E185 (7% in E195).
  - The current results may not enable extrapolation.
  - Performance map interpolation techniques for E120, and possible improper use of dry coil sensible capacity data (based on E120 humidity ratio result) should be checked.
  - The fan heat is low in E170.
- TRNSYS-ideal/TUD
  - No apparent disagreements.
- TRNSYS-real/TUD
  - The following disagreements are acceptable and are caused by using a realistic controller model with 36-s (0.01-h) timesteps:
    - Transient variations of IDB of 0.5°C to 2.1°C (except for E200)
    - Mean IDB is 0.3°–0.6°C from set point in some cases
    - Greater transient variations of COP than the other simulations. This does not cause disagreements for *mean* COP, consumption, and loads. Greatest transient variation is at low PLR.

Summary Ranges of Simulation Results Disagreements

As shown in Table 4-2, the mean results of COP and total energy consumption for the programs are on average within <1% of the analytical solution results, with variations up to 2% for the low PLR dry coil cases (E130 and E140). Ranges of disagreement are further tabulated below. This summary excludes results for PROMETHEUS; KST suspected error(s) in its software but was unable to correct them or complete this project.

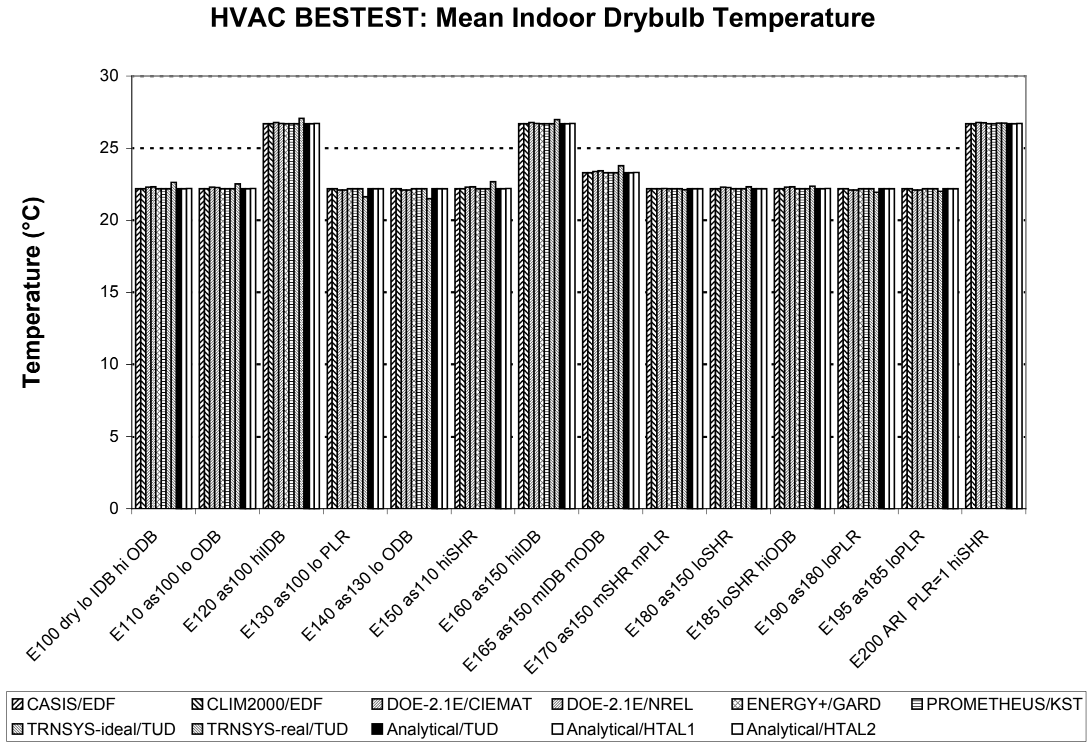
Table 4-2. Ranges of Disagreement versus Analytical Solutions

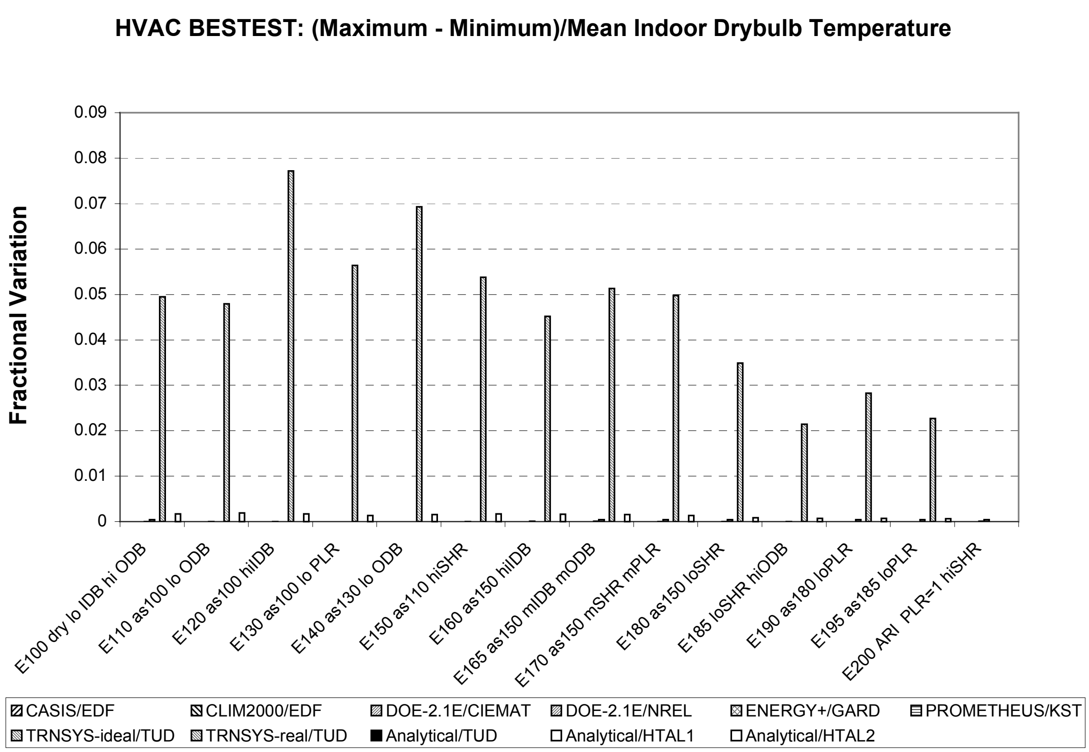
Cases	Dry coil	Wet coil
Energy	0%–6%[a]	0%–3%[a]
COP	0%–6%[a]	0%–3%[a]
Humidity Ratio	0%–11%[a]	0%–7%[a]
Zone Temperature	0.0°C–0.7°C (0.1°C)[b]	0.0°C–0.5°C (0.0°C–0.1°C)[b]

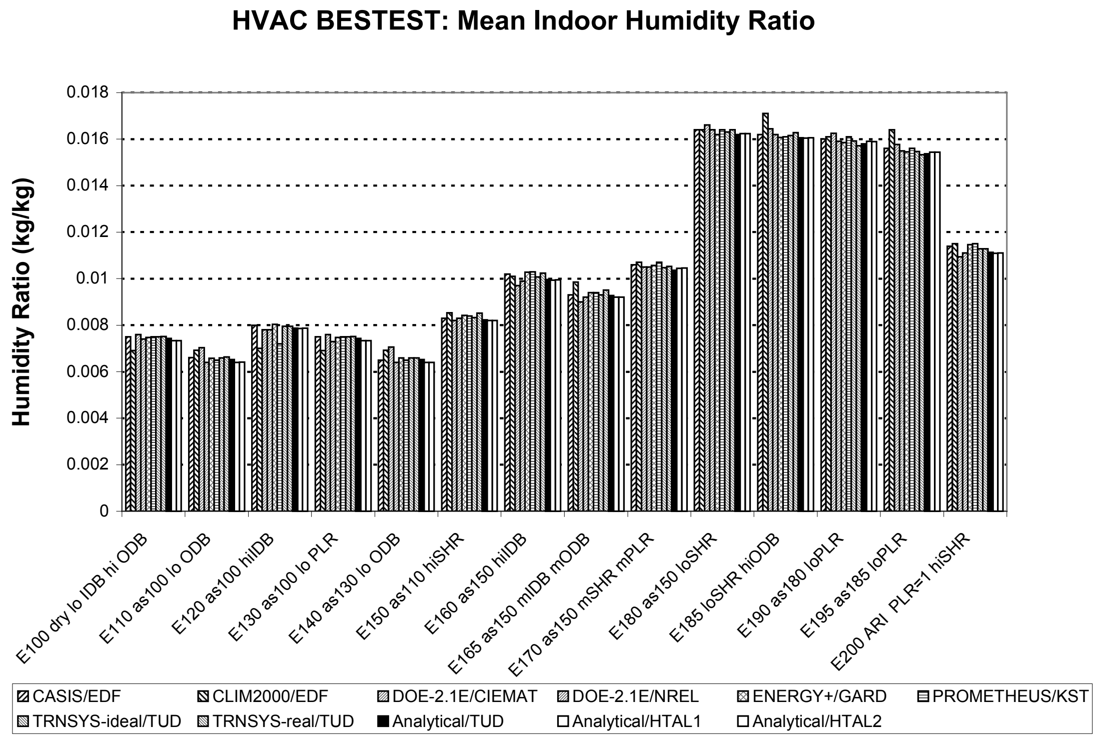
[a] % = (ABS(Sim - AnSoln))/AnSoln × 100%; sim = each simulation result, AnSoln = avg(TUD,HTAL1).

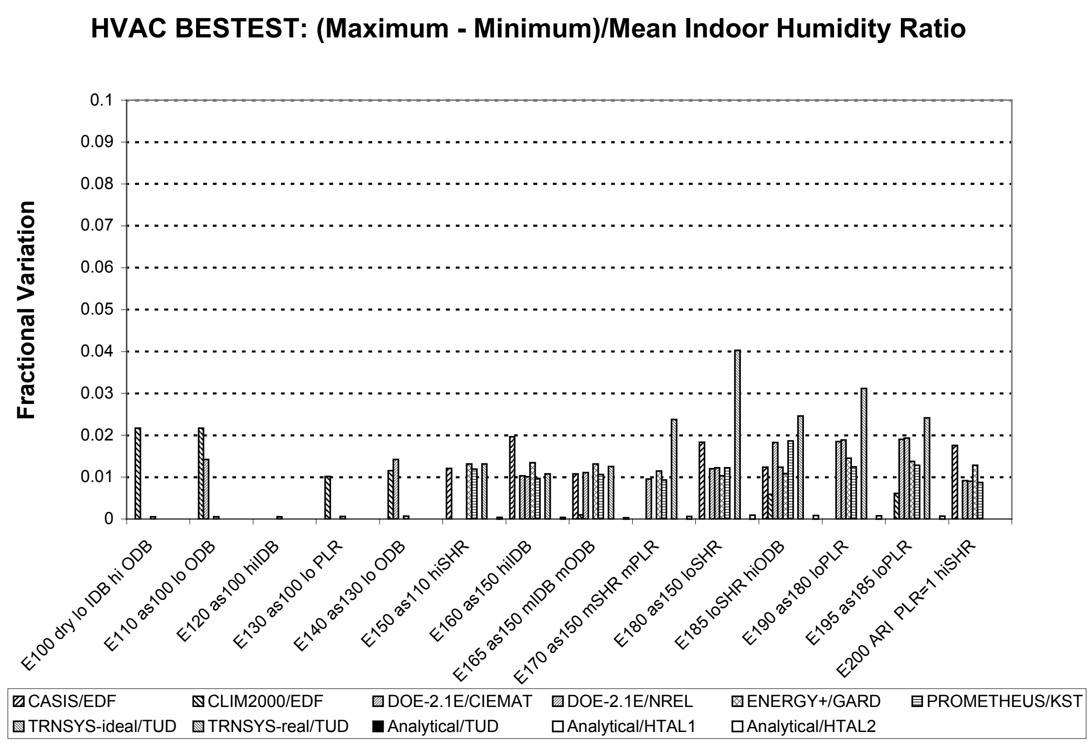
[b] Excludes results for TRNSYS-TUD with realistic controller.

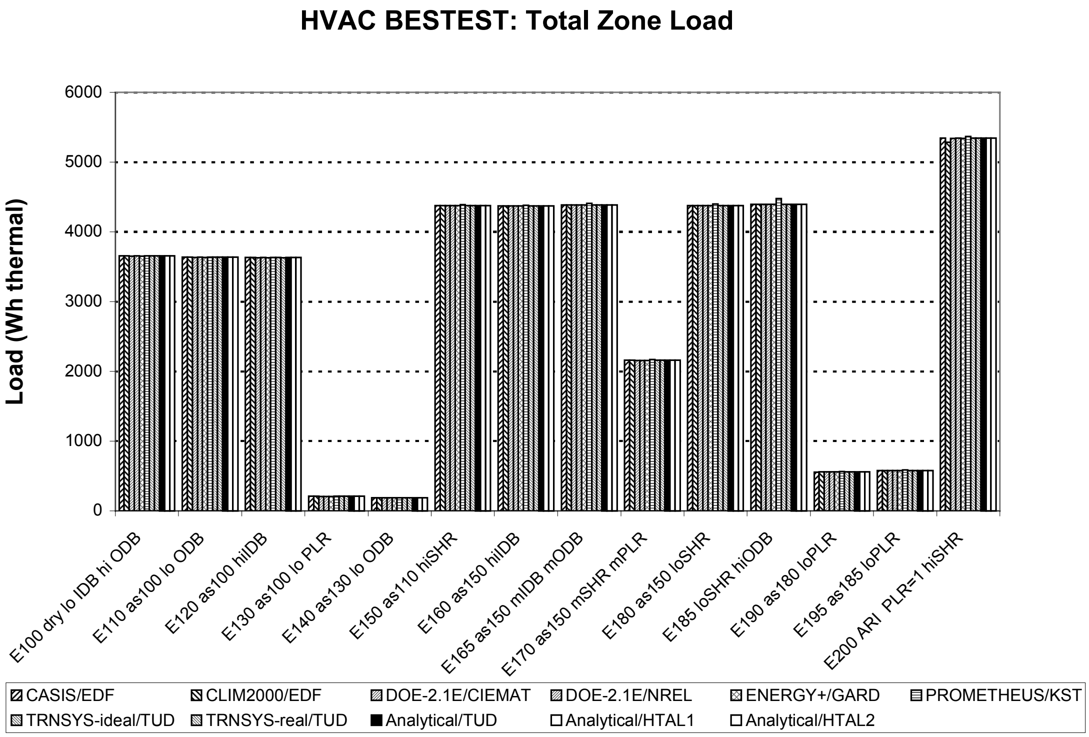
The higher level of disagreement in the dry-coil cases occurs for the case with lowest PLR and is related to some potential problems that have been documented for DOE-2.1E (ESTSC version 088 and JJH version 133) in both the CIEMAT and NREL results (see Part III). The larger disagreements for zone humidity ratio are caused by disagreements for the CLIM2000 and DOE21E/CIEMAT results (CIEMAT used a PTAC, which only allows for a blow-through fan rather than the draw-through fan of the test specification). The disagreement in zone temperature results is primarily from the TRNSYS-TUD simulation results, where a realistic controller was applied on a short timestep (36 s); all other simulation results apply ideal control.

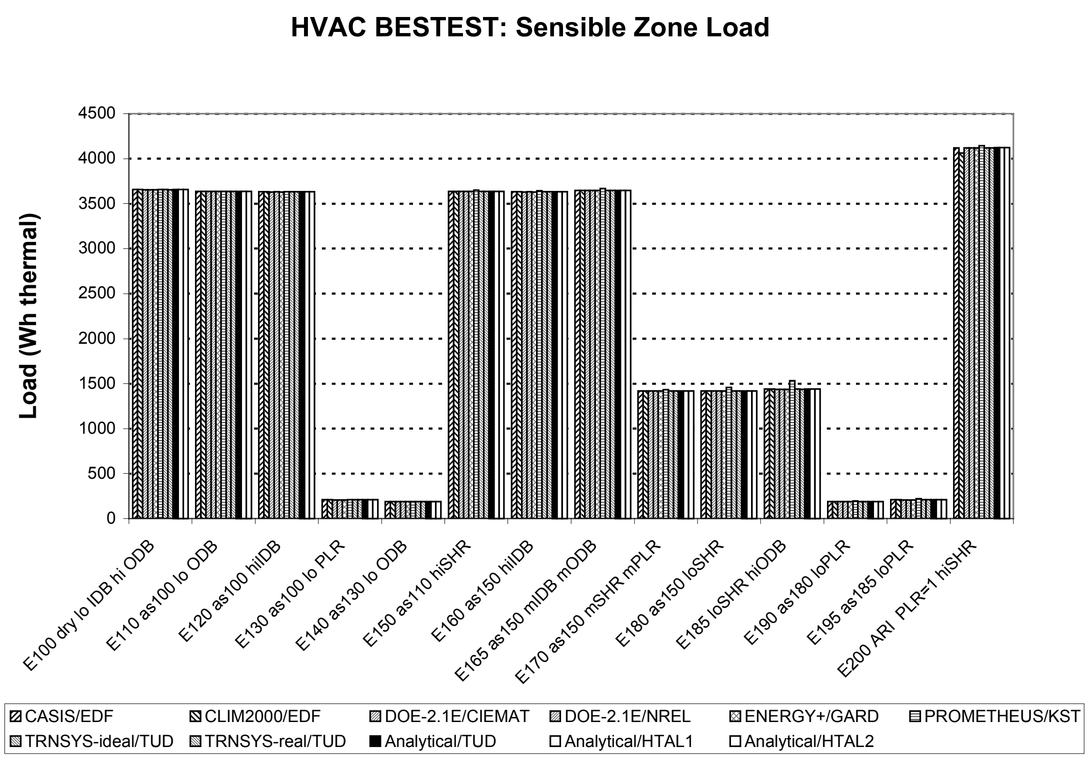

HVAC BESTEST: Mean Indoor Drybulb Temperature
Temperature (°C)
30
25
20
15
10
5
0
E100 dry lo IDB hi ODB
E110 as100 lo ODB
E120 as100 hiIDB
E130 as100 lo PLR
E140 as130 lo ODB
E150 as110 hiSHR
E160 as150 hiIDB
E165 as150 mIDB mODB
E170 as150 mSHR mPLR
E180 as150 loSHR
E185 loSHR hiODB
E190 as180 loPLR
E195 as185 loPLR
E200 ARI PLR=1 hiSHR
CASIS/EDF
CLIM2000/EDF
DOE-2.1E/CIEMAT
DOE-2.1E/NREL
ENERGY+/GARD
PROMETHEUS/KST
TRNSYS-ideal/TUD
TRNSYS-real/TUD
Analytical/TUD
Analytical/HTAL1
Analytical/HTAL2


HVAC BESTEST: (Maximum - Minimum)/Mean Indoor Drybulb Temperature
Fractional Variation
0.09
0.08
0.07
0.06
0.05
0.04
0.03
0.02
0.01
0
E100 dry lo IDB hi ODB
E110 as100 lo ODB
E120 as100 hiIDB
E130 as100 lo PLR
E140 as130 lo ODB
E150 as110 hiSHR
E160 as150 hiIDB
E165 as150 mIDB mODB
E170 as150 mSHR mPLR
E180 as150 loSHR
E185 loSHR hiODB
E190 as180 loPLR
E195 as185 loPLR
E200 ARI PLR=1 hiSHR
CASIS/EDF
CLIM2000/EDF
DOE-2.1E/CIEMAT
DOE-2.1E/NREL
ENERGY+/GARD
PROMETHEUS/KST
TRNSYS-ideal/TUD
TRNSYS-real/TUD
Analytical/TUD
Analytical/HTAL1
Analytical/HTAL2

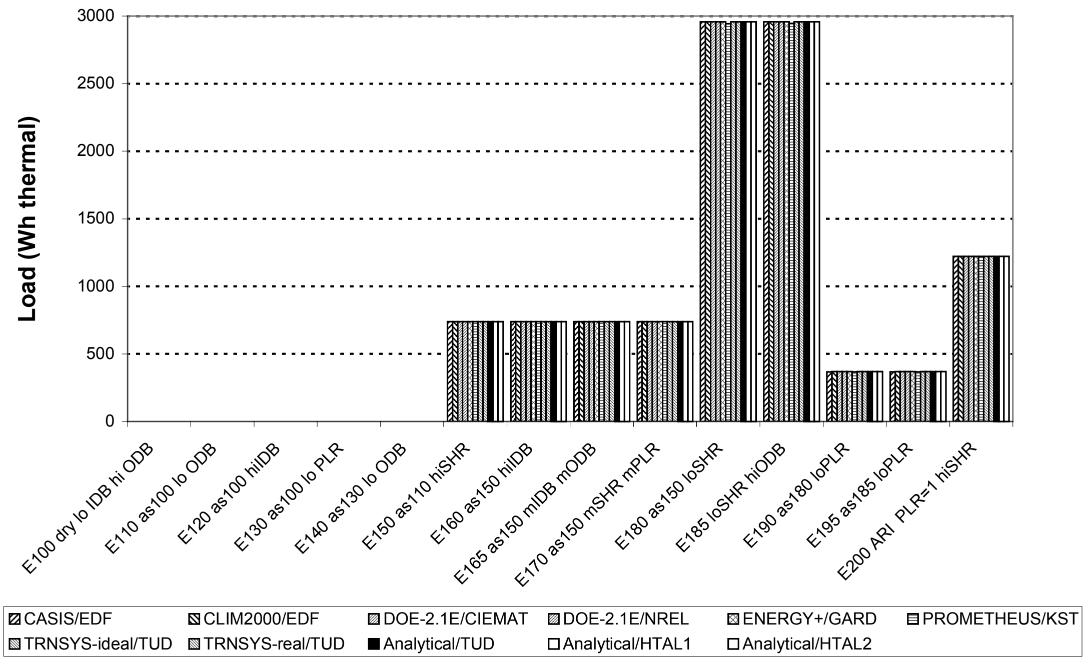

HVAC BESTEST: Mean Indoor Humidity Ratio
Humidity Ratio (kg/kg)
0.018
0.016
0.014
0.012
0.01
0.008
0.006
0.004
0.002
0
E100 dry lo IDB hi ODB
E110 as100 lo ODB
E120 as100 hiIDB
E130 as100 lo PLR
E140 as130 lo ODB
E150 as110 hiSHR
E160 as150 hiIDB
E165 as150 mIDB mODB
E170 as150 mSHR mPLR
E180 as150 loSHR
E185 loSHR hiODB
E190 as180 loPLR
E195 as185 loPLR
E200 ARI PLR=1 hiSHR
CASIS/EDF
CLIM2000/EDF
DOE-2.1E/CIEMAT
DOE-2.1E/NREL
ENERGY+/GARD
PROMETHEUS/KST
TRNSYS-ideal/TUD
TRNSYS-real/TUD
Analytical/TUD
Analytical/HTAL1
Analytical/HTAL2

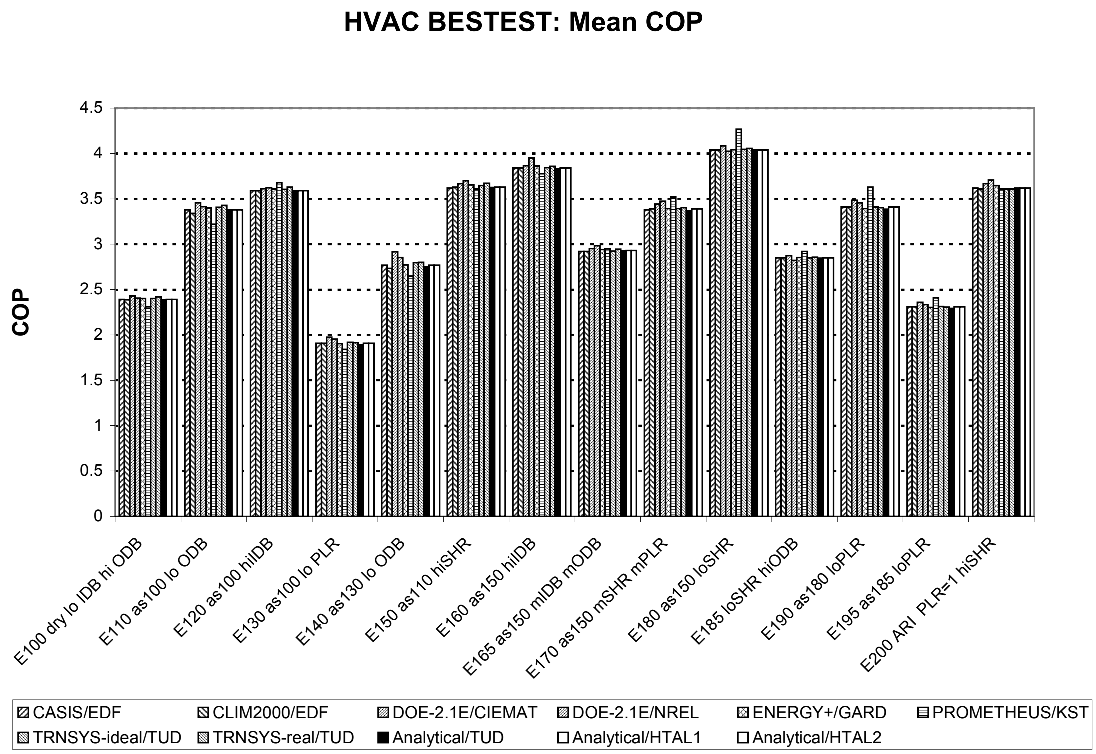

HVAC BESTEST: (Maximum - Minimum)/Mean Indoor Humidity Ratio
Fractional Variation
0.1
0.09
0.08
0.07
0.06
0.05
0.04
0.03
0.02
0.01
0
E100 dry lo IDB hi ODB
E110 as100 lo ODB
E120 as100 hiIDB
E130 as100 lo PLR
E140 as130 lo ODB
E150 as110 hiSHR
E160 as150 hiIDB
E165 as150 mIDB mODB
E170 as150 mSHR mPLR
E180 as150 loSHR
E185 loSHR hiODB
E190 as180 loPLR
E195 as185 loPLR
E200 ARI PLR=1 hiSHR
CASIS/EDF
CLIM2000/EDF
DOE-2.1E/CIEMAT
DOE-2.1E/NREL
ENERGY+/GARD
PROMETHEUS/KST
TRNSYS-ideal/TUD
TRNSYS-real/TUD
Analytical/TUD
Analytical/HTAL1
Analytical/HTAL2

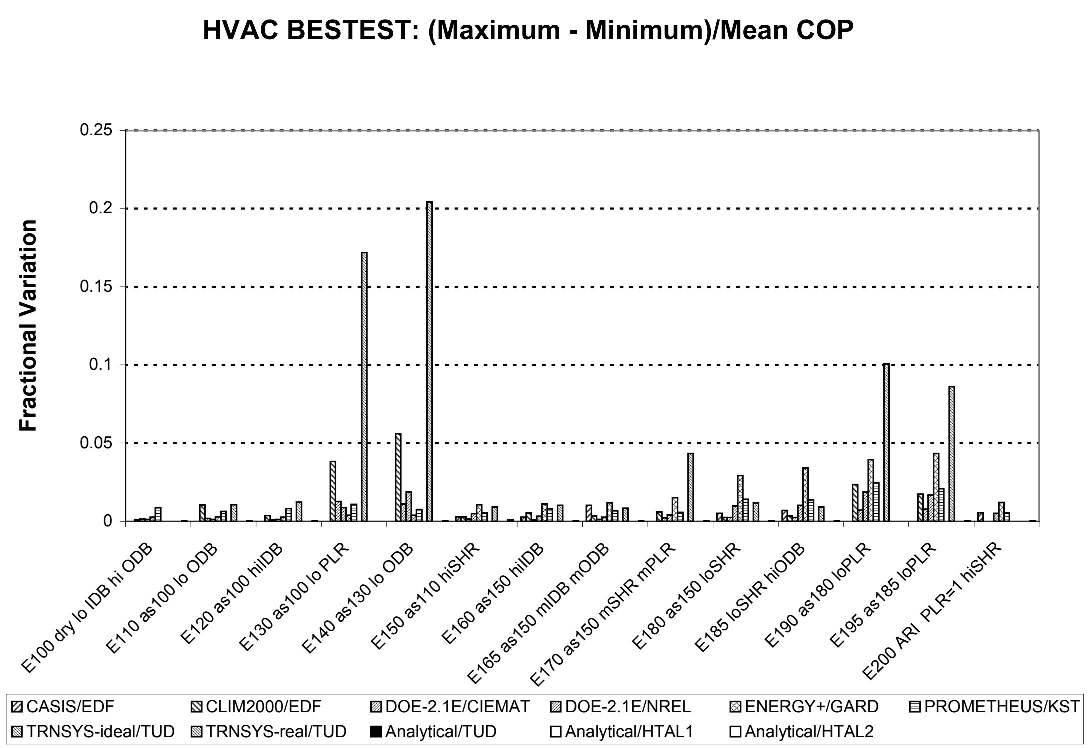

HVAC BESTEST: Total Zone Load
Load (Wh thermal)
6000
5000
4000
3000
2000
1000
0
E100 dry lo IDB hi ODB
E110 as100 lo ODB
E120 as100 hiIDB
E130 as100 lo PLR
E140 as130 lo ODB
E150 as110 hiSHR
E160 as150 hiIDB
E165 as150 mIDB mODB
E170 as150 mSHR mPLR
E180 as150 loSHR
E185 loSHR hiODB
E190 as180 loPLR
E195 as185 loPLR
E200 ARI PLR=1 hiSHR
CASIS/EDF
CLIM2000/EDF
DOE-2.1E/CIEMAT
DOE-2.1E/NREL
ENERGY+/GARD
PROMETHEUS/KST
TRNSYS-ideal/TUD
TRNSYS-real/TUD
Analytical/TUD
Analytical/HTAL1
Analytical/HTAL2

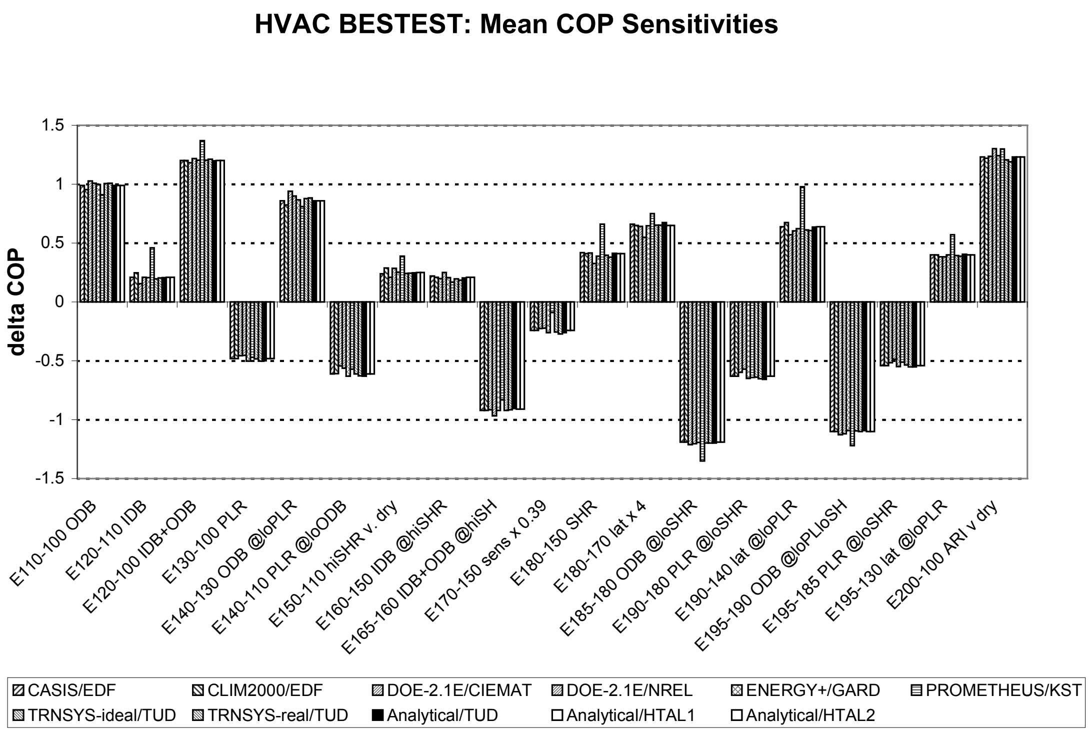

HVAC BESTEST: Sensible Zone Load
Load (Wh thermal)
4500
4000
3500
3000
2500
2000
1500
1000
500
0
E100 dry lo IDB hi ODB
E110 as100 lo ODB
E120 as100 hiIDB
E130 as100 lo PLR
E140 as130 lo ODB
E150 as110 hiSHR
E160 as150 hiIDB
E165 as150 mIDB mODB
E170 as150 mSHR mPLR
E180 as150 loSHR
E185 loSHR hiODB
E190 as180 loPLR
E195 as185 loPLR
E200 ARI PLR=1 hiSHR
CASIS/EDF
CLIM2000/EDF
DOE-2.1E/CIEMAT
DOE-2.1E/NREL
ENERGY+/GARD
PROMETHEUS/KST
TRNSYS-ideal/TUD
TRNSYS-real/TUD
Analytical/TUD
Analytical/HTAL1
Analytical/HTAL2

HVAC BESTEST: Latent Zone Load

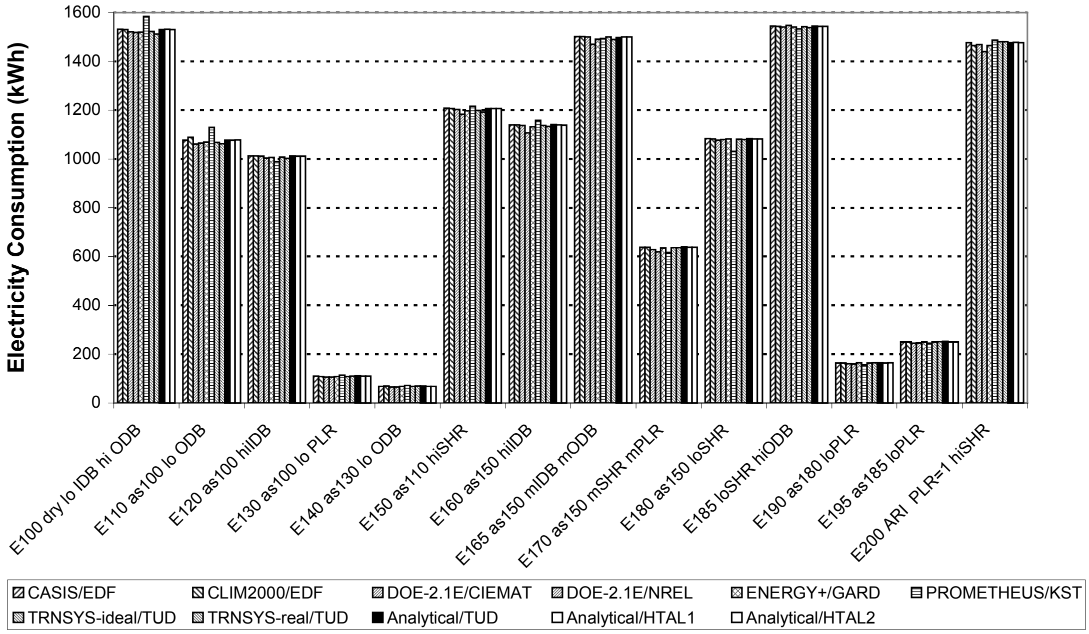



HVAC BESTEST: Mean COP
COP
4.5
4
3.5
3
2.5
2
1.5
1
0.5
0
E100 dry lo IDB hi ODB
E110 as100 lo ODB
E120 as100 hiIDB
E130 as100 lo PLR
E140 as130 lo ODB
E150 as110 hiSHR
E160 as150 hiIDB
E165 as150 mIDB mODB
E170 as150 mSHR mPLR
E180 as150 loSHR
E185 loSHR hiODB
E190 as180 loPLR
E195 as185 loPLR
E200 ARI PLR=1 hiSHR
CASIS/EDF
CLIM2000/EDF
DOE-2.1E/CIEMAT
DOE-2.1E/NREL
ENERGY+/GARD
PROMETHEUS/KST
TRNSYS-ideal/TUD
TRNSYS-real/TUD
Analytical/TUD
Analytical/HTAL1
Analytical/HTAL2

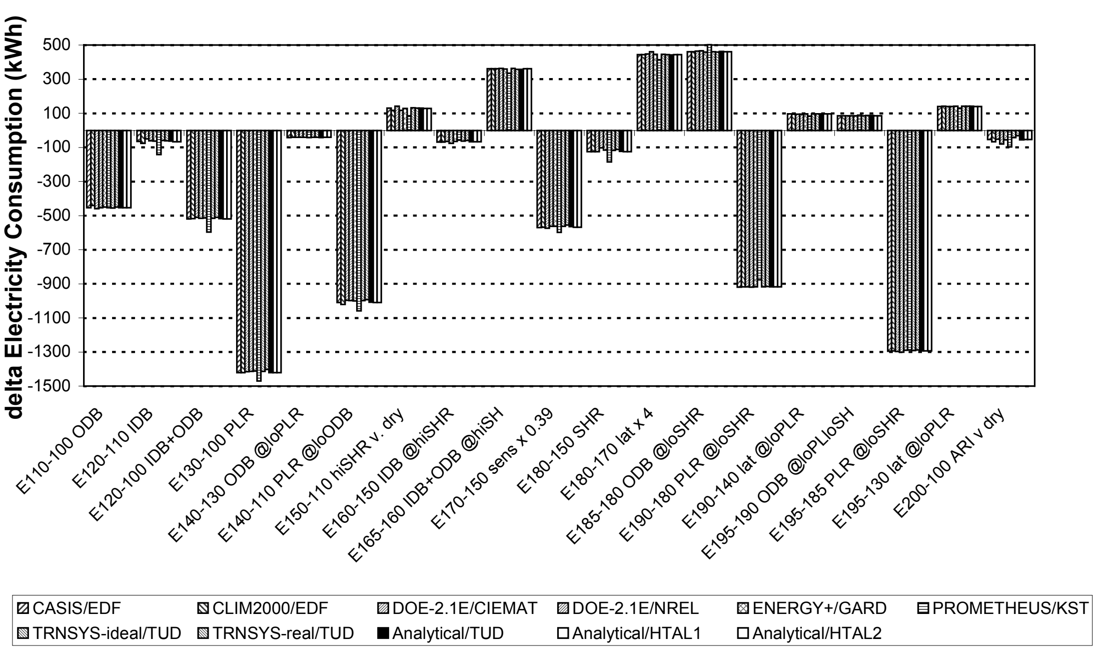

HVAC BESTEST: (Maximum - Minimum)/Mean COP
Fractional Variation
0.25
0.2
0.15
0.1
0.05
0
E100 dry lo IDB hi ODB
E110 as100 lo ODB
E120 as100 hiIDB
E130 as100 lo PLR
E140 as130 lo ODB
E150 as110 hiSHR
E160 as150 hiIDB
E165 as150 mIDB mODB
E170 as150 mSHR mPLR
E180 as150 loSHR
E185 loSHR hiODB
E190 as180 loPLR
E195 as185 loPLR
E200 ARI PLR=1 hiSHR
CASIS/EDF
CLIM2000/EDF
DOE-2.1E/CIEMAT
DOE-2.1E/NREL
ENERGY+/GARD
PROMETHEUS/KST
TRNSYS-ideal/TUD
TRNSYS-real/TUD
Analytical/TUD
Analytical/HTAL1
Analytical/HTAL2


HVAC BESTEST: Mean COP Sensitivities
delta COP
1.5
1
0.5
0
-0.5
-1
-1.5
E110-100 ODB
E120-110 IDB
E120-100 IDB+ODB
E130-100 PLR
E140-130 ODB @loPLR
E140-110 PLR @loODB
E150-110 hiSHR v. dry
E160-150 IDB @hiSHR
E165-160 IDB+ODB @hiSH
E170-150 sens x 0.39
E180-150 SHR
E180-170 lat x 4
E185-180 ODB @loSHR
E190-180 PLR @loSHR
E190-140 lat @loPLR
E195-190 ODB @loPLloSH
E195-185 PLR @loSHR
E195-130 lat @loPLR
E200-100 ARI v dry
CASIS/EDF
CLIM2000/EDF
DOE-2.1E/CIEMAT
DOE-2.1E/NREL
ENERGY+/GARD
PROMETHEUS/KST
TRNSYS-ideal/TUD
TRNSYS-real/TUD
Analytical/TUD
Analytical/HTAL1
Analytical/HTAL2

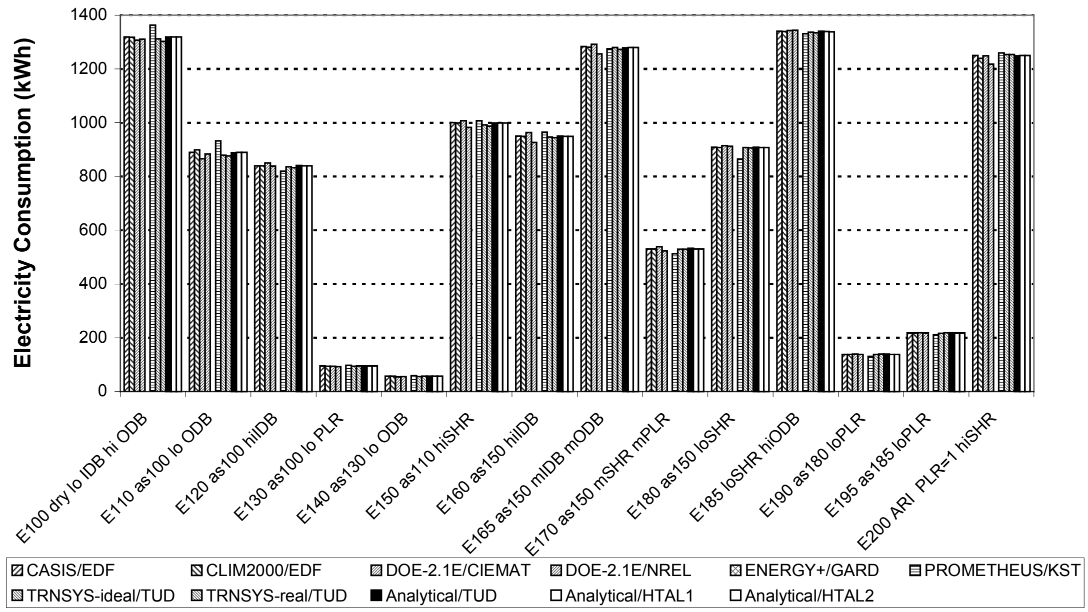
HVAC BESTEST: Total Space Cooling Electricity Consumption



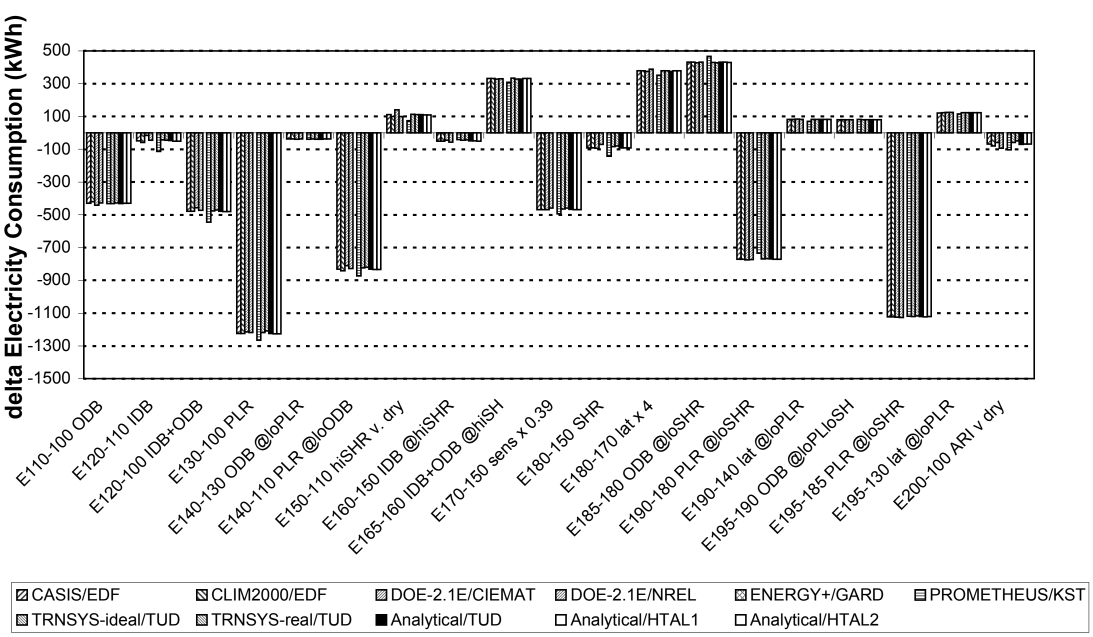
HVAC BESTEST: Total Space Cooling Electricity Sensitivities

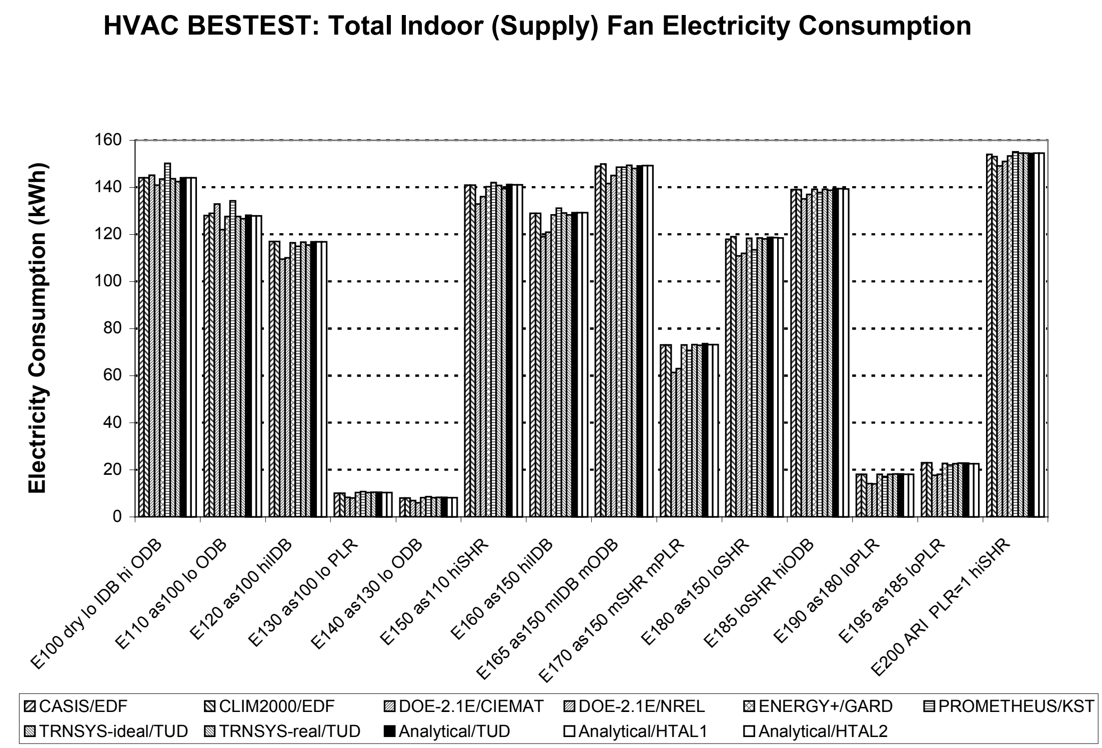


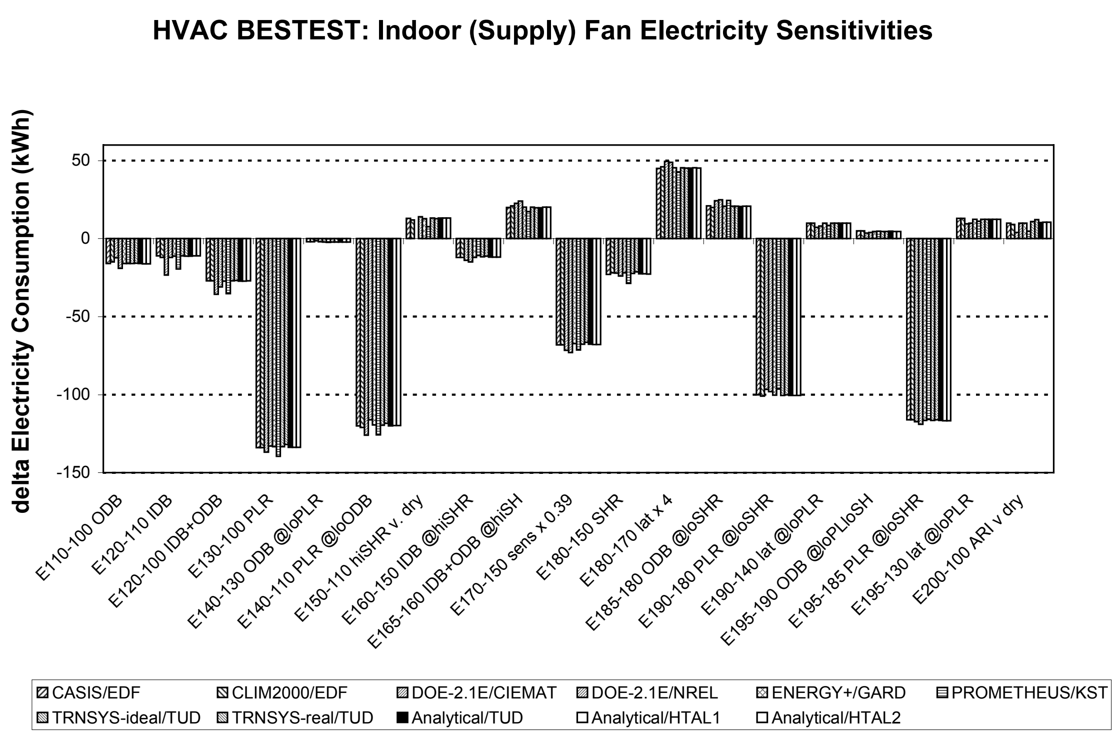
HVAC BESTEST: Compressor Electricity Consumption



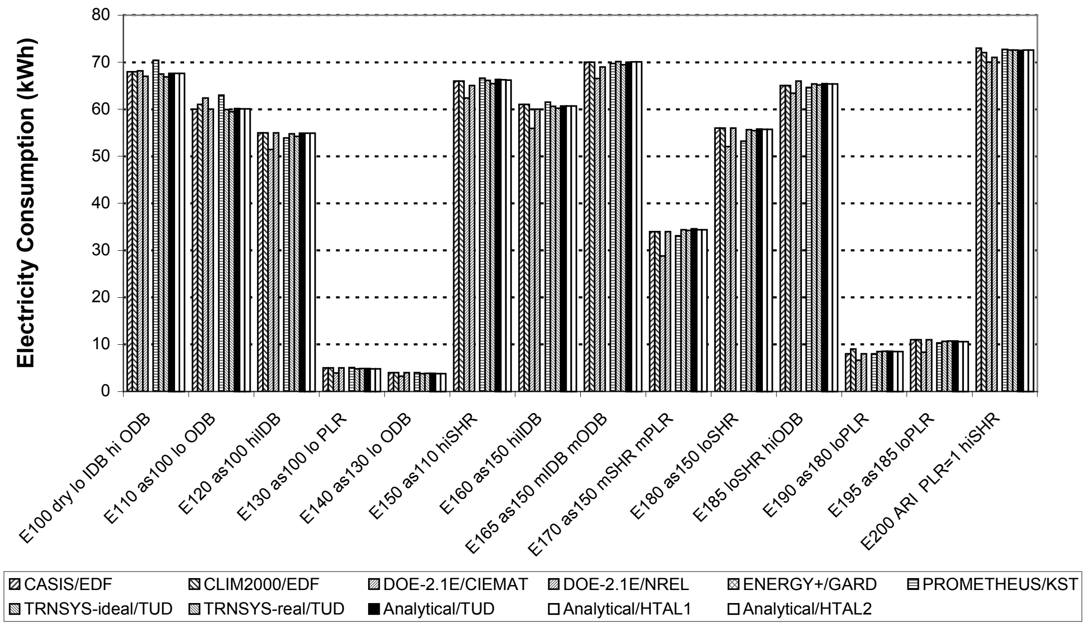
HVAC BESTEST: Total Compressor Electricity Sensitivities



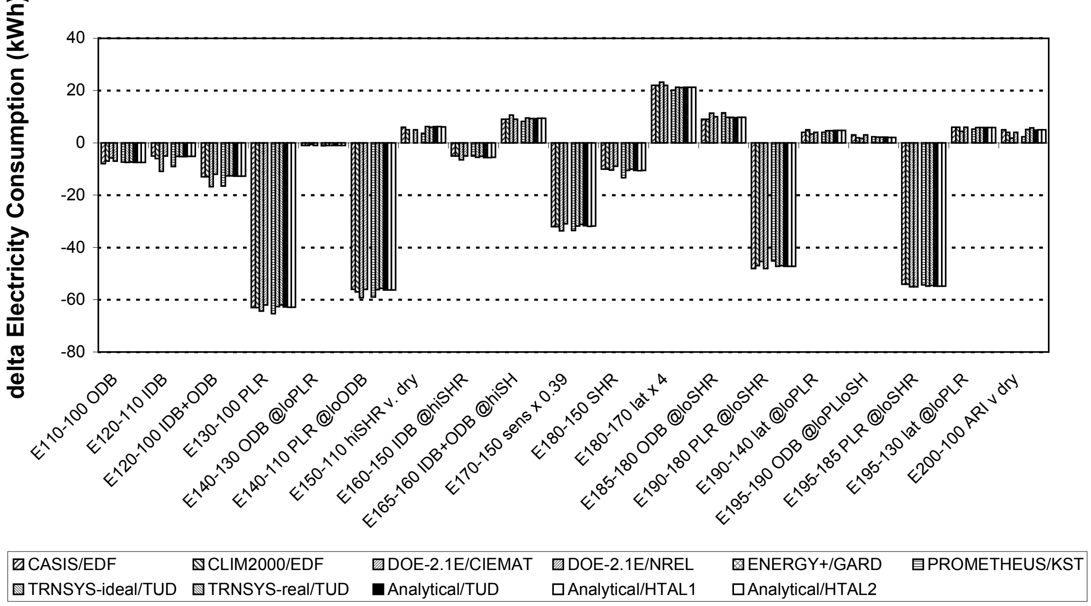

HVAC BESTEST: Total Indoor (Supply) Fan Electricity Consumption
Electricity Consumption (kWh)
160
140
120
100
80
60
40
20
0
E100 dry lo IDB hi ODB
E110 as100 lo ODB
E120 as100 hiIDB
E130 as100 lo PLR
E140 as130 lo ODB
E150 as110 hiSHR
E160 as150 hiIDB
E165 as150 mIDB mODB
E170 as150 mSHR mPLR
E180 as150 loSHR
E185 loSHR hiODB
E190 as180 loPLR
E195 as185 loPLR
E200 ARI PLR=1 hiSHR
CASIS/EDF
CLIM2000/EDF
DOE-2.1E/CIEMAT
DOE-2.1E/NREL
ENERGY+/GARD
PROMETHEUS/KST
TRNSYS-ideal/TUD
TRNSYS-real/TUD
Analytical/TUD
Analytical/HTAL1
Analytical/HTAL2


HVAC BESTEST: Indoor (Supply) Fan Electricity Sensitivities
delta Electricity Consumption (kWh)
50
0
-50
-100
-150
E110-100 ODB
E120-110 IDB
E120-100 IDB+ODB
E130-100 PLR
E140-130 ODB @loPLR
E140-110 PLR @loODB
E150-110 hiSHR v. dry
E160-150 IDB @hiSHR
E165-160 IDB+ODB @hiSH
E170-150 sens x 0.39
E180-150 SHR
E180-170 lat x 4
E185-180 ODB @loSHR
E190-180 PLR @loSHR
E190-140 lat @loPLR
E195-190 ODB @loPLloSH
E195-185 PLR @loSHR
E195-130 lat @loPLR
E200-100 ARI v dry
CASIS/EDF
CLIM2000/EDF
DOE-2.1E/CIEMAT
DOE-2.1E/NREL
ENERGY+/GARD
PROMETHEUS/KST
TRNSYS-ideal/TUD
TRNSYS-real/TUD
Analytical/TUD
Analytical/HTAL1
Analytical/HTAL2

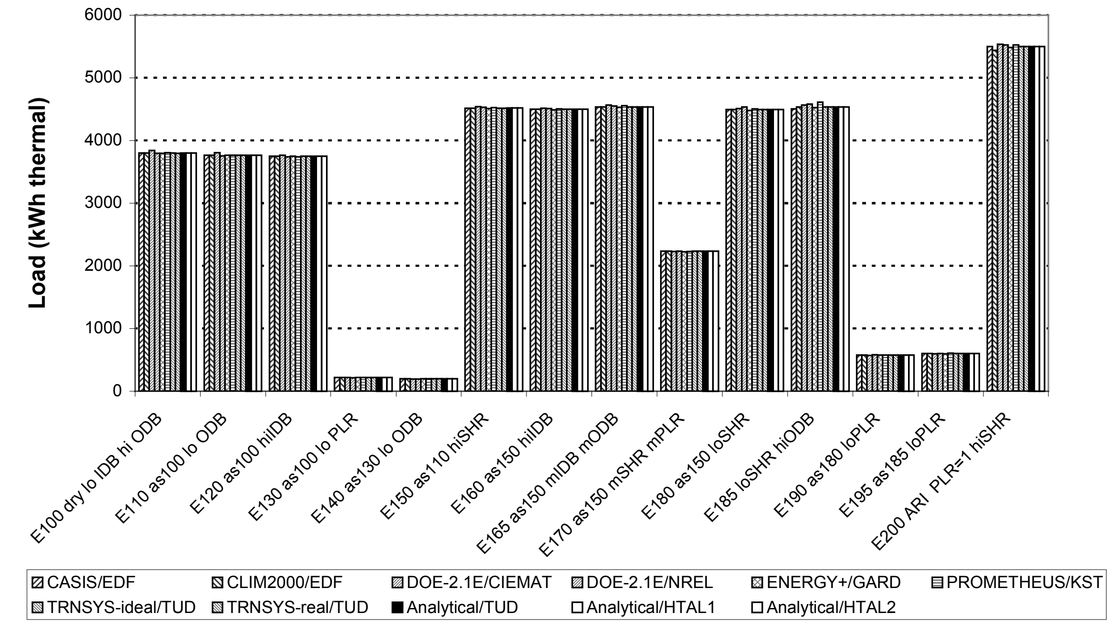
HVAC BESTEST: Outdoor (Condenser) Fan Electricity Consumption



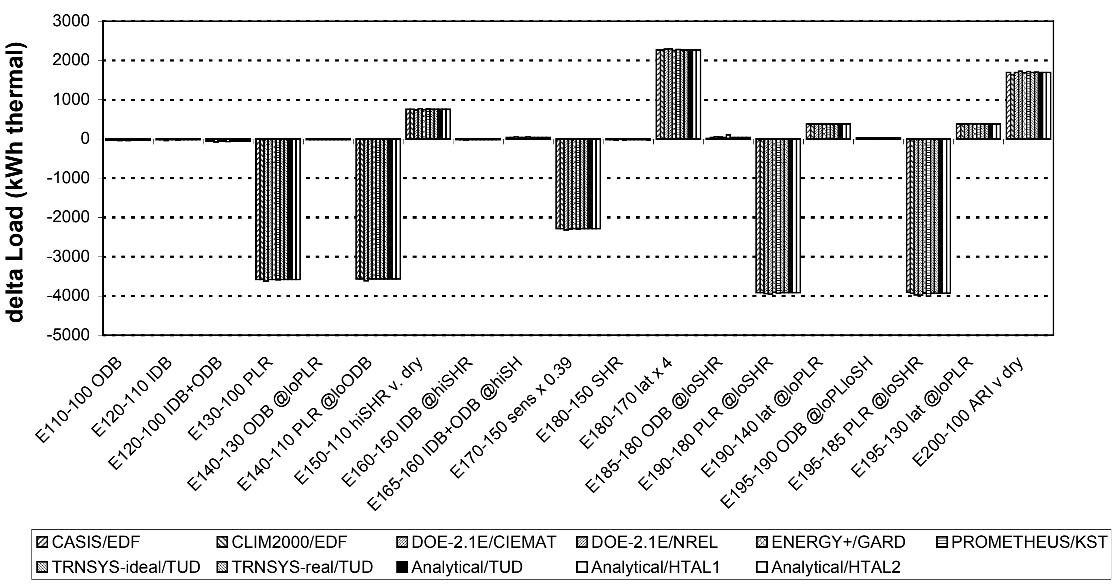
HVAC BESTEST: Outdoor (Condenser) Fan Electricity Sensitivities



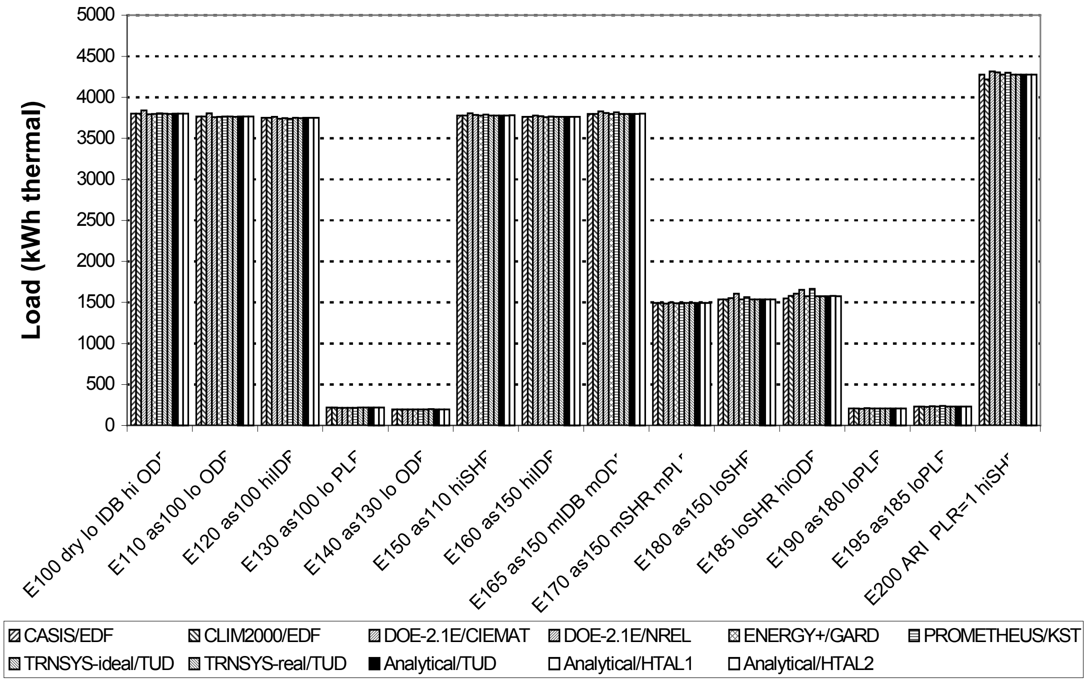
HVAC BESTEST: Total Coil Load



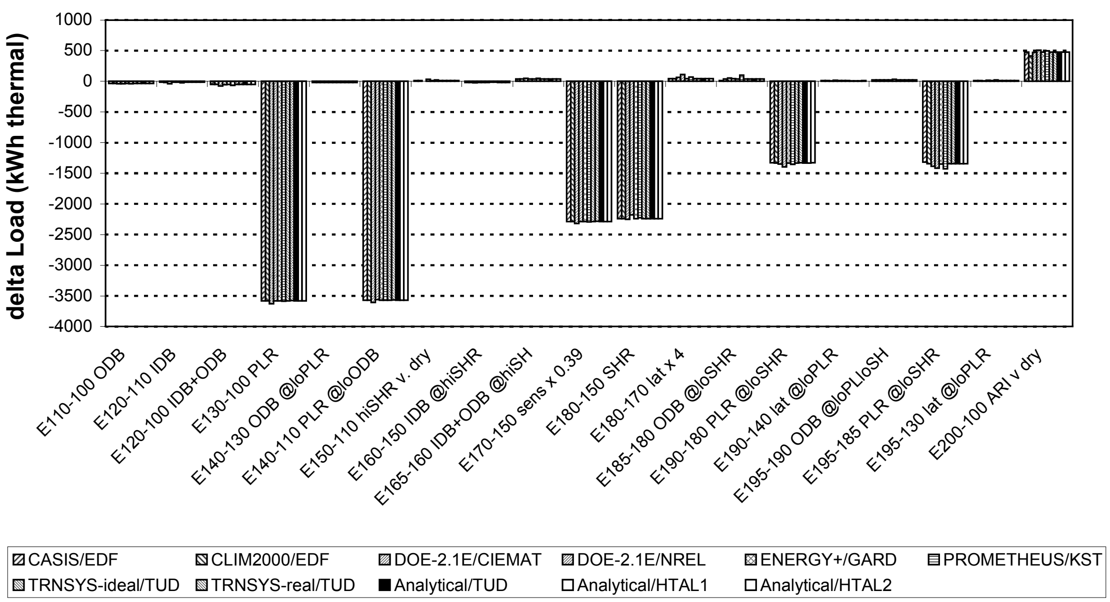
HVAC BESTEST: Total Coil Load Sensitivities



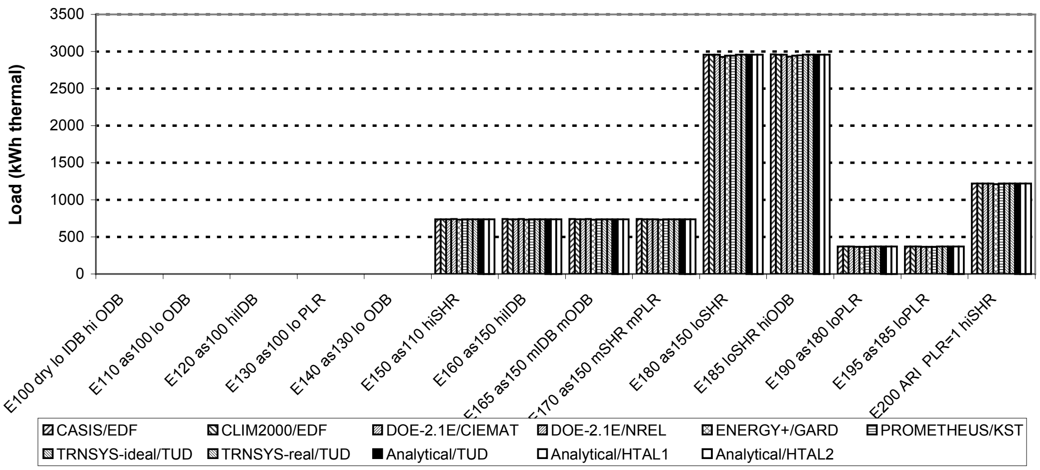
HVAC BESTEST: Sensible Coil Load



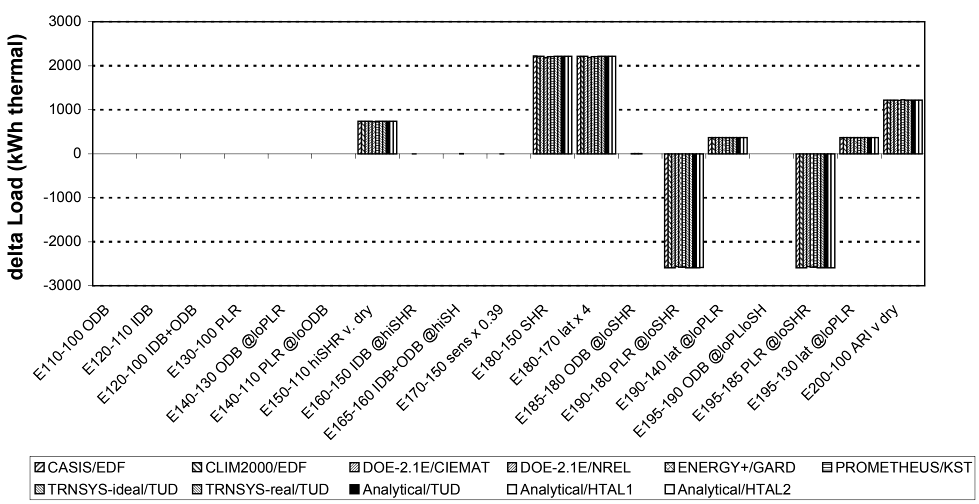
HVAC BESTEST: Sensible Coil Load Sensitivities



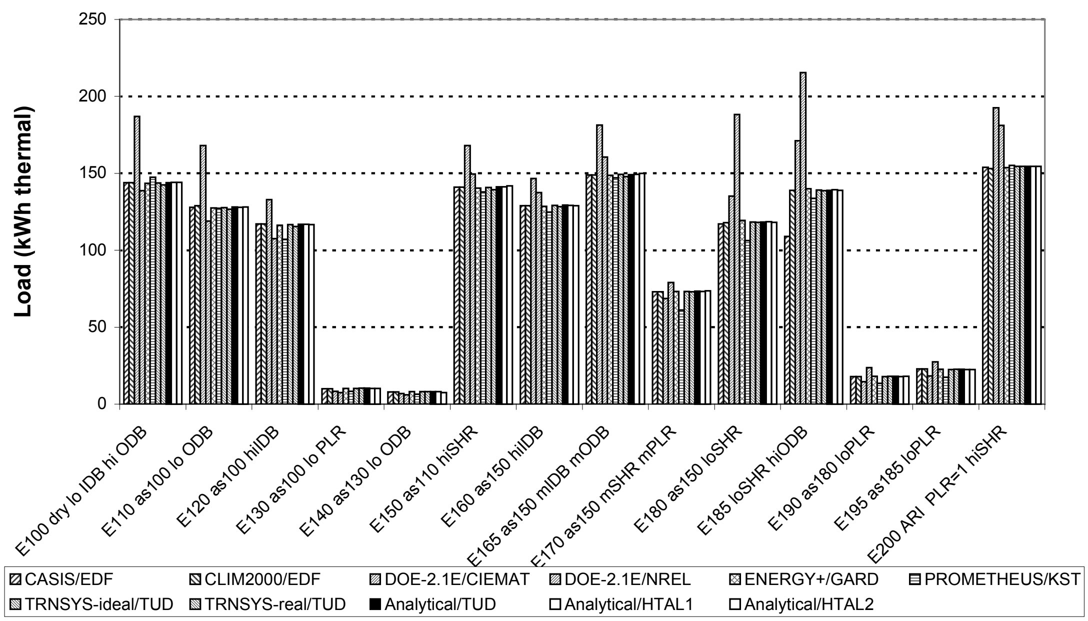
HVAC BESTEST: Latent Coil Load



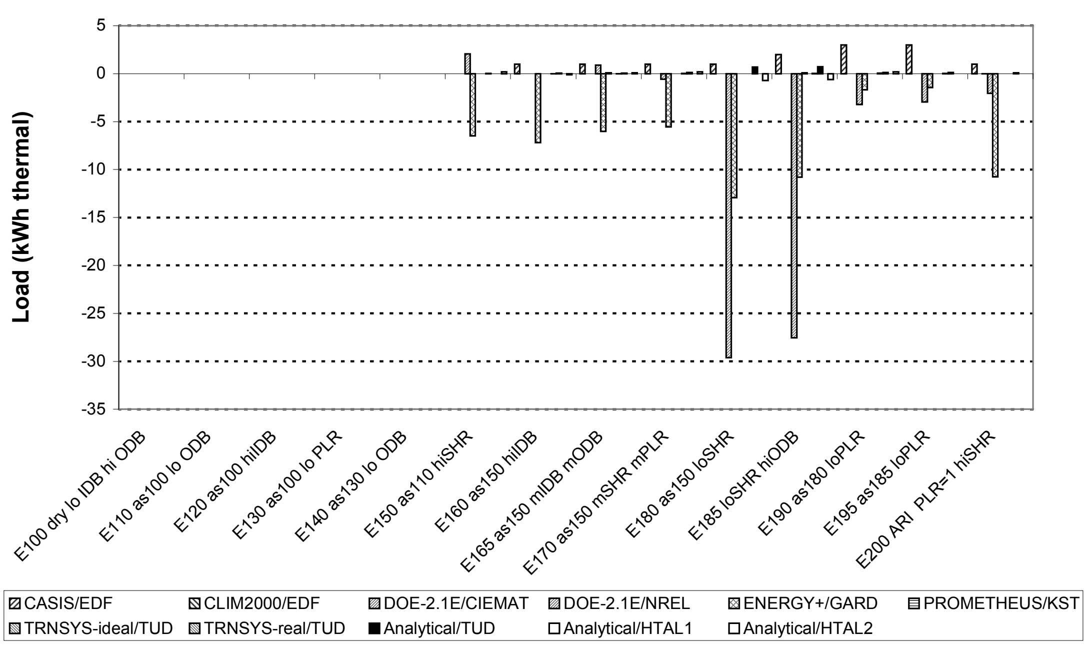
HVAC BESTEST: Latent Coil Load Sensitivities



HVAC BESTEST: Sensible Coil Load - Zone Load (Fan Heat)



HVAC BESTEST: Latent Coil Load - Latent Zone Load (Should = 0)



Space Cooling Electricity Consumption

Energy Consumption, Total (kWh,e)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) / Analytical	Analytical TUD	HTAL1	HTAL2
E100	1531	1530	1521	1519	1520	1584	1522	1512	1512	1584	4.7%	1531	1531	1531
E110	1077	1089	1061	1065	1069	1130	1067	1062	1061	1130	6.3%	1076	1077	1077
E120	1012	1012	1011	1003	1006	988	1007	1002	988	1012	2.4%	1013	1011	1011
E130	110	109	105	106	109	114	109	110	105	114	7.5%	111	110	110
E140	68	69	65	66	68	72	68	69	65	72	9.7%	69	69	68
E150	1208	1207	1202	1183	1197	1215	1199	1192	1183	1215	2.7%	1206	1207	1207
E160	1140	1139	1138	1107	1132	1158	1137	1133	1107	1158	4.5%	1140	1139	1139
E165	1502	1501	1499	1470	1491	1493	1500	1490	1470	1502	2.1%	1498	1500	1500
E170	638	638	629	620	635	616	636	636	616	638	3.4%	641	638	638
E180	1083	1082	1077	1080	1082	1031	1081	1080	1031	1083	4.8%	1083	1082	1082
E185	1544	1543	1541	1547	1540	1533	1542	1538	1533	1547	0.9%	1545	1543	1543
E190	164	164	160	160	164	155	164	165	155	165	6.1%	165	164	164
E195	250	250	245	246	250	245	250	252	245	252	2.8%	252	250	250
E200	1477	1464	1468	1440	1465	1487	1480	1480	1440	1487	3.2%	1476	1477	1477

Energy Consumption, Compressor (kWh,e)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) / Analytical	Analytical TUD	HTAL1	HTAL2
E100	1319	1318	1307	1311		1363	1311	1303	1303	1363	4.6%	1319	1319	1319
E110	889	899	866	883		932	879	876	866	932	7.5%	888	889	889
E120	840	840	850	838		819	836	832	819	850	3.6%	841	839	839
E130	95	94	93	93		98	94	95	93	98	5.1%	95	94	94
E140	57	57	55	56		59	56	57	55	59	7.7%	57	57	56
E150	1000	999	1007	982		1007	992	987	982	1007	2.5%	999	999	999
E160	950	949	963	926		965	947	944	926	965	4.1%	950	949	949
E165	1283	1281	1291	1256		1275	1280	1272	1256	1291	2.8%	1279	1280	1280
E170	531	530	539	523		513	528	529	513	539	4.9%	533	530	530
E180	909	908	914	912		864	907	906	864	914	5.5%	908	908	908
E185	1340	1339	1343	1344		1331	1337	1334	1331	1344	1.0%	1340	1339	1338
E190	138	138	139	138		130	138	138	130	139	6.8%	138	138	138
E195	217	217	219	217		212	216	218	212	219	3.1%	219	217	217
E200	1250	1239	1249	1218		1260	1253	1253	1218	1260	3.3%	1249	1250	1250

Energy Consumption, Supply Fan (kWh,e)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) / Analytical	Analytical TUD	HTAL1	HTAL2
E100	144	144	145	141	144	150	144	142	141	150	6.4%	144	144	144
E110	128	129	133	122	128	134	128	127	122	134	9.6%	128	128	128
E120	117	117	110	110	116	115	117	115	110	117	6.3%	117	117	117
E130	10	10	8	8	10	11	10	10	8	11	26.9%	10	10	10
E140	8	8	7	6	8	9	8	8	6	9	30.4%	8	8	8
E150	141	141	133	136	140	142	141	139	133	142	6.4%	141	141	141
E160	129	129	119	121	128	131	129	128	119	131	9.4%	129	129	129
E165	149	150	142	145	149	149	149	148	142	150	5.6%	149	149	149
E170	73	73	61	63	73	71	73	73	61	73	16.1%	74	73	73
E180	118	119	111	112	118	113	118	118	111	119	6.9%	119	119	119
E185	139	139	135	137	139	138	139	139	135	139	3.0%	139	139	139
E190	18	18	14	14	18	17	18	18	14	18	22.8%	18	18	18
E195	23	23	18	18	23	22	23	23	18	23	23.1%	23	23	23
E200	154	153	149	151	153	155	155	155	149	155	3.9%	154	155	155

Energy Consumption, Condenser Fan (kWh,e)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) / Analytical	Analytical TUD	HTAL1	HTAL2
E100	68	68	68	67		70	67	67	67	70	5.3%	68	68	68
E110	60	61	62	60		63	60	59	59	63	5.9%	60	60	60
E120	55	55	51	55		54	55	54	51	55	6.5%	55	55	55
E130	5	5	4	5		5	5	5	4	5	24.5%	5	5	5
E140	4	4	3	4		4	4	4	3	4	19.1%	4	4	4
E150	66	66	62	65		67	66	65	62	67	6.3%	66	66	66
E160	61	61	56	60		62	61	60	56	62	9.2%	61	61	61
E165	70	70	67	69		70	70	69	67	70	5.2%	70	70	70
E170	34	34	29	34		33	34	34	29	34	16.1%	35	34	34
E180	56	56	52	56		53	56	55	52	56	7.1%	56	56	56
E185	65	65	63	66		65	65	65	63	66	3.9%	65	65	65
E190	8	9	7	8		8	8	9	7	9	27.6%	9	9	9
E195	11	11	8	11		10	11	11	8	11	25.0%	11	11	11
E200	73	72	70	71		73	73	73	70	73	4.1%	73	73	73

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COP: Mean, and (Max-Min)/ Mean

Mean COP	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) / Analytical	Analytical TUD	HTAL1	HTAL2
E100	2.39	2.39	2.43	2.41	2.40	2.31	2.40	2.42	2.31	2.43	5.0%	2.39	2.39	2.39
E110	3.38	3.34	3.46	3.41	3.40	3.22	3.41	3.43	3.22	3.46	7.1%	3.38	3.38	3.38
E120	3.59	3.59	3.61	3.62	3.61	3.68	3.61	3.63	3.59	3.68	2.5%	3.59	3.59	3.59
E130	1.91	1.91	1.98	1.95	1.90	1.84	1.92	1.92	1.84	1.98	7.2%	1.89	1.91	1.91
E140	2.77	2.73	2.92	2.85	2.77	2.65	2.80	2.80	2.65	2.92	9.6%	2.75	2.77	2.77
E150	3.62	3.63	3.67	3.70	3.65	3.61	3.65	3.67	3.61	3.70	2.5%	3.63	3.63	3.63
E160	3.84	3.84	3.87	3.95	3.86	3.78	3.85	3.86	3.78	3.95	4.4%	3.83	3.84	3.84
E165	2.92	2.92	2.95	2.99	2.94	2.95	2.93	2.94	2.92	2.99	2.2%	2.93	2.93	2.93
E170	3.38	3.39	3.44	3.48	3.40	3.52	3.39	3.40	3.38	3.52	4.2%	3.37	3.39	3.39
E180	4.04	4.04	4.08	4.03	4.04	4.27	4.05	4.06	4.03	4.27	6.0%	4.04	4.04	4.04
E185	2.85	2.85	2.87	2.82	2.85	2.92	2.85	2.86	2.82	2.92	3.4%	2.85	2.85	2.85
E190	3.41	3.41	3.49	3.46	3.39	3.63	3.41	3.40	3.39	3.63	7.0%	3.39	3.41	3.41
E195	2.31	2.31	2.36	2.34	2.30	2.41	2.32	2.31	2.30	2.41	4.7%	2.29	2.31	2.31
E200	3.62	3.61	3.67	3.71	3.65	3.61	3.61	3.61	3.61	3.71	2.7%	3.62	3.62	3.62

(Max - Min)/ Mean COP	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) / Analytical	Analytical TUD	HTAL1	HTAL2
E100	0.000	0.001	0.002	0.001	0.003	0.009	0.000	0.000	0.000	0.009		0.000	0.000	0.000
E110	0.000	0.010	0.002	0.001	0.003	0.006	0.000	0.011	0.000	0.011		0.000	0.000	0.000
E120	0.000	0.004	0.001	0.001	0.003	0.008	0.000	0.012	0.000	0.012		0.000	0.000	0.000
E130	0.000	0.038	0.013	0.009	0.004	0.011	0.000	0.172	0.000	0.172		0.000	0.000	0.000
E140	0.000	0.056	0.011	0.019	0.004	0.008	0.000	0.204	0.000	0.204		0.000	0.000	0.000
E150	0.003	0.003	0.001	0.005	0.011	0.006	0.000	0.009	0.000	0.011		0.000	0.000	0.001
E160	0.003	0.005	0.001	0.003	0.011	0.008	0.000	0.010	0.000	0.011		0.000	0.000	0.000
E165	0.010	0.003	0.001	0.003	0.012	0.007	0.000	0.008	0.000	0.012		0.000	0.000	0.000
E170	0.000	0.006	0.002	0.004	0.015	0.006	0.000	0.043	0.000	0.043		0.000	0.000	0.000
E180	0.005	0.002	0.002	0.010	0.029	0.014	0.000	0.012	0.000	0.029		0.000	0.000	0.000
E185	0.007	0.004	0.002	0.010	0.034	0.014	0.000	0.009	0.000	0.034		0.000	0.000	0.000
E190	0.000	0.023	0.007	0.019	0.040	0.025	0.000	0.101	0.000	0.101		0.000	0.000	0.000
E195	0.000	0.017	0.008	0.017	0.043	0.021	0.000	0.086	0.000	0.086		0.000	0.000	0.000
E200	0.006	0.000	0.000	0.005	0.012	0.006	0.000	0.000	0.000	0.012		0.000	0.000	0.000

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Coil Loads: Total, Sensible, and Latent

Coil Load, Total (kWh,thermal)									Statistics, All Results					
	CA-SIS	CLM2000	DOE21E	DOE21E	Energy+	Prometh	TRN-id	TRN-re			(Max-Min)		Analytical	
	EDF	EDF	CIEMAT	NREL	GARD	KST	TUD	TUD	Min	Max	/ Analytical	TUD	HTAL1	HTAL2
E100	3800	3800	3841	3794	3798	3804	3800	3798	3794	3841	1.3%	3800	3800	3800
E110	3765	3766	3804	3756	3763	3765	3765	3763	3756	3804	1.3%	3765	3765	3765
E120	3749	3749	3763	3739	3747	3740	3748	3747	3739	3763	0.6%	3749	3749	3749
E130	219	219	216	215	217	218	219	220	215	220	2.1%	219	219	219
E140	198	198	196	195	196	197	198	199	195	199	2.0%	198	198	197
E150	4517	4517	4543	4528	4509	4527	4517	4515	4509	4543	0.8%	4518	4517	4518
E160	4501	4500	4516	4508	4491	4506	4500	4499	4491	4516	0.6%	4501	4500	4500
E165	4538	4538	4567	4549	4529	4556	4537	4535	4529	4567	0.9%	4537	4537	4538
E170	2233	2232	2226	2237	2225	2230	2232	2232	2225	2237	0.5%	2232	2232	2233
E180	4495	4495	4510	4535	4481	4507	4495	4494	4481	4535	1.2%	4495	4495	4494
E185	4507	4535	4565	4583	4523	4611	4535	4534	4507	4611	2.3%	4535	4535	4534
E190	578	577	573	579	574	576	577	578	573	579	1.0%	578	577	578
E195	602	601	595	602	598	607	601	601	595	607	1.9%	601	601	601
E200	5498	5436	5534	5522	5484	5522	5498	5498	5436	5534	1.8%	5498	5498	5498

Coil Load, Sensible (kWh,thermal)									Statistics, All Results					
	CA-SIS	CLM2000	DOE21E	DOE21E	Energy+	Prometh	TRN-id	TRN-re			(Max-Min)		Analytical	
	EDF	EDF	CIEMAT	NREL	GARD	KST	TUD	TUD	Min	Max	/ Analytical	TUD	HTAL1	HTAL2
E100	3800	3800	3841	3794	3798	3804	3800	3798	3794	3841	1.3%	3800	3800	3800
E110	3765	3766	3804	3756	3763	3765	3765	3763	3756	3804	1.3%	3765	3765	3765
E120	3749	3749	3763	3739	3747	3740	3748	3747	3739	3763	0.6%	3749	3749	3749
E130	219	219	216	215	217	218	219	220	215	220	2.1%	219	219	219
E140	198	198	196	195	196	197	198	199	195	199	2.0%	198	198	197
E150	3778	3778	3804	3786	3776	3788	3778	3776	3776	3804	0.7%	3778	3778	3779
E160	3761	3761	3777	3769	3759	3766	3761	3760	3759	3777	0.5%	3761	3761	3761
E165	3798	3798	3828	3809	3795	3817	3798	3796	3795	3828	0.9%	3798	3798	3799
E170	1493	1493	1487	1498	1491	1493	1492	1492	1487	1498	0.7%	1493	1493	1493
E180	1537	1538	1553	1607	1537	1563	1538	1537	1537	1607	4.5%	1538	1538	1538
E185	1548	1578	1608	1653	1577	1665	1578	1577	1548	1665	7.4%	1578	1578	1578
E190	208	208	203	212	206	208	208	208	203	212	4.4%	208	208	208
E195	232	232	226	235	230	239	231	232	226	239	5.6%	232	232	232
E200	4276	4215	4313	4303	4274	4300	4277	4277	4215	4313	2.3%	4277	4277	4277

Coil Load, Latent (kWh,thermal)									Statistics, All Results					
	CA-SIS	CLM2000	DOE21E	DOE21E	Energy+	Prometh	TRN-id	TRN-re			(Max-Min)		Analytical	
	EDF	EDF	CIEMAT	NREL	GARD	KST	TUD	TUD	Min	Max	/ Analytical	TUD	HTAL1	HTAL2
E100	0	0	0	0	0	0	0	0	0	0		0	0	0
E110	0	0	0	0	0	0	0	0	0	0		0	0	0
E120	0	0	0	0	0	0	0	0	0	0		0	0	0
E130	0	0	0	0	0	0	0	0	0	0		0	0	0
E140	0	0	0	0	0	0	0	0	0	0		0	0	0
E150	739	739	739	742	733	739	739	739	733	742	1.2%	739	739	739
E160	740	739	739	739	732	739	739	739	732	740	1.1%	739	739	739
E165	740	739	739	740	733	739	739	739	733	740	1.0%	739	739	739
E170	740	739	739	739	734	738	739	739	734	740	0.9%	739	739	739
E180	2958	2957	2957	2928	2944	2943	2957	2957	2928	2958	1.0%	2957	2957	2956
E185	2959	2957	2957	2930	2946	2946	2957	2957	2930	2959	1.0%	2958	2957	2956
E190	370	370	370	366	368	368	370	370	366	370	1.0%	370	370	370
E195	370	370	370	367	368	368	370	370	367	370	0.9%	370	370	370
E200	1222	1221	1221	1219	1210	1222	1221	1221	1210	1222	1.0%	1221	1221	1221

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Sensible Coil - Zone Load, (Fan Heat) (kWh,thermal)									Statistics, All Results					
	CA-SIS	CLM2000	DOE21E	DOE21E	Energy+	Prometh	TRN-id	TRN-re			(Max-Min)		Analytical	
	EDF	EDF	CIEMAT	NREL	GARD	KST	TUD	TUD	Min	Max	/ Analytical	TUD	HTAL1	HTAL2
E100	144	144	187	139	144	148	144	142	139	187	33.6%	144	144	144
E110	128	129	168	119	128	127	128	127	119	168	38.2%	128	128	128
E120	117	117	133	108	116	107	117	115	107	133	22.1%	117	117	117
E130	10	10	8	8	10	8	10	10	8	10	26.7%	10	10	10
E140	8	8	7	6	8	7	8	8	6	8	25.0%	8	8	8
E150	141	141	168	149	140	138	141	139	138	168	21.3%	141	141	142
E160	129	129	147	137	129	125	129	128	125	147	16.9%	129	129	129
E165	149	149	181	161	149	147	149	148	147	181	23.0%	149	149	150
E170	73	73	69	79	73	61	73	73	61	79	24.5%	74	73	74
E180	117	118	135	188	119	106	118	118	106	188	69.2%	118	119	118
E185	109	139	171	215	140	134	139	139	109	215	76.6%	139	139	139
E190	18	18	15	24	18	14	18	18	14	24	56.1%	18	18	18
E195	23	23	18	28	23	18	23	23	18	28	43.8%	23	23	23
E200	154	153	193	181	154	155	155	155	153	193	25.7%	154	155	155

Zone Loads: Total, Sensible, and Latent

Zone Load, Total (kWh,thermal)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) /Analytical	Analytical TUD	HTAL1	HTAL2
E100	3656	3656	3654	3655	3654	3657	3656	3656	3654	3657	0.1%	3656	3656	3656
E110	3637	3637	3636	3637	3636	3637	3637	3637	3636	3637	0.1%	3637	3637	3637
E120	3632	3632	3630	3632	3631	3632	3632	3631	3630	3632	0.1%	3632	3632	3632
E130	209	209	207	208	207	209	209	209	207	209	1.3%	209	209	209
E140	190	190	189	188	188	190	190	190	188	190	1.1%	190	190	190
E150	4376	4376	4375	4376	4375	4389	4376	4376	4375	4389	0.3%	4376	4376	4376
E160	4371	4371	4370	4371	4370	4381	4371	4371	4370	4381	0.3%	4371	4371	4371
E165	4388	4388	4386	4387	4386	4409	4388	4387	4386	4409	0.5%	4388	4388	4388
E170	2159	2159	2157	2158	2157	2169	2159	2159	2157	2169	0.6%	2159	2159	2159
E180	4376	4376	4375	4376	4375	4401	4376	4376	4375	4401	0.6%	4376	4376	4376
E185	4396	4396	4394	4395	4393	4477	4395	4395	4393	4477	1.9%	4396	4396	4396
E190	557	559	558	558	558	563	559	559	557	563	1.0%	559	559	559
E195	576	579	577	577	576	589	578	579	576	589	2.3%	579	579	579
E200	5343	5283	5342	5343	5342	5367	5343	5343	5283	5367	1.6%	5343	5343	5343

Zone Load, Sensible (kWh,thermal)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) /Analytical	Analytical TUD	HTAL1	HTAL2
E100	3656	3656	3654	3655	3654	3657	3656	3656	3654	3657	0.1%	3656	3656	3656
E110	3637	3637	3636	3637	3636	3637	3637	3637	3636	3637	0.1%	3637	3637	3637
E120	3632	3632	3630	3632	3631	3632	3632	3631	3630	3632	0.1%	3632	3632	3632
E130	209	209	207	208	207	209	209	209	207	209	1.3%	209	209	209
E140	190	190	189	188	188	190	190	190	188	190	1.1%	190	190	190
E150	3637	3637	3636	3637	3636	3650	3637	3636	3636	3650	0.4%	3637	3637	3637
E160	3632	3632	3630	3632	3631	3642	3632	3631	3630	3642	0.3%	3632	3632	3632
E165	3649	3649	3647	3648	3647	3670	3649	3648	3647	3670	0.6%	3649	3649	3649
E170	1420	1420	1418	1419	1418	1432	1419	1419	1418	1432	1.0%	1420	1420	1420
E180	1420	1420	1418	1419	1418	1457	1419	1419	1418	1457	2.8%	1420	1420	1420
E185	1439	1439	1437	1437	1437	1531	1438	1438	1437	1531	6.6%	1439	1439	1439
E190	190	190	188	188	188	195	190	190	188	195	3.5%	190	190	190
E195	209	209	207	208	207	221	209	209	207	221	6.8%	209	209	209
E200	4122	4062	4121	4122	4121	4145	4122	4122	4062	4145	2.0%	4122	4122	4122

Zone Load, Latent (kWh,thermal)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) /Analytical	Analytical TUD	HTAL1	HTAL2
E100	0	0	0	0	0	0	0	0	0	0		0	0	0
E110	0	0	0	0	0	0	0	0	0	0		0	0	0
E120	0	0	0	0	0	0	0	0	0	0		0	0	0
E130	0	0	0	0	0	0	0	0	0	0		0	0	0
E140	0	0	0	0	0	0	0	0	0	0		0	0	0
E150	739	739	739	739	739	739	739	739	739	739	0.1%	739	739	739
E160	739	739	739	739	739	739	739	739	739	739	0.1%	739	739	739
E165	739	739	739	739	739	739	739	739	739	739	0.1%	739	739	739
E170	739	739	739	739	739	738	739	739	738	739	0.2%	739	739	739
E180	2957	2957	2957	2958	2957	2943	2957	2957	2943	2958	0.5%	2957	2957	2957
E185	2957	2957	2957	2958	2957	2946	2957	2957	2946	2958	0.4%	2957	2957	2957
E190	367	370	370	370	370	368	370	370	367	370	0.8%	370	370	370
E195	367	370	370	370	370	368	370	370	367	370	0.8%	370	370	370
E200	1221	1221	1221	1221	1221	1222	1221	1221	1221	1222	0.1%	1221	1221	1221

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Latent Coil - Zone Load, (Should be 0) (kWh,thermal)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) /Analytical	Analytical TUD	HTAL1	HTAL2
E100	0	0	0	0	0	0	0	0	0	0		0	0	0
E110	0	0	0	0	0	0	0	0	0	0		0	0	0
E120	0	0	0	0	0	0	0	0	0	0		0	0	0
E130	0	0	0	0	0	0	0	0	0	0		0	0	0
E140	0	0	0	0	0	0	0	0	0	0		0	0	0
E150	0	0	0	2	-7	0	0	0	-7	2		0	0	0
E160	1	0	0	0	-7	0	0	0	-7	1		0	0	0
E165	1	0	0	1	-6	0	0	0	-6	1		0	0	0
E170	1	0	0	-1	-6	0	0	0	-6	1		0	0	0
E180	1	0	0	-30	-13	0	0	0	-30	1		1	0	-1
E185	2	0	0	-28	-11	0	0	0	-28	2		1	0	-1
E190	3	0	0	-3	-2	0	0	0	-3	3		0	0	0
E195	3	0	0	-3	-1	0	0	0	-3	3		0	0	0
E200	1	0	0	-2	-11	0	0	0	-11	1		0	0	0

Sensitivities for Space Cooling Electricity Consumption

Delta Qtot (kWh,e)									Statistics, All Results				Analytical	
	CA-SIS	CLM2000	DOE21E	DOE21E	Energy+	Prometh	TRN-id	TRN-re			Abs(Max-		Analytical	
	EDF	EDF	CIEMAT	NREL	GARD	KST	TUD	TUD	Min	Max	Min)/Analy.	TUD	HTAL1	HTAL2
E110-E100	-454	-441	-460	-454	-451	-454	-455	-450	-460	-441	4.1%	-454	-454	-453
E120-E110	-65	-77	-50	-62	-63	-141	-60	-60	-141	-50	143.5%	-64	-66	-66
E120-E100	-519	-518	-510	-516	-514	-596	-515	-510	-596	-510	16.6%	-518	-520	-520
E130-E100	-1421	-1421	-1415	-1413	-1411	-1470	-1414	-1402	-1470	-1402	4.8%	-1420	-1421	-1421
E140-E130	-42	-40	-40	-40	-41	-42	-41	-41	-42	-40	4.8%	-42	-41	-41
E140-E110	-1009	-1020	-996	-999	-1001	-1058	-999	-993	-1058	-993	6.4%	-1007	-1009	-1009
E150-E110	131	118	141	118	128	86	132	130	86	141	42.5%	130	129	129
E160-E150	-68	-68	-65	-76	-65	-58	-62	-59	-76	-58	28.0%	-66	-67	-68
E165-E160	362	362	362	363	359	335	363	357	335	363	7.8%	357	360	361
E170-E150	-570	-569	-573	-563	-562	-599	-563	-556	-599	-556	7.6%	-565	-569	-569
E180-E150	-125	-125	-125	-103	-115	-185	-118	-112	-185	-103	66.1%	-124	-124	-125
E180-E170	445	444	448	460	447	414	445	444	414	460	10.4%	442	445	444
E185-E180	461	461	464	467	458	502	460	458	458	502	9.5%	462	461	461
E190-E180	-919	-918	-917	-920	-918	-876	-917	-915	-920	-876	4.9%	-917	-918	-918
E190-E140	96	95	95	94	96	83	96	96	83	96	13.6%	96	96	96
E195-E190	86	86	85	86	86	89	86	86	85	89	5.4%	87	86	86
E195-E185	-1294	-1293	-1296	-1301	-1290	-1288	-1292	-1287	-1301	-1287	1.1%	-1292	-1293	-1293
E195-E130	140	141	140	140	142	131	141	141	131	142	7.7%	142	141	141
E200-E100	-54	-66	-53	-79	-55	-96	-42	-32	-96	-32	117.6%	-55	-53	-54

Del Qcomp (kWh,e)									Statistics, All Results					
	CA-SIS	CLM2000	DOE21E	DOE21E	Energy+	Prometh	TRN-id	TRN-re			Abs(Max-		Analytical	
	EDF	EDF	CIEMAT	NREL	GARD	KST	TUD	TUD	Min	Max	Min)/Analy.	TUD	HTAL1	HTAL2
E110-E100	-430	-419	-442	-428		-431	-432	-427	-442	-419	5.2%	-431	-430	-430
E120-E110	-49	-59	-16	-45		-113	-43	-44	-113	-16	206.3%	-47	-50	-50
E120-E100	-479	-478	-457	-473		-544	-475	-471	-544	-457	18.0%	-478	-480	-480
E130-E100	-1224	-1224	-1214	-1218		-1265	-1218	-1208	-1265	-1208	4.7%	-1224	-1225	-1225
E140-E130	-38	-37	-38	-37		-39	-38	-38	-39	-37	4.4%	-38	-38	-38
E140-E110	-832	-842	-811	-827		-873	-823	-819	-873	-811	7.5%	-831	-833	-833
E150-E110	111	100	141	99		75	113	111	75	141	60.1%	111	110	110
E160-E150	-50	-50	-44	-56		-42	-45	-42	-56	-42	29.7%	-49	-50	-50
E165-E160	333	332	329	330		310	333	328	310	333	7.2%	328	331	331
E170-E150	-469	-469	-468	-459		-494	-464	-458	-494	-458	7.7%	-466	-469	-469
E180-E150	-91	-91	-93	-70		-143	-85	-80	-143	-70	80.0%	-91	-91	-92
E180-E170	378	378	375	389		352	379	378	352	389	10.0%	375	378	378
E185-E180	431	431	428	432		466	430	428	428	466	8.9%	432	431	431
E190-E180	-771	-770	-775	-774		-734	-770	-768	-775	-734	5.3%	-770	-770	-770
E190-E140	81	81	85	82		71	82	82	71	85	17.0%	82	81	81
E195-E190	79	79	79	79		82	79	80	79	82	3.9%	80	79	79
E195-E185	-1123	-1122	-1124	-1127		-1118	-1120	-1116	-1127	-1116	1.0%	-1121	-1122	-1121
E195-E130	122	123	126	124		114	123	123	114	126	9.3%	123	122	123
E200-E100	-69	-79	-58	-93		-103	-58	-50	-103	-50	76.3%	-70	-69	-69

Del Q IDfan (kWh,e)									Statistics, All Results					
	CA-SIS	CLM2000	DOE21E	DOE21E	Energy+	Prometh	TRN-id	TRN-re			Abs(Max-		Analytical	
	EDF	EDF	CIEMAT	NREL	GARD	KST	TUD	TUD	Min	Max	Min)/Analy.	TUD	HTAL1	HTAL2
E110-E100	-16	-15	-12	-19	-16	-16	-16	-16	-19	-12	42.3%	-16	-16	-16
E120-E110	-11	-12	-23	-12	-11	-19	-11	-11	-23	-11	110.0%	-11	-11	-11
E120-E100	-27	-27	-36	-31	-27	-35	-27	-27	-36	-27	32.2%	-27	-27	-27
E130-E100	-134	-134	-137	-133	-133	-139	-133	-132	-139	-132	5.6%	-134	-134	-134
E140-E130	-2	-2	-1	-2	-2	-2	-2	-2	-2	-1	42.4%	-2	-2	-2
E140-E110	-120	-121	-126	-116	-119	-126	-120	-118	-126	-116	8.3%	-120	-120	-120
E150-E110	13	12	0	14	13	8	13	13	0	14	107.0%	13	13	13
E160-E150	-12	-12	-14	-15	-12	-11	-12	-11	-15	-11	35.3%	-12	-12	-12
E165-E160	20	21	23	24	20	17	20	20	17	24	33.4%	20	20	20
E170-E150	-68	-68	-72	-73	-67	-71	-68	-66	-73	-66	9.7%	-68	-68	-68
E180-E150	-23	-22	-22	-24	-22	-29	-22	-21	-29	-21	32.6%	-22	-23	-23
E180-E170	45	46	49	49	45	43	45	45	43	49	15.0%	45	45	45
E185-E180	21	20	24	25	21	24	21	21	20	25	24.3%	21	21	21
E190-E180	-100	-101	-97	-98	-100	-96	-100	-100	-101	-96	4.7%	-101	-101	-101
E190-E140	10	10	7	8	10	9	10	10	7	10	28.1%	10	10	10
E195-E190	5	5	4	4	5	5	5	5	4	5	30.6%	5	5	5
E195-E185	-116	-116	-117	-119	-116	-116	-117	-116	-119	-116	2.7%	-117	-117	-117
E195-E130	13	13	9	10	12	11	12	12	9	13	28.9%	12	12	12
E200-E100	10	9	4	10	10	5	11	12	4	12	79.1%	10	11	11

Del Q ODfan (kWh,e)									Statistics, All Results					
	CA-SIS	CLM2000	DOE21E	DOE21E	Energy+	Prometh	TRN-id	TRN-re			Abs(Max-		Analytical	
	EDF	EDF	CIEMAT	NREL	GARD	KST	TUD	TUD	Min	Max	Min)/Analy.	TUD	HTAL1	HTAL2
E110-E100	-8	-7	-6	-7		-7	-7	-7	-8	-6	29.9%	-7	-7	-7
E120-E110	-5	-6	-11	-5		-9	-5	-5	-11	-5	113.1%	-5	-5	-5
E120-E100	-13	-13	-17	-12		-17	-13	-13	-17	-12	37.0%	-13	-13	-13
E130-E100	-63	-63	-64	-62		-65	-63	-62	-65	-62	5.3%	-63	-63	-63
E140-E130	-1	-1	-1	-1		-1	-1	-1	-1	-1	44.4%	-1	-1	-1
E140-E110	-56	-57	-59	-56		-59	-56	-56	-59	-56	6.3%	-56	-56	-56
E150-E110	6	5	0	5		4	6	6	0	6	100.8%	6	6	6
E160-E150	-5	-5	-7	-5		-5	-6	-5	-7	-5	27.0%	-6	-6	-6
E165-E160	9	9	11	9		8	10	9	8	11	26.1%	9	9	9
E170-E150	-32	-32	-34	-31		-34	-32	-31	-34	-31	8.2%	-32	-32	-32
E180-E150	-10	-10	-10	-9		-13	-11	-10	-13	-9	41.7%	-11	-11	-11
E180-E170	22	22	23	22		20	21	21	20	23	14.8%	21	21	21
E185-E180	9	9	11	10		11	10	10	9	11	24.8%	10	10	10
E190-E180	-48	-47	-45	-48		-45	-47	-47	-48	-45	5.9%	-47	-47	-47
E190-E140	4	5	3	4		4	5	5	3	5	34.6%	5	5	5
E195-E190	3	2	2	3		2	2	2	2	3	60.5%	2	2	2
E195-E185	-54	-54	-55	-55		-54	-55	-54	-55	-54	2.0%	-55	-55	-55
E195-E130	6	6	4	6		5	6	6	4	6	27.1%	6	6	6
E200-E100	5	4	2	4		2	5	6	2	6	79.1%	5	5	5

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Sensitivities for COP and Coil Loads

Delta COP (kWh,t)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	Abs(Max-Min)/Analy.	Analytical TUD	HTAL1	HTAL2
E110-E100	0.99	0.95	1.03	1.01	1.00	0.91	1.01	1.01	0.91	1.03	12.0%	0.99	0.99	0.99
E120-E110	0.21	0.25	0.16	0.21	0.21	0.46	0.20	0.20	0.16	0.46	147.3%	0.21	0.21	0.21
E120-E100	1.20	1.20	1.18	1.22	1.20	1.37	1.20	1.21	1.18	1.37	15.6%	1.20	1.20	1.20
E130-E100	-0.48	-0.48	-0.46	-0.45	-0.50	-0.47	-0.48	-0.50	-0.50	-0.45	9.8%	-0.50	-0.48	-0.48
E140-E130	0.86	0.83	0.94	0.90	0.87	0.81	0.88	0.88	0.81	0.94	15.1%	0.86	0.86	0.86
E140-E110	-0.61	-0.61	-0.54	-0.56	-0.63	-0.57	-0.61	-0.63	-0.63	-0.54	13.6%	-0.63	-0.61	-0.61
E150-E110	0.24	0.29	0.21	0.29	0.25	0.39	0.24	0.25	0.21	0.39	73.2%	0.25	0.25	0.25
E160-E150	0.22	0.21	0.20	0.25	0.21	0.17	0.20	0.19	0.17	0.25	38.9%	0.21	0.21	0.21
E165-E160	-0.92	-0.92	-0.91	-0.96	-0.92	-0.83	-0.92	-0.92	-0.96	-0.83	14.9%	-0.90	-0.91	-0.91
E170-E150	-0.24	-0.24	-0.23	-0.22	-0.26	-0.09	-0.26	-0.27	-0.27	-0.09	69.2%	-0.26	-0.24	-0.24
E180-E150	0.42	0.41	0.42	0.33	0.39	0.66	0.40	0.38	0.33	0.66	80.5%	0.42	0.41	0.41
E180-E170	0.66	0.65	0.64	0.55	0.65	0.75	0.65	0.65	0.55	0.75	29.8%	0.68	0.65	0.65
E185-E180	-1.19	-1.19	-1.21	-1.20	-1.19	-1.35	-1.20	-1.20	-1.35	-1.19	13.4%	-1.20	-1.19	-1.19
E190-E180	-0.63	-0.63	-0.60	-0.57	-0.65	-0.64	-0.64	-0.65	-0.65	-0.57	12.4%	-0.66	-0.63	-0.63
E190-E140	0.64	0.68	0.57	0.60	0.62	0.98	0.61	0.61	0.57	0.98	64.2%	0.64	0.64	0.64
E195-E190	-1.10	-1.10	-1.13	-1.12	-1.09	-1.22	-1.09	-1.10	-1.22	-1.09	11.8%	-1.09	-1.10	-1.10
E195-E185	-0.54	-0.54	-0.51	-0.49	-0.55	-0.51	-0.54	-0.55	-0.55	-0.49	12.0%	-0.55	-0.54	-0.54
E195-E130	0.40	0.40	0.38	0.38	0.40	0.57	0.40	0.39	0.38	0.57	45.9%	0.40	0.40	0.40
E200-E100	1.23	1.22	1.24	1.30	1.24	1.30	1.21	1.19	1.19	1.30	8.9%	1.23	1.23	1.23

Del Q coil,t (kWh,t)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	Abs(Max-Min)/Analy.	Analytical TUD	HTAL1	HTAL2
E110-E100	-35	-34	-38	-38	-35	-40	-35	-35	-40	-34	15.7%	-35	-35	-35
E120-E110	-16	-17	-40	-16	-16	-25	-16	-16	-40	-16	147.5%	-16	-16	-17
E120-E100	-51	-51	-78	-55	-51	-65	-51	-51	-78	-51	52.9%	-51	-52	-52
E130-E100	-3581	-3581	-3626	-3579	-3581	-3587	-3581	-3578	-3626	-3578	1.3%	-3581	-3581	-3581
E140-E130	-21	-21	-20	-21	-21	-21	-21	-21	-21	-20	5.0%	-21	-21	-22
E140-E110	-3567	-3568	-3608	-3561	-3567	-3568	-3567	-3565	-3608	-3561	1.3%	-3567	-3567	-3568
E150-E110	752	751	739	772	746	762	752	752	739	772	4.4%	752	752	753
E160-E150	-16	-17	-26	-19	-18	-21	-17	-16	-26	-16	61.3%	-17	-17	-18
E165-E160	37	38	51	40	38	50	37	36	36	51	40.8%	36	37	38
E170-E150	-2284	-2285	-2317	-2291	-2284	-2296	-2285	-2283	-2317	-2283	1.5%	-2285	-2286	-2286
E180-E150	-22	-22	-33	7	-28	-20	-22	-21	-33	7	178.9%	-22	-23	-25
E180-E170	2262	2263	2284	2298	2256	2276	2263	2262	2256	2298	1.8%	2263	2263	2261
E185-E180	12	40	55	48	41	104	40	40	12	104	231.3%	40	40	40
E190-E180	-3917	-3918	-3937	-3956	-3907	-3931	-3917	-3916	-3956	-3907	1.3%	-3918	-3918	-3916
E190-E140	380	379	377	384	378	380	380	379	377	384	1.8%	380	379	380
E195-E190	24	24	23	23	23	31	24	24	23	31	33.0%	24	24	24
E195-E185	-3905	-3934	-3970	-3981	-3925	-4004	-3934	-3933	-4004	-3905	2.5%	-3934	-3934	-3933
E195-E130	383	382	379	387	381	389	382	382	379	389	2.5%	382	382	382
E200-E100	1698	1636	1693	1728	1687	1718	1698	1700	1636	1728	5.4%	1697	1697	1697

Del Q coil,s (kWh,t)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	Abs(Max-Min)/Analy.	Analytical TUD	HTAL1	HTAL2
E110-E100	-35	-34	-38	-38	-35	-40	-35	-35	-40	-34	15.7%	-35	-35	-35
E120-E110	-16	-17	-40	-16	-16	-25	-16	-16	-40	-16	147.5%	-16	-16	-17
E120-E100	-51	-51	-78	-55	-51	-65	-51	-51	-78	-51	52.9%	-51	-52	-52
E130-E100	-3581	-3581	-3626	-3579	-3581	-3587	-3581	-3578	-3626	-3578	1.3%	-3581	-3581	-3581
E140-E130	-21	-21	-20	-21	-21	-21	-21	-21	-21	-20	5.0%	-21	-21	-22
E140-E110	-3567	-3568	-3608	-3561	-3567	-3568	-3567	-3565	-3608	-3561	1.3%	-3567	-3567	-3568
E150-E110	13	12	0	30	13	23	13	13	0	30	232.9%	13	13	14
E160-E150	-17	-17	-26	-17	-17	-21	-17	-16	-26	-16	60.3%	-17	-17	-18
E165-E160	37	37	51	40	36	50	37	36	36	51	40.8%	36	37	38
E170-E150	-2285	-2285	-2317	-2288	-2285	-2295	-2285	-2283	-2317	-2283	1.5%	-2285	-2286	-2286
E180-E150	-2241	-2240	-2250	-2179	-2239	-2224	-2240	-2239	-2250	-2179	3.2%	-2241	-2240	-2241
E180-E170	44	45	66	109	46	71	45	45	44	109	145.3%	45	45	45
E185-E180	11	40	55	46	39	101	40	40	11	101	226.6%	40	40	40
E190-E180	-1329	-1330	-1350	-1394	-1331	-1355	-1330	-1329	-1394	-1329	4.9%	-1330	-1330	-1330
E190-E140	10	10	7	18	10	12	10	9	7	18	102.7%	10	10	11
E195-E190	24	24	23	23	23	30	24	24	23	30	31.3%	24	24	24
E195-E185	-1316	-1346	-1382	-1418	-1347	-1426	-1346	-1345	-1426	-1316	8.2%	-1346	-1347	-1346
E195-E130	13	13	10	20	13	21	12	12	10	21	89.0%	12	12	12
E200-E100	476	415	472	509	477	495	477	479	415	509	19.7%	476	476	476

Del Qcoil,lat (kWh,t)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	Abs(Max-Min)/Analy.	Analytical TUD	HTAL1	HTAL2
E110-E100	0	0	0	0	0	0	0	0	0	0		0	0	0
E120-E110	0	0	0	0	0	0	0	0	0	0		0	0	0
E120-E100	0	0	0	0	0	0	0	0	0	0		0	0	0
E130-E100	0	0	0	0	0	0	0	0	0	0		0	0	0
E140-E130	0	0	0	0	0	0	0	0	0	0		0	0	0
E140-E110	0	0	0	0	0	0	0	0	0	0		0	0	0
E150-E110	739	739	739	742	733	739	739	739	733	742	1.2%	739	739	739
E160-E150	1	0	0	-2	-1	0	0	0	-2	1		0	0	0
E165-E160	0	0	0	1	1	0	0	0	0	1		0	0	0
E170-E150	1	0	0	-3	1	-1	0	0	-3	1		0	0	0
E180-E150	2219	2218	2218	2186	2211	2205	2218	2218	2186	2219	1.5%	2218	2218	2217
E180-E170	2218	2218	2218	2189	2210	2206	2218	2218	2189	2218	1.3%	2218	2218	2217
E185-E180	1	0	0	2	2	3	0	0	0	3		0	0	0
E190-E180	-2588	-2587	-2587	-2562	-2576	-2575	-2587	-2587	-2588	-2562	1.0%	-2588	-2587	-2586
E190-E140	370	370	370	366	368	368	370	370	366	370	1.0%	370	370	370
E195-E190	0	0	0	0	0	0	0	0	0	0		0	0	0
E195-E185	-2589	-2587	-2587	-2563	-2578	-2578	-2587	-2587	-2589	-2563	1.0%	-2588	-2587	-2587
E195-E130	370	370	370	367	368	368	370	370	367	370	0.9%	370	370	370
E200-E100	1222	1221	1221	1219	1210	1222	1221	1221	1210	1222	1.0%	1221	1221	1221

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Indoor Drybulb Temperature: Mean and (Max-Min)/ Mean

Mean IDB (°C)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) / Analytical	Analytical TUD	HTAL1	HTAL2
E100	22.2	22.2	22.3	22.3	22.2	22.2	22.2	22.6	22.2	22.6	2.0%	22.2	22.2	22.2
E110	22.2	22.2	22.3	22.3	22.2	22.2	22.2	22.5	22.2	22.5	1.5%	22.2	22.2	22.2
E120	26.7	26.7	26.8	26.7	26.7	26.7	26.7	27.1	26.7	27.1	1.4%	26.7	26.7	26.7
E130	22.2	22.2	22.1	22.1	22.2	22.2	22.2	21.6	21.6	22.2	2.5%	22.2	22.2	22.2
E140	22.2	22.2	22.1	22.1	22.2	22.2	22.2	21.5	21.5	22.2	3.1%	22.2	22.2	22.2
E150	22.2	22.2	22.3	22.3	22.2	22.2	22.2	22.7	22.2	22.7	2.1%	22.2	22.2	22.2
E160	26.7	26.7	26.8	26.7	26.7	26.7	26.7	27.0	26.7	27.0	1.1%	26.7	26.7	26.7
E165	23.3	23.3	23.4	23.4	23.3	23.3	23.3	23.8	23.3	23.8	2.1%	23.3	23.3	23.3
E170	22.2	22.2	22.2	22.2	22.2	22.2	22.2	22.1	22.1	22.2	0.5%	22.2	22.2	22.2
E180	22.2	22.2	22.3	22.3	22.2	22.2	22.2	22.3	22.2	22.3	0.6%	22.2	22.2	22.2
E185	22.2	22.2	22.3	22.3	22.2	22.2	22.2	22.4	22.2	22.4	0.8%	22.2	22.2	22.2
E190	22.2	22.2	22.1	22.1	22.2	22.2	22.2	21.9	21.9	22.2	1.1%	22.2	22.2	22.2
E195	22.2	22.2	22.1	22.1	22.2	22.2	22.2	22.0	22.0	22.2	0.9%	22.2	22.2	22.2
E200	26.7	26.7	26.8	26.8	26.7	26.7	26.7	26.7	26.7	26.8	0.4%	26.7	26.7	26.7

(Max - Min)/ Mean IDB (°C)	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) / Analytical	Analytical TUD	HTAL1	HTAL2
E100	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.049	0.000	0.049		0.000	0.000	0.002
E110	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.048	0.000	0.048		0.000	0.000	0.002
E120	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.077	0.000	0.077		0.000	0.000	0.002
E130	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.056	0.000	0.056		0.000	0.000	0.001
E140	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.069	0.000	0.069		0.000	0.000	0.002
E150	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.054	0.000	0.054		0.000	0.000	0.002
E160	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.045	0.000	0.045		0.000	0.000	0.002
E165	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.051	0.000	0.051		0.000	0.000	0.002
E170	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.050	0.000	0.050		0.000	0.000	0.001
E180	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.035	0.000	0.035		0.000	0.000	0.001
E185	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.021	0.000	0.021		0.000	0.000	0.001
E190	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.028	0.000	0.028		0.000	0.000	0.001
E195	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.023	0.000	0.023		0.000	0.000	0.001
E200	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000		0.000	0.000	0.000

Humidity Ratio: Mean and (Max-Min)/ Mean

Mean Humidity Ratio	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) / Analytical	Analytical TUD	HTAL1	HTAL2
E100	0.0075	0.0069	0.0076	0.0074	0.0075	0.0075	0.0075	0.0075	0.0069	0.0076	9.3%	0.0074	0.0073	0.0073
E110	0.0066	0.0069	0.0070	0.0064	0.0066	0.0065	0.0066	0.0066	0.0064	0.0070	9.7%	0.0065	0.0064	0.0064
E120	0.0080	0.0070	0.0078	0.0078	0.0080	0.0072	0.0080	0.0080	0.0070	0.0080	13.2%	0.0079	0.0079	0.0079
E130	0.0075	0.0069	0.0076	0.0073	0.0075	0.0075	0.0075	0.0075	0.0069	0.0076	9.3%	0.0074	0.0073	0.0073
E140	0.0065	0.0069	0.0071	0.0064	0.0066	0.0065	0.0066	0.0066	0.0064	0.0071	10.1%	0.0065	0.0064	0.0064
E150	0.0083	0.0085	0.0082	0.0083	0.0084	0.0084	0.0083	0.0085	0.0082	0.0085	4.0%	0.0082	0.0082	0.0082
E160	0.0102	0.0101	0.0097	0.0099	0.0103	0.0103	0.0101	0.0102	0.0097	0.0103	6.0%	0.0100	0.0099	0.0099
E165	0.0093	0.0099	0.0090	0.0092	0.0094	0.0094	0.0093	0.0095	0.0090	0.0099	9.1%	0.0093	0.0092	0.0092
E170	0.0106	0.0107	0.0105	0.0105	0.0106	0.0107	0.0105	0.0105	0.0105	0.0107	2.2%	0.0104	0.0105	0.0105
E180	0.0164	0.0164	0.0166	0.0164	0.0162	0.0164	0.0163	0.0164	0.0162	0.0166	2.6%	0.0162	0.0162	0.0162
E185	0.0162	0.0171	0.0164	0.0162	0.0161	0.0161	0.0162	0.0163	0.0161	0.0171	6.4%	0.0161	0.0161	0.0161
E190	0.0160	0.0161	0.0163	0.0159	0.0159	0.0161	0.0159	0.0157	0.0157	0.0163	3.5%	0.0158	0.0159	0.0159
E195	0.0156	0.0164	0.0158	0.0155	0.0154	0.0156	0.0155	0.0153	0.0153	0.0164	7.0%	0.0154	0.0154	0.0154
E200	0.0114	0.0115	0.0109	0.0111	0.0115	0.0115	0.0113	0.0113	0.0109	0.0115	5.1%	0.0111	0.0111	0.0111

(Max - Min)/ Mean Humidity Ratio	CA-SIS EDF	CLM2000 EDF	DOE21E CIEMAT	DOE21E NREL	Energy+ GARD	Prometh KST	TRN-id TUD	TRN-re TUD	Statistics, All Results Min	Max	(Max-Min) / Analytical	Analytical TUD	HTAL1	HTAL2
E100	0.000	0.022	0.000	0.000	0.001	0.000	0.000	0.000	0.0000	0.0217		0.000	0.000	0.000
E110	0.000	0.022	0.014	0.000	0.000	0.000	0.000	0.000	0.0000	0.0217		0.000	0.000	0.000
E120	0.000	0.000	0.000	0.000	0.001	0.000	0.000	0.000	0.0000	0.0005		0.000	0.000	0.000
E130	0.000	0.010	0.000	0.000	0.001	0.000	0.000	0.000	0.0000	0.0101		0.000	0.000	0.000
E140	0.000	0.012	0.014	0.000	0.001	0.000	0.000	0.000	0.0000	0.0142		0.000	0.000	0.000
E150	0.012	0.000	0.000	0.000	0.013	0.012	0.000	0.013	0.0000	0.0132		0.000	0.000	0.000
E160	0.020	0.000	0.010	0.010	0.013	0.010	0.000	0.011	0.0000	0.0196		0.000	0.000	0.000
E165	0.011	0.001	0.011	0.000	0.013	0.011	0.000	0.013	0.0000	0.0131		0.000	0.000	0.000
E170	0.000	0.000	0.010	0.000	0.011	0.009	0.000	0.024	0.0000	0.0238		0.000	0.000	0.001
E180	0.018	0.000	0.012	0.012	0.010	0.012	0.000	0.040	0.0000	0.0402		0.000	0.000	0.001
E185	0.012	0.006	0.018	0.012	0.011	0.019	0.000	0.025	0.0000	0.0246		0.000	0.000	0.001
E190	0.000	0.000	0.018	0.019	0.014	0.012	0.000	0.031	0.0000	0.0312		0.000	0.000	0.001
E195	0.000	0.006	0.019	0.019	0.014	0.013	0.000	0.024	0.0000	0.0241		0.000	0.000	0.001
E200	0.018	0.000	0.009	0.009	0.013	0.009	0.000	0.000	0.0000	0.0175		0.000	0.000	0.000

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This report describes the Building Energy Simulation Test for Heating, Ventilating, and Air-Conditioning Equipment Models (HVAC BESTEST) project conducted by the Tool Evaluation and Improvement International Energy Agency (IEA) Experts Group. The group was composed of experts from the Solar Heating and Cooling (SHC) Programme, Task 22, Subtask A. The current test cases, E100–E200, represent the beginning of work on mechanical equipment test cases; additional cases that would expand the current test suite have been proposed for future development.

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