Empirical Validation of Iowa Energy Resource Station Building Energy Analysis Simulation Models

A Report of Task 22, Subtask A
Building Energy Analysis Tools
Project A.1 Empirical Validation
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INTERNATIONAL ENERGY AGENCY
Solar Heating & Cooling Programme

PREFACE

INTRODUCTION TO THE INTERNATIONAL ENERGY AGENCY

BACKGROUND

The International Energy Agency (IEA) was established in 1974 as an autonomous agency within the framework of the 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.

Collaboration in the research, development and demonstration of new energy technologies has been an important part of the agency's program. The R&D activities within the IEA 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 designates by them). Currently 40 Implementing Agreements are in place, covering different aspects on energy generation, consumption and efficiency technologies.

SOLAR HEATING AND COOLING PROGRAM

One of the Implementing Agreements is the Solar Heating and Cooling Program. Since 1977, its 21 members have been collaborating to advance active solar, passive solar and photovoltaic technologies and their applications in buildings.

The members are:

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

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 management of the program is by an Executive Committee comprised of one representative from each of the 21 members.

The tasks of the IEA Solar Heating and Cooling Program 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 Insulation 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 Materials for Solar Applications
TASK 19	Solar Air Systems

TASK 20	Solar Energy in Building Renovation
TASK 21	Daylight in Buildings

Current Tasks:

TASK 22	Building – Energy Analysis Tools
TASK 23	Optimization of Solar Energy Use in Large Buildings
TASK 24	Active Solar Procurement
TASK 25	Solar Assisted Air Conditioning of Buildings
TASK 26	Solar Combisystems
TASK 27	Performance Assessment of Solar Building Envelope Components
TASK 28	Solar Sustainable Housing
TASK 29	Solar Drying in Agriculture
TASK 30	Solar Cities
TASK 31	Daylighting Buildings in the 21st Century

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 or analyzing solar, low-energy buildings with available and emerging building energy analysis tools. This goal will be pursued by accomplishing the following objectives:

- Assess the accuracy of available building energy analysis tools in predicting the performance o 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; and
- Assess and document the impact (value) of improved building analysis tools in analyzing solar, low-energy buildings, and widely disseminate research results tools, 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.1, Empirical Validation.

CONTENTS

Executive Summary

This is a report on the Iowa Energy Resource Station (ERS) empirical validation project conducted by the International Energy Agency (IEA) Building Energy Analysis Tools Experts Group. The group was composed of experts from the Solar Heating and Cooling (SHC) Program, Task 22, Subtask A. The objective of this subtask has been to develop practical implementation procedures and data for an overall IEA/SHC validation methodology which has been under development since the early 1980s. This report documents empirical validation testing for models related to the thermal behavior of buildings and commercial HVAC equipment installed in typical commercial buildings. Other projects (reported elsewhere) conducted by this group include work on comparative testing, analytical verification, and other empirical validation tests.

Empirical validation is about comparing the performance of building energy simulation software to measured data. Therefore, it must be understood that this exercise is a test of the model, the modeler, the test specification and the experiment itself. Because of the expense of acquiring good measured data and the difficulty of matching experimental setups with typical simulation modeling assumptions, empirical validation experiments have been historically more difficult to do than comparative tests and analytical verification tests. Therefore where empirical validation has been successful, it has only covered a very limited number of test cases.

The rationale for the Iowa ERS validation exercise work is as follows:

- Completion of the Iowa ERS empirical validation study address designer needs for greater confidence in software tools used to design and analyze passive solar buildings, because realistic commercial construction material and practices are considered.

- To properly evaluate the amount of "conventional" energy displaced by passive solar design and active solar mechanical equipment, it must be shown that simulations are properly and accurately modeling "conventional" mechanical equipment.

- The ERS exercises intends to create a suite of test cases for evaluating the capability of building energy analysis tools to model HVAC system and realistic commercial construction buildings.

- This exercise could complement the HVAC BESTEST and the IEA BESTEST comparative test cases because the cases are more realistic and more program assumptions and default values are tested.

In this empirical validation study predictions from several building energy simulation programs were compared to measured results for three separate experiments. The simulation programs participating in the various studies are listed below, preceded by abbreviations used in this report and including the organization that performed the simulation (and its country).

Notation	Program	Implemented by
TRNSYS-TUD	TRNSYS-TUD (modified V.14.2)	University of Dresden Dresden, Germany
PROMETHEUS	PROMETHEUS	Klima System Technik Berlin, Germany
DOE-IOWA	DOE-2.1E	Iowa State University Ames, Iowa
DOE-CIEMAT	DOE-2.1E (V.088)	DOE-CIEMAT Madrid, Spain
IDA-ICE	IDA-ICE (V.2.11.06)	Hochschule Technik + Architektur Luzern, Switzerland

All of the programs listed above participated in the three experiments. The facility where the experiments were done and the three exercises developed are summarized as follows.

ES.1. The Energy Resource Station

The ERS is part of the Iowa Energy Center, a non-profit research and education organization funded through and by utilities operating in Iowa (USA). It has four matched sets of test rooms (Interior, South, East, and West). Each pair of test rooms is identical in construction and exterior exposure. The "A" test room and "B" test room of the matched pair are served by separate HVAC systems.

The rooms can be configured to test a variety of HVAC, control and architectural strategies. It is the only public facility in the United States with the ability to simultaneously test full-scale commercial building systems. Detailed data can be collected on any aspect of mechanical and electrical system behavior.

Because of all this, the ERS offers a unique opportunity to have a highly controlled experimental setting for data collection required for simulation tool validation.

ES.2. FIRST EXERCISE. STEADY CASE. (Constant Air Volume System with Low Internal Loads)

This first exercise was defined as a steady case, in order to check the estimations of the thermal behavior of the building and its sensibility to the weather conditions. For the test conducted, the systems were operated utilizing Constant-Air-Volume Re-Heat (CAVRH). Electric heating coils were used in the rooms to provide terminal heating.

The systems were operated 24 hours per day with a fixed amount of outside air while the thermostats in the test rooms were programmed without a night setback temperature, and the set-points for heating and cooling for the test remained constant.

Internal loads were held constant 24 hours/day in the test rooms..

The test specifications defined an almost constant behavior where the system dynamic variations are mainly caused by weather variations.

Results and Conclusions of the Steady Case

The weather processor is very similar and accurate for every model tested. The main differences are due to different interpretation of the hour. Some programs considered that the data were provided in solar time and made some corrections to consider the local standard time. Differences were very small and can be neglected.

All the models predicted very accurately the different temperatures on the AHU. Considering the accuracy of the predictions for the airflow and the temperatures, the cooling loads must be very well estimated. There are two possibilities to measure the cooling loads: waterside measures and airside measures.

If the airside measures are used, the simulations are very accurate and agree with measures, especially for System A.

If the waterside measures are used possible-measuring errors of 0.362ºC for System A and 0.2ºC for System B might be causing cooling loads disagreements. If this error is assumed, the results of all the models are close together and very similar (both, mean values and fast dynamics) to the cooling loads.

After the room analysis of the simulations, the following general conclusions were reached:

- DOE-IOWA, DOE-CIEMAT and TRNSYS-TUD models accurately estimated mean values and fast dynamics. They only had some problems to estimate the large solar heat gains. Possible solar radiation gains modeling error or erroneous window specifications would be the cause.

- PROMETHEUS and IDA-ICE error on the solar gains are higher. They showed larger errors in the mean values and fast dynamics.

The simulation of all the building and the comparison between measured and predicted results in the following conclusions:

- DOE-IOWA and DOE-CIEMAT estimations were very accurate. They showed mean errors on the reheat energy around 2 W/m^2.

- The other models were accurate enough obtaining mean errors on the reheat energy around 10 W/m^2. This error could be caused by a misinterpretation on the losses through the floor.

The results of all the models were very accurate, especially for DOE-IOWA and DOE-CIEMAT which had errors smaller that 5%. All the models had small problems to estimate the higher cooling loads, probably due to an error on the solar gains simulation or on the window specifications.

ES.3. SECOND EXERCISE. DYNAMIC CASE. (Variable Air Volume System with Scheduled Internal Loads)

This second exercise was defined as a dynamic case, in order to evaluate the estimations of commercial HVAC equipment coupled to the thermal behavior of the building and the weather conditions.

Therefore, this is a realistic case that could be considered a typical office building working with typical commercial HVAC equipment. Only the operating schedule has been changed. The systems were operated 24 hours per day utilizing Variable-Air-Volume Re-Heat (VAVRH). Electric heating coils were used in the rooms to provide terminal heating.

Another feature of this test was the use of thermostat schedules as well as scheduled internal sensible loads for the test rooms. The thermostats in the test rooms were programmed for a night setback temperature. The electric baseboard heaters in the test rooms were programmed to come on during the day to provide a scheduled internal load. This internal load simulates a near-commercial internal load.

The test specifications defined a realistic empirical validation exercise. Together with the specifications, sets of accurate and high quality measurements have been gathered, making this a useful empirical validation.

Results and Conclusions of the Dynamic Case

All the models accurately predicted the temperatures on the AHU. Only the DOE-IOWA model, which showed an error on the input, considered a different supply air temperature based on the test specifications.

As this is a Variable Air Volume case, the supply airflow is one of the most important parameters to be checked. All the models presented errors smaller that 3%, which is negligible.

All the models simulated similar economizer behavior. Only IDA-ICE had some disagreements with the other models. The reason is explained in the modeler report, Section 5.5.

After the room analysis, the following general conclusions were reached:

- All the models had some problems in estimating the first hour after the night setback of the thermostats.

- The predictions are very accurate, especially for TRNSYS-TUD, IDA-ICE and DOE-CIEMAT models.

Some conclusions are also obtained from the analysis of the simulation for the entire building:

- All the models showed an error when the HVAC system was turned on, in the morning.
- The DOE-IOWA model presented a strange behavior in the mornings and had some problems on the dynamics on the evenings. It might have too much thermal inertia.
- DOE-CIEMAT, PROMETHEUS and TRNSYS-TUD models were very accurate. The differences were caused by a calculation error when the HVAC system was turned on.
- IDA-ICE model lightly overestimated the large values.

Simulation results showed good agreement with measured values for all the models. Some of the models were very accurate and made a good prediction of real behavior.

ES.4. THIRD EXERCISE. VERY DYNAMIC CASE (Variable Air Volume System with Variable Internal Loads And Scheduled System)

This third exercise was defined as a very dynamic and realistic case, in order to evaluate the prediction of commercial HVAC equipment working in real conditions, schedules, building and weather.

Therefore, this is a realistic case that could be considered a typical office building working with typical commercial HVAC equipment. Almost every parameter has been established in real conditions (except the uncontrollable occupant behavior on real buildings). Therefore, the results could be interpolated to real life, considering that human beings are non predictable and this has to be assumed in building simulation.

The HVAC systems were scheduled to be off during the unoccupied period. Another feature of this test was the use of thermostat schedules as well as scheduled internal sensible load for the test rooms. The thermostats in the test rooms were programmed for a night setback temperature. The electric baseboard heaters in the test rooms were programmed to come on during the day to provide a scheduled internal load.

Results and Conclusions of the Very Dynamic Case

All the models accurately predicted the temperatures on the AHU. Only the DOE-IOWA model showed an error on the input, similar to previous case.

As a Variable Air Volume case, the supply airflow of the AHU is calculated. All the models presented errors smaller that 3%, which is negligible. Only DOE-IOWA showed larger errors, which is consistent with the error previously commented on the supply air temperature.

An analysis on the cooling loads of the AHU showed how only DOE-IOWA and IDA-ICE models presented non-negligible disagreements. IDA-ICE's dynamical behavior seems to be very constant. The other models, especially TRNSYS-TUD, made good predictions of the cooling load. The errors are within the measuring error band.

The following general conclusions can be made:

- All the models had some problems in estimating the fast dynamics.
- All the models overestimated the influence of the midday setback. All the models might be showing less thermal inertia than reflected in the building.
- The Iowa model overpredicted the supply airflow due to an error on the input of the supply air temperature.
- All the other models accurately predicted the supply airflow and cooling loads.

All the models made very good predictions on the global building energyneeds. The errors are lower that 5% in almost every case. Only PROMETHEUS showed some disagreements and its underestimation is around 10%. In all the other cases, the mean value and fast dynamics were predicted very accurately.

ES.5. FINAL CONCLUSIONS

An empirical validation exercise is a test of the model, the modeler, the test specification, and the experiment itself. As a result of this empirical validation exercise conducted by the IEA SHC Task 22 Subtask A Participants, the participating experts agreed that the following conclusions can be made:

- The Energy Resource Station and the collected data represent an excellent source for empirical validation of building energy analysis tools for commercial buildings and HVAC equipment.
- Agreement of simulation predictions with measurements confirms that test specifications for the project were well defined and are usable for empirical validation. The careful and complete definition of the facility recommends the use of this building for future validation work of HVAC equipment simulation.
- The building energy analysis tools evaluated had good agreement with the measured data. Most of the building energy analysis simulations studied showed small disagreements, similar to the measurement uncertainty. Isolated disagreements are noted previously.
- The building energy analysis tools tested made accurate predictions of the mean values and showed good agreement with fast dynamics. These results should increase confidence in the use of simulation tools to model the types of HVAC systems used in the study.

- After the first round (blind simulation), all participants used the measured data to make legitimate changes to their models to improve their predictions and modeling assumptions.

- The comparison of measured data to the predictions from multiple simulation programs helped improve the models and the experiments. The use of multiple simulation tools is essential in evaluating the validity and accuracy of the measured data. Measurement errors were identified in the first round of the exercises. These errors were fixed for subsequent rounds. (Such as air leakage, outside air duct heating, revision on the measuring method of the cooling loads at the AHU, etc).

As a result of the experiences of the Task 22 Participants in conducting this empirical validation, the following recommendations are made:

- In order to isolate a program's disagreement with measured data, ERS specifications require a more precise definition of window parameters and floor losses (ground temperature and/or floor materials).

- Further empirical experiments are needed to expand the range of variables that can be evaluated versus measured data, and to isolate the validity of specific algorithms applied in the simulation models. The ERS facility is recommended for those new exercises on empirical validation. In some cases, separate laboratory experiments will also be necessary.

- The ERS has the potential for being an excellent facility for empirical validation as long as the integrity of each experiment is maintained through out the testing period.

Future Work: Recommended Additional Cases

The participating experts identified the following new empirical validation test cases as high priority should further empirical validation exercises be undertaken as part of Task 22:

- Daylighting – HVAC Interaction

- Economizer Control

- Hydronic (four pipe)

- Heat Recovery

- Controller Effects

1. Introduction

1.1. Background and Motivation for the Work

The main goal of this project is to assess the accuracy of building energy analysis tools in predicting the performance of a realistic commercial building with real operating conditions and HVAC equipment.

The rationale for the Iowa ERS validation exercise work is as follows:

- The Iowa ERS empirical validation study would address designer needs for greater confidence in software tools used to design and analyze passive solar buildings, because realistic commercial construction material and practices are considered.

- To properly evaluate the amount of "conventional" energy displaced by passive solar design and active solar mechanical equipment, it must be shown that simulations properly and accurately model "conventional" mechanical equipment.

- The ERS exercises will create a suite of test cases for evaluating the capability of building energy analysis tools to model HVAC system and realistic commercial construction buildings.

- This exercise complements the HVAC BESTEST and the IEA BESTEST comparative test cases because these test cases are more realistic and more program assumptions and default values are tested.

1.2. Overview of the Energy Resource Station

The Energy Resource Station (ERS) building is an excellent test facility for conducting empirical validation because it is representative of commercial construction practices and operating conditions.

The ERS is part of the Iowa Energy Center, a non-profit research and education organization funded through and by utilities operating in Iowa. It has four matched sets of test rooms (Interior, South, East, and West). Each pair of test rooms is identical in construction and exterior exposure. The "A" test room and "B" test room of the matched pair are served by separate HVAC systems.

The rooms can be configured to test a variety of HVAC, control and architectural strategies. It is the only public facility in the United States with the ability to simultaneously test full-scale commercial building systems. Detailed data can be collected on any aspect of mechanical and electrical system behavior. Consequently, the ERS offers a unique opportunity to have a highly controlled experimental setting for data collection required for simulation tool validation.

A description of the ERS is provided in Appendix A. This description should be sufficient for a modeler to create an input file for energy simulation.

1.3. Overview of the Testing Conducted

Three tests have been conducted in this facility as an integrated simulation exercise.

The first exercise is an unreal "theoretical" case, where the dynamic variations on the behavior are caused only by the weather conditions. This first exercise can be used to check the model predictions of the U values and the general behavior of the mechanical equipment and strategies (thermostat setpoints, etc.)

Once this first exercise has been passed, the second exercise can be undertaken. This is a partially dynamic case, where the internal loads and the control strategies have been defined in more realistic conditions. This second exercise will evaluate the thermal inertia and the prediction of the real response of the HVAC equipment to thermal variations.

The third and last exercise intends to be a realistic case, where the most real behavior is evaluated. If the model passes this exercise, it would be properly responding to a realistic commercial building situation. The main difference between the test conditions and real life is that baseboard heaters have substituted for the occupant heat gain.

1.4. Overview of the Simulation Tools Used in the Study and Participating Organizations

Five sets of results have been developed with four different computer programs. The organizations and models are identified in Table 1.1. General descriptions of the tools are provided in the modeler report, Section 5.

Table 1.1 Participants

Notation	Program	Implemented by
TRNSYS-TUD	TRNSYS-TUD (modified V.14.2)	University of Dresden Dresden, Germany
PROMETHEUS	PROMETHEUS	Klima System Technik Berlin, Germany
DOE-IOWA	DOE-2.1E	Iowa State University Ames, Iowa
DOE-CIEMAT	DOE-2.1E (V.088)	DOE-CIEMAT Madrid, Spain
IDA-ICE	IDA-ICE (V.2.11.06)	Hochschule Technik + Architektur Luzern, Switzerland

1.5. Analysis Procedure

1.5.1. Programs Results

An output format was defined, so every participant supplied the same hourly output data. The data considered are given in Table 1.2.

Table 1.2 Output Data

GLOBAL REPORT		
Notation	**Description**	**Units**
OADB	Outside Air Dry Bulb	ºC
OAWB	Outside Air Wet Bulb	ºC
DNSR	Direct Normal Solar Radiation	W/m^2
THSR	Total Horizontal Solar Radiation	W/m^2
ZONE REPORT		
Notation	**Description**	**Units**
LOAD	Load without ventilation.	W
ZT	Zone Temperature	ºC
SAF	Supply Airflow	m^3/h
REHEAT	Reheat Energy	W
SYSTEM REPORT		
Notation	**Description**	**Units**
SAF	Supply Airflow	m^3/h
OAF	Outside Air flow	m^3/h
Tin	Temp. of entering cooling coil	ºC
Tout	Temp. of leaving cooling coil	ºC
Tret	Temperature of return air	ºC
Cool En	Cooling coil energy input	W

1.5.2. Hourly Results

As in the ETNA validation exercise, simple statistical measures were used to quantify the differences between the measurements and the predictions. Table 1.3 summarizes the statistical parameters.

Table 1.3 Statistical Parameters

Parameter	Notation	Equation
Mean	MEAN	$\overline{X} = \sum_{t=1}^{N} X_t / N$
Difference	DT	$D_t = X_t - M_t$
Smallest difference	DTMIN	$D_{MIN}=\text{Min}\ (D_t)$
Largest difference	DTMAX	$D_{MAX}=\text{Max}\ (D_t)$
Mean difference	MEANDT	$\overline{D} = \sum_{t=1}^{N} D_t / N$
Absolute mean difference	ABMEANDT	$\lvert\overline{D}\rvert = \sum_{t=1}^{N} \lvert D_t / N\rvert$
Root mean square differ.	RSQMEANDT	$\sqrt{D^2} = \sqrt{\sum_{t=1}^{N} D_{tt}^2 / N}$
Standard Error	STDERR	$\sigma = \sqrt{\frac{1}{N}\sum_{t=1}^{N}(D_t - \overline{D})^2}$

Where X_t: predicted value at hour t.

M_t; measured value at hour t

2. FIRST EXERCISE. STEADY CASE (Constant Air Volume System with Low Internal Loads)

2.1. Description of the Exercise

This section contains information regarding the operating parameters and conditions used for a CAV test conducted at the Iowa Energy Center's Energy Resource Station as part of the empirical validation study for the International Energy Agency Task 22. The test was conducted over a five-day period from June 18-22,1999.

For the test conducted, the "A" and "B" systems were operated in an identical manner and were operated utilizing Constant-Air-Volume Re-Heat (CAVRH). Electric heating coils were used in the rooms to provide terminal heating. The air handling units were operated 24 hours per day with a fixed amount of outside air. The chiller was available throughout the test.

The thermostats in the test rooms were programmed without a night setback temperature, and the set-points for heating and cooling for the test remained constant. Baseboard heaters were used to simulate internal loads in the test rooms for this test. The thermostats in the rest of the ERS were programmed for a constant set-point schedule. Hence, the temperature in the spaces adjacent to the test rooms remained fairly constant during the test. Actual adjacent space temperature data are provided in the file 990618adjtemp.dat. This file contains hourly temperature data.

The air handling units were operated with continuous fan operation and a fixed amount of outdoor air. The chiller was available throughout the test. The thermostats in the test rooms were scheduled as shown in Table 2.2. The baseboard heaters were used to provide internal sensible loading to the rooms. Table 2.1 gives the values of baseboard heaters' power for different stages.

The lights in the test rooms were turned off. The HVAC system that serves the remaining spaces at the ERS (i.e. computer room, classroom, etc.) was run to provide nearly constant temperature conditions in these spaces. The thermostats in the spaces adjacent to the test rooms were set at 22.7 °C.

2.1.1. Run Period and General Weather Conditions

This item is used to specify the initial and final dates of the desired simulation period and also the general conditions and location of the ERS facility.

- The dates for this test are June 18, 1999 through June 22,1999.
- Weather data for Ankeny, Iowa is organized into a TMY format. In this file, the measured data for the dates previously specified are included. This file is called "Ankeny.ia1" and is attached to this report.
- Building Location
  - LATITUDE: 41.71 degrees North
  - LONGITUDE: 93.61 degrees West
  - ALTITUDE: 938.0 feet (285.9 m)
  - TIME-ZONE: 6, central time zone in U.S.
  - DAYLIGHT-SAVINGS: YES

2.1.2. Test Rooms Operation and Control Parameters

The following conditions apply to all of the test rooms. These conditions do not apply to the rest of the building where occupants may be present and lighting and window shading devices are used.

2.1.2.1. Internal Loads and General Room Conditions

Inside each test room is installed a baseboard heater. Those baseboard heaters were used to simulate internal loads in the test rooms for this test (additional information about the baseboard heaters is provided in the Appendix C). The first stage of the baseboard in each room was on during the test.

Table 2.1 Baseboard heater power for different stages

Rooms	Stage 1 (kW)	Stage 2 (kW)
East A	0.900	0.880
East B	0.875	0.845
South A	0.885	0.875
South B	0.870	0.875
West A	0.855	0.845
West B	0.885	0.885
Interior A	0.865	0.880
Interior B	0.915	0.900

The first stage of these baseboards was on during 24 hours per day during the test period. Therefore, the East A room had an internal load of 0.900 kW.

Besides the baseboard heaters, other general room characteristics must be considered:

- No lights or miscellaneous equipment other than the baseboard heaters.
- No shading device on windows.
- No infiltration.

2.1.2.2. Room HVAC specifications

Each test room has its own thermostat and some HVAC specifications can be considered.

- **Thermostat Schedule**

The set point value is the same for all test rooms. These values were used for both test rooms.

- Design heat temperature: 22.2 °C
- Design cool temperature: 22.7 °C
- Heat temperature schedule: see Table 2.2
- Cool temperature schedule: see Table 2.2
- Dead-ban: 1.7 °C

Table 2.2 Set point temperature, internal loads, and AHU fan schedules

Hour	Cooling set-point temperature (°C)	Heating set-point temperature (°C)	Internal loads (stage of baseboard heat)	AHU fan
1-24	22.8	22.2	1	ON

- **Room Airflow and Reheat Specifications**

The airflow rates are constant and were specified for each test room.

- Exterior test rooms (east, south and west): max 1019 m^3/hr, min 1019 m^3/hr
- Interior test rooms: max 459 m^3/hr, min 459 m^3/hr
- The zone heat source installed are: 2 stage electric, max 3.34 kW (1.67 kW/stage) for exterior rooms and max 2 kW (1 kW/stage) for interior rooms

2.1.3. Air Handling Unit Operation and Control

Both AHU (A and B) were working in the same conditions to supply air to the four set of rooms.

2.1.3.1. Set Points and System Controls

The air handling system parameters were specified as follows.

- Supply air temperature: max 29.4 °C, min 15.6 °C
- Heating schedule: NOT available
- Cooling schedule: 24 hours available
- Cool control: supply air set point, 13.3 °C after the fan
- Preheat: NOT available
- Humidity control: NOT available
- Economizer: NOT available
- Outside air control: NOT available

2.1.3.2. System Air and Fans

System airflow rates were specified as follows.

- Supply airflow: max 6116 m^3/hr

- Return air path: Plenum
- Constant outside airflow: 680 m^3/hr
- Outside air control: NOT available
- Duct air loss: None
- Duct heat gain: 0.3 °C (increase)

The air handling unit fans were specified as follows.

- Supply air static pressure: 1.4 inch H_2O
- Fan schedule: On 24 hours per day
- Supply Fan control: Duct static pressure of 1.4 inch H_2O
- Return Fan control: 90 % of supply fan speed
- Motor placement: In-Air flow
- Fan placement: Draw-Through

2.2. Participating Organizations

Five sets of results were developed with four different computer programs. Participating organizations and models are identified in Table 2.3.

Table 2.3 Participants

Notation	**Program**	**Implemented By**	**Date of simulation/round**
TRNSYS-TUD	TRNSYS-TUD (modified V.14.2)	University of Dresden Dresden, Germany	March 2000/2nd round
PROMETHEUS	PROMETHEUS	Klima System Technik Berlin, Germany	October 1999/1st round
DOE-IOWA	DOE-2.1E	Iowa State University Ames, Iowa	March 2000/2nd round

DOE-CIEMAT	DOE-2.1E (V.088)	DOE-CIEMAT Madrid, Spain	June 2000/3rd round
IDA-ICE	IDA-ICE (V.2.11.06)	Hochschule Technik + Architektur Luzern, Switzerland	June 2000/3rd round

2.3. Comparison between A and B Room Type Measurements

The ERS has four sets of identical test rooms. Each pair of test room allows simultaneous, side by side comparison testing of many types of HVAC systems. For the test conducted, the pair of rooms, called "A" and "B", systems were operated in an identical manner. This should cause identical results for the A and B rooms and system.

Before presenting the measured data and the model predictions, the errors assumed by the measurements must be presented.

2.3.1. Systems Comparison

The first step to compare both room types measures is to analyze the systems behavior. The reason is: if the central system is supplying the air to each room at different temperatures, this should cause different reheat needs in each test room of a pair.

The parameters used for the analysis are:

- A SYSTEM. Mean value for the A system.
- B SYSTEM. Mean value for the B system.
- B/A MEAN VALUE: Relation between both mean values

The cooling energy has not been measured directly. It can be calculated using two different methods:

- WATERSIDE METHOD: Measurements of the water flow and the temperature entering and leaving the cooling coil are used to calculate the cooling energy.

- AIRSIDE METHOD: Same measurements, but in the airside. As for the waterside method, the calculations would be very simple, only the supply airflow and the temperatures entering and leaving the cooling coil are needed. By using this method only the sensible cooling energy is calculated.

The waterside method should be more accurate that the airside method. When the air measurements are used, the latent heat is not considered and the airflow measurements are less accurate.

As Table 2.4 shows, there are measuring differences on the cooling energy. Small differences exist for all the parameters, except for the cooling energy. The differences of each measurement on the airside are always around 1%, but the waterside measurements showed differences around 10%. But all the other measurements seem to be very similar. If the supply airflow and the entering and leaving temperatures are similar, the cooling energy should be also similar.

Table 2.4 Comparison between system measurements

	SUPPLY AIR	TEMP ENTERING COIL	TEMP LEAVING COIL	OUTSIDE AIR	TEMP RETURN AIR	COOLING waterside
	m^3/h	ºC	ºC	m^3/h	ºC	W
A SYSTEM	3449	22.35	11.69	678.4	22.04	11589.4
B SYSTEM	3458	22.30	11.90	679.0	22.03	12767.4
B/A MEAN VALUE	100.28%	99.77%	101.81%	100.09%	99.93%	110.16%
MEAS. UNCERTAINTY	2%	1%	1%	2%	1%	8%

A different way to calculate the cooling energy is as follows: using the airside. In that case, the latent loads are not considered. Table 2.5 shows the results of the cooling energy calculated by the airside and the waterside.

Table 2.5 Cooling energy comparison

	COOLING waterside	COOLING airside
	W	W
A SYSTEM	11589.4	12638.8
B SYSTEM	12767.4	12361.0
B/A MEAN VALUE	110.16%	97.80%

This table is expressing two interesting aspects:

- The cooling energy calculated at the waterside should be larger than that calculated at the airside, because it considers both the sensible and the latent loads. The latent loads are probably negligible in this case because there are not humidity gains in the internal loads. Although the latent loads are very small they might not be zero. As Table 2.5 shows, the results obtained by using the waterside are smaller that the ones calculated from the airside, which is theoretically impossible.

- The results obtained by using the airside are more similar than the ones obtained using the waterside. The ranges of the discrepancies using the airside are similar to the observed for the supply airflow and the air temperatures.

The errors in the measurements at the waterside are probably larger that at the airside. For a better understanding of the reason, it is interesting to analyze the method used for the regulation of the cooling energy at the AHU. It is made by a three-way mixing-valve, which maintains a constant supply water flow and a variable entering water temperature. To calculate the cooling energy, the temperature differences between entering and leaving water temperatures has been used. Those temperatures have been measured separately and then subtracted. Therefore, the uncertainties of the measures have been doubled. The mean value of the temperature differences is around 3ºC. Consequently, an uncertainty band of

0.1ºC on the measurements, could be the reason for a cooling error calculation error of 7%. Therefore, the uncertainty in measurements at the waterside is 8%, which is an important value.

Conclusions**:** If the airside measurements are accurate enough, both room types should demand similar reheat energy, but if the waterside are more accurate that the airside, differences as big as 10% have to be accepted.

Possible measuring errors in the airside of 0.1ºC could be causing errors of 7%.

2.3.2. Rooms Comparison

2.3.2.1. Interior Room

Table 2.6 shows the comparison between the A and B type interior room temperatures, supply airflow and reheat energy.

Table 2.6 Interior rooms parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	22.2	458.7	438.6
B ROOM	22.2	458.7	446.1
B/A MEAN	100.0%	100.0%	101.7%
MEAS. UNCERTAINTY	1%	2%	5%

There are relatively small differences between both room types (less that 2%). They are always within the uncertainty band.

These discrepancies could be also caused by an air leakage in the B room. Therefore, not all the air is supplied into the room and part of it goes directly into the plenum. This could cause small differences on the behavior of the room but not on the energy balance.

The air supplied into the room might be too hot for an appropriate thermal comfort but this aspect (human comfort) is not being analyzed in this exercise. The global energy balance of the room does not have large differences; there is only a small difference on the airflow and the air changes per hour that could be the reason for the 4% disagreement.

To compare and analyze possible measuring errors one hypothesis will be assumed:

1. The A and B rooms are similar and are being operated in identical manner.

Assuming this hypothesis, the energy supplied to the room should be the same for both of them. The cooling energy supplied to the room will be:

$$\text{Energy supplied} = m \cdot \left(h_{room} - h_{entering}\right)$$

Where, m: total air mass supplied into the room.

h_{room}: air enthalpy of the room.

$h_{entering}$: air enthalpy supplied into the room.

Considering:

1. Same air density for both rooms (1.2 kg/m^3).
2. Neglecting the humidity content variations.

A good parameter to compare the energy supplied into the room would be:

$$ES = c_p \rho Q \left(t_{room} - t_{entering}\right)$$

ES; is the cooling energy supplied into the room. As the measurements and the results have been made in a timestep of 1 hour, the units can be W·h or Joules. In this case, the Wh have been chosen.

C_p; specific heat of the air (1 kJ/kgºC)

ρ; Air density for both rooms (1.2 kg/m^3).

Q; Supply air flow (m^3/h)

t_{room}: temperature of the room.

$t_{entering}$: temperature of the air entering the room.

Both rooms should demand the same cooling load. Table 2.7 shows the leaving reheat coil temperature, the supply airflow and the energy factor for both rooms.

Table 2.7. Interior room cooling energy

	SUPPLY AIR FLOW	ROOM TEMP.	ENTERING TEMP.	E.S.
	m^3/h	ºC	ºC	Wh
A ROOM	458.7	22.2	14.6	1173.8
B ROOM	458.7	22.2	14.8	1133.5
B/A MEAN	100.0%	100.0%	101.8%	96.6%
MEAS. UNCERTAINTY	2%	1%	1%	

The differences are only 3%. It can be concluded that those differences are not representing large thermal behavior differences.

Conclusions: The thermal behavior of both room types is practically identical.

2.3.2.2. East Room

As for the interior room, results of the comparison for the A and B rooms are shown on Table 2.8.

Table 2.8. East rooms parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	22.3	1019.4	1387.4
B ROOM	22.3	1019.4	1190.9
B/A MEAN	100.0%	100.0%	85.8%
MEAS. UNCERTAINTY	1%	2%	5%

There is no difference between both room types in the indoor temperature or the supply airflow. But there are important discrepancies on the reheat energy.

Figure 2.1 shows the reheat energy for both rooms. It is clearly presented how the reheat measured in the B room is always bigger that for the A room. This could be caused by the adjacent room temperatures, the supply air temperature from the air-handling unit, air leakage or construction differences.

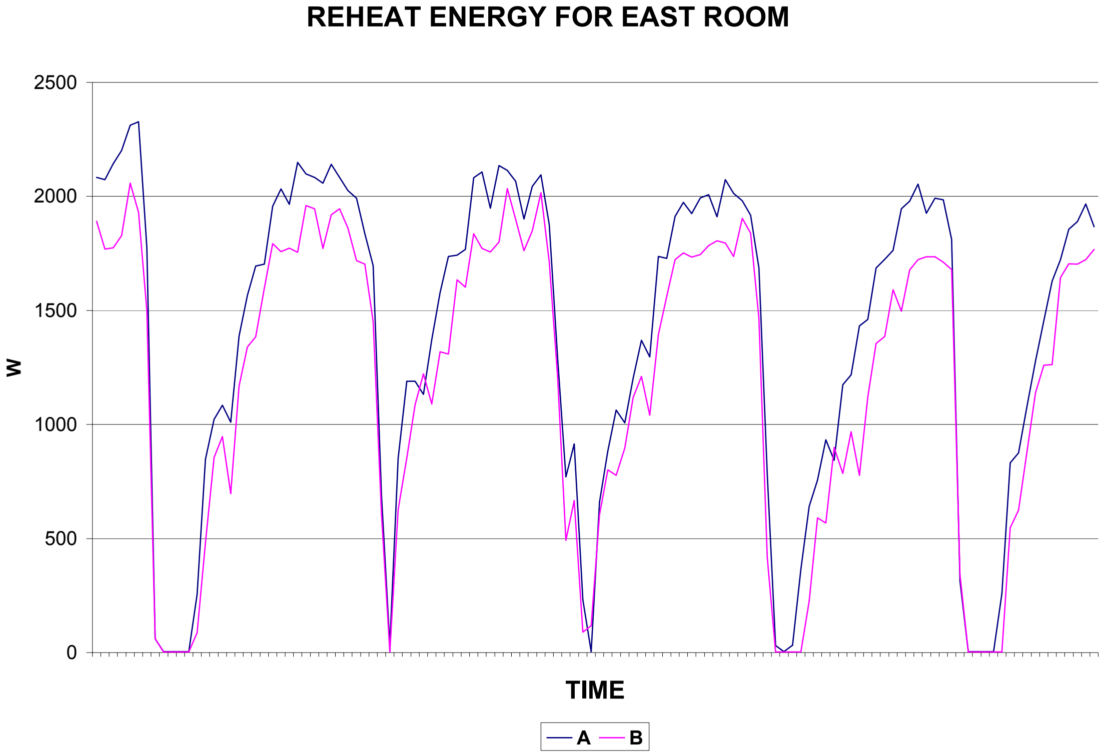

Figure 2.1 Reheat energy for east room

The parameter defined in the previous item, called ES will be used again to confirm if both room types are having the same thermal behavior. Table 2.9 shows this parameter and the disagreements are around 6%, which could be justified by the adjacent room temperatures or measuring errors.

Table 2.9. East rooms cooling energy

	SUPPLY AIR FLOW	ROOM TEMP.	ENTERING TEMP.	E.S.
	m^3/h	ºC	ºC	Wh
A ROOM	1019.4	22.3	15.8	2209.0
B ROOM	1019.4	22.3	15.4	2334.9
B/A MEAN	100.0%	100.0%	97.7%	105.7%
ERROR UNCERTAINTY	2%	1%	1%	

Conclusions: The B EAST ROOM demanded 15% more reheat energy than the A room. The thermal behavior of both room types are almost identical.

2.3.2.3. South Room

As for the East room, results of the comparison for the South A and B rooms are shown on Table 2.10.

Table 2.10. South rooms parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	22.2	1019.4	1733.6
B ROOM	22.2	1019.5	1758.9
B/A MEAN	100.0%	100.0%	101.5%
MEAS. UNCERTAINTY	1%	2%	5%

The measurement differences are very small. Both rooms have almost the same thermal behavior. This is confirmed by the ES parameter. Table 2.11 lists the supply air flow, room temperature and entering air temperature for room A and for room B.

Table 2.11. South rooms cooling energy

	SUPPLY AIR FLOW	ROOM TEMP.	ENTERING TEMP.	ES
	m^3/h	ºC	ºC	Wh
A ROOM	1019.4	22.2	16.8	1836.1
B ROOM	1019.5	22.2	17.1	1738.7
B/A MEAN	100.0%	100.0%	101.7%	94.7%
MEAS. UNCERTAINTY	2%	1%	1%	

Conclusions: The measurement disagreements are only 1% for the reheat energy and 5% for the thermal behavior. Differences can be neglected.

2.3.2.4. West Room

As for the West room, results of the comparison for the A and B rooms are shown in Table 2.12.

Table 2.12. West rooms parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	22.2	1019.4	1627.2
B ROOM	22.2	1019.4	1500.6
B/A MEAN	100.0%	100.0%	92.2%
MEAS. UNCERTAINTY	1%	2%	5%

There is no difference between both rooms in the indoor temperature or the supply airflow. The reheat energy disagreements could be caused by differences on the air temperature provided by the AHU or by differences on the temperature of the adjacent rooms. To confirm if this is true, the ES parameter is used. Table 2.13 summarizes the air flow rate, room temperature, and entering air temperature for both West rooms.

Table 2.13. West rooms cooling energy

	SUPPLY AIR FLOW	ROOM TEMP.	ENTERING TEMP.	ES
	m^3/h	ºC	ºC	Wh
A ROOM	1019.4	22.2	16.5	1952.4
B ROOM	1019.4	22.2	16.3	2006.7
B/A MEAN	100.0%	100.0%	99.0%	102.8%
MEAS. UNCERTAINTY	2%	1%	1%	

This table shows how although the differences on the reheat energy are relatively large, the mean value is very small, so the differences are not so important when the thermal behavior of the room is considered.

Conclusions: For the WEST ROOMS, there are measuring disagreements of the reheat energy of 7% (which is out of the uncertainty band) but the thermal behavior is practically identical.

2.4. Comparison Between Experimental Results and Simulation Results

2.4.1. Weather Data

The weather data have been delivered to every user in TMY format. The simulation programs have different weather processors, so those data could be modified or misinterpreted by the program.

The first step, before analyzing the predictions on the HVAC system, was to check that every program is considering the same weather conditions. This evaluation must be done only for the first case. Once the weather processors are proved to be similar and that they do not modify the weather data, it does will not be checked for the other exercises.

As this data has been provided to all the modelers as a true value, no uncertainty on measurements has been considered.

2.4.1.1. Air Dry Bulb Temperature

All the programs considered almost the same dry bulb temperature. The mean values are almost the same, so they are estimating the same overall value. The standard errors are very small, so it can be assumed that all them are considering the same temperatures at each hour. Table 2.14 gives the statistical comparison of dry bulb temperature from different participants.

Table 2.14. Statistical comparison of the dry bulb temperature (ºC)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEAS.
dtmin	-0.30	-0.30	0.00	-1.30	-1.30	
dtmax	0.30	0.30	0.00	0.85	0.87	
meandt	0.01	0.01	0.00	-0.03	-0.03	
min	11.70	11.70	11.90	12.00	12.01	11.90
max	28.30	28.30	28.40	28.30	28.29	28.40
mean	21.15	21.15	21.14	21.11	21.11	21.1
abmeandt	0.15	0.15	0.00	0.33	0.33	
rsqmeandt	0.18	0.18	0.00	0.44	0.44	
stderr	0.18	0.18	0.00	0.44	0.44	
stderr/mean	0.01	0.01	0.00	0.02	0.02	

2.4.1.2. Outside Air Wet Bulb Temperature

The wet bulb temperature data were not included in the weather file, but other humidity parameters like the dew point were provided. Each program calculated the wet bulb temperature, so if they are doing it right, they should obtain the same wet bulb temperature for each hour. The measured wet bulb has been calculated using equations recommended by the ASHRAE Handbook of Fundamentals, considering the actual pressure at each time. Table 2.15 gives statistical comparison of wet bulb temperature from all participants except from IDA-ICE.

Table 2.15. Statistical comparison of wet bulb temperature (ºC)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	ASHRAE
dtmin	-0.26	-0.26	-0.08	-0.88	
dtmax	0.32	0.32	0.14	1.17	
meandt	0.02	0.02	0.02	-0.09	
min	10.00	10.00	10.10	10.11	10.04
max	22.20	22.20	22.00	21.79	21.97
mean	17.06	17.06	17.07	16.96	17.0
abmeandt	0.14	0.14	0.04	0.18	
rsqmeandt	0.16	0.16	0.05	0.26	
stderr	0.16	0.16	0.05	0.25	
stderr/mean	0.01	0.01	0.00	0.01	

For every program the mean differences and the standard errors are very small, so all them are predicting almost the same wet bulb temperatures. The differences between the DOE-2 weather processor (DOE-IOWA and DOE-CIEMAT models) and the TRNSYS-TUD one versus the measurements are similar to those observed in the dry bulb temperature. Those differences are negligible.

2.4.1.3. Direct Normal Solar Radiation

All the programs consider almost the same direct normal radiation. The data generated by each program have some small differences, as shown in Table 2.16

Table 2.16. Statistical comparison of direct normal solar radiation (W/m^2)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEAS.
dtmin	-254.26	-254.26	-82.50	0.00	-257.94	
dtmax	118.47	118.47	79.66	0.00	183.33	
meandt	-0.12	-0.12	-2.35	0.00	-0.12	
min	0.00	0.00	0.00	0.00	0.00	0.00
max	825.90	825.90	793.90	843.33	745.38	843.33
mean	100.54	100.54	98.30	100.66	100.53	100.7
abmeandt	18.65	18.65	9.19	0.00	31.13	
rsqmeandt	41.33	41.33	19.94	0.00	61.92	
stderr	41.33	41.33	19.80	0.00	61.92	
stderr/mean	0.41	0.41	0.20	0.00	0.62	

The mean values are almost the same; the biggest difference is given by PROMETHEUS and it is just 2%. Nevertheless, the fast variations are acceptably predicted by the PROMETHEUS program, as its standard error is not very high.

Both DOE-2 models and the IDA-ICE programs had a bigger standard error than TRNSYS-TUD. As it has already explained, the weather data were provided in TMY format. This format requires the data by solar standard time. The DOE-2 and IDA-ICE programs might be considering the difference between the solar time and the local standard time.

The input data were provided at solar time and the programs calculated the radiation at standard time. Figure 2, where is presented the direct solar radiation for one day (as an example, June 18th) confirms this point. For this day, the difference between the solar time and the local standard time is 15 minutes. When it is 1 hour at solar time, the local standard time is 1:15.

As Figure 2.2 shows, besides the time correction, both weather processors are modifying the solar radiation and re-calculating it.

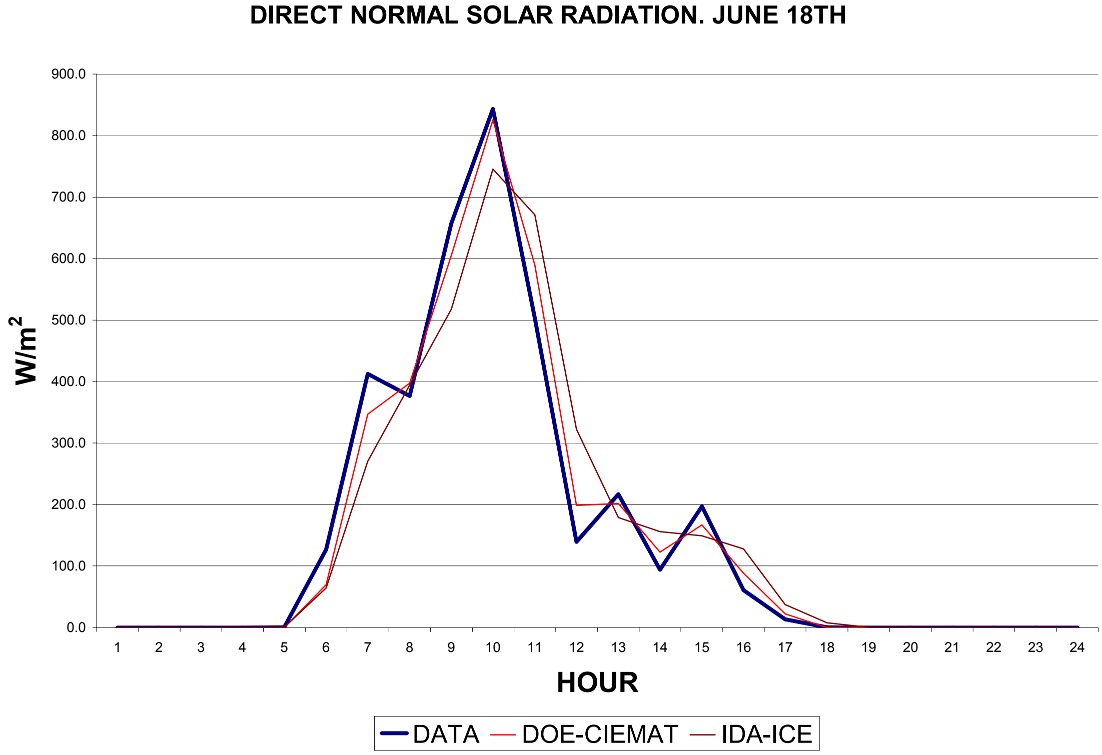

Figure 2.2 Comparison of direct normal solar radiation for June 18th

The TRNSYS-TUD model uses directly the data provided by the Energy Resource Station without any modification or correction.

2.4.1.4. Global Solar Radiation

The global solar radiation is almost the same for every model. The DOE-IOWA, DOE-CIEMAT and IDA-ICE models have a small error on the overall value, but it is negligible. The error is very similar to the one observed for the direct normal solar radiation. Table 2.17 gives statistical summaries of global solar radiation from all participants. Figure 2.3 shows the global solar radiation.

Table 2.17 Statistical summaries of global solar radiation (W/m^2)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEAS.
dtmin	-158.67	-158.67	-0.04	0.00	-229.53	
dtmax	77.91	77.91	0.04	0.00	167.78	
meandt	-0.02	-0.02	0.00	0.00	-4.87	
min	0.00	0.00	0.00	0.00	0.00	0.00
max	775.50	775.50	801.90	801.94	733.51	801.94
mean	213.95	213.95	213.98	213.97	209.11	214.0
abmeandt	19.07	19.07	0.02	0.00	39.56	
rsqmeandt	30.78	30.78	0.02	0.00	63.52	
stderr	30.78	30.78	0.02	0.00	63.34	
stderr/mean	0.14	0.14	0.00	0.00	0.30	

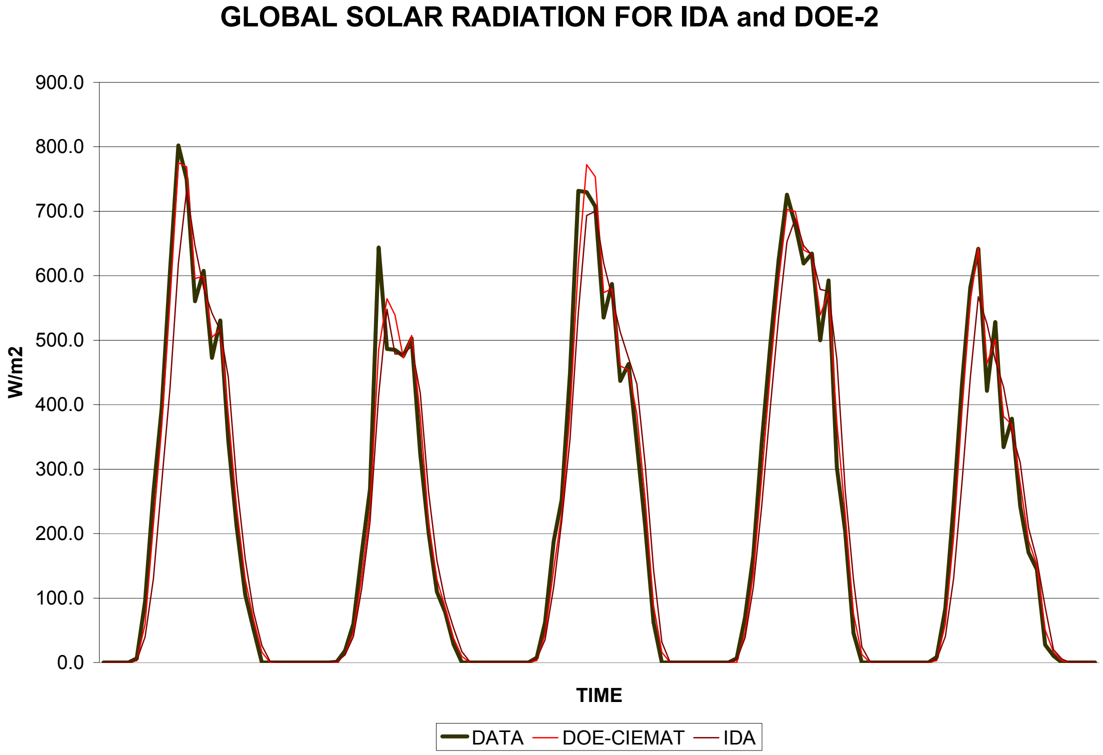

Figure 2.3 Comparison of global solar radiation

2.4.2. AHU A System Data

Once the measurements have been studied, the simulation results for the Air Handling Units will be compared with the measurements.

For a better analysis, two kinds of parameters / simulation results will be checked: the test defined parameters and the non-specified parameters, which are consequences of the thermal and system behavior.

2.4.2.1. Supply Airflow: Test Defined Parameter

The supply airflow has been defined as a constant air volume of 3516 m^3/h. All the programs simulated this condition accurately, as shown in Table 2.18.

Table 2.18 Statistical summary of supply air flow rates (m^3/h)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERTAINTY
dtmin	34.69	33.69	33.69	35.19	34.18		**2%**
dtmax	172.04	171.04	171.04	172.54	171.54		
meandt	68.31	67.31	67.31	68.81	67.80		
min	3517.00	3516.00	3516.00	3517.50	3516.49	3344.956	66.9
max	3517.00	3516.00	3516.00	3517.50	3516.49	3482.313	69.6
mean	3517.00	3516.00	3516.00	3517.50	3516.49	3448.7	69.0
abmeandt	68.31	67.31	67.31	68.81	67.80		
rsqmeandt	69.87	68.89	68.89	70.36	69.37		
stderr	14.67	14.67	14.67	14.67	14.67		
stderr/mean	0.00	0.00	0.00	0.00	0.00		
MEAN%	2%	2%	2%	2%	2%		

All the programs presented the same error, 2%, which is just the supply airflow uncertainty in measures. A small disagreement between the set point specified for the system and the actual airflow is detected. It was suppose to be 3516 m^3/h and the measurements obtained are 3449 m^3/h. Besides this, the airflow was suppose to be constant but the real AHU was not able to provide a constant airflow, as shown in Figure 2.4.

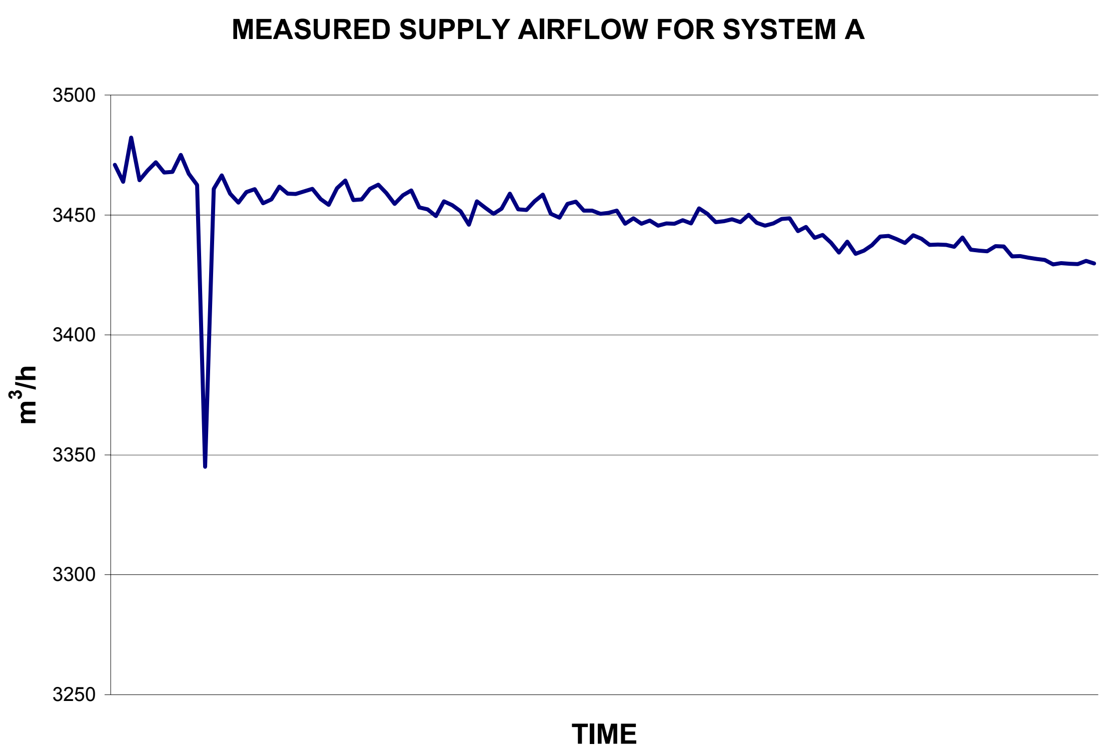

Figure 2.4 Supply air flow-rate for system A

Conclusions: The differences between measurements and simulations can be neglected. The discrepancies with measurements are caused by a non-ideal behavior of the real AHU.

2.4.2.2. Outside Airflow. Test Defined Parameter

The outside airflow was defined as constant at 680 m^3/h. All the models considered accurately this value.

The disagreements between measurements and simulations are caused by the differences between actual controls, where the control systems are not perfect, and theoretical controls. Those differences can be neglected. Table 2.19 gives statistical summary of outside airflow rate from different participants.

Table 2.19 Statistical summary of outside airflow rate (m^3/h)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERTAINTY
dtmin	-1.40	-15.66	-14.25	-14.25	-13.84		**2%**
dtmax	159.28	145.02	146.43	146.43	146.84		
meandt	14.43	0.17	1.58	1.58	2.00		
min	692.85	678.59	680.00	680.00	680.41	533.572	10.7
max	692.85	678.59	680.00	680.00	680.41	694.25	13.9
mean	692.85	678.59	680.00	680.00	680.41	678.4	13.6
abmeandt	14.46	5.06	4.87	4.87	4.90		
rsqmeandt	20.21	14.14	14.23	14.23	14.28		
stderr	14.14	14.14	14.14	14.14	14.14		
stderr/mean	0.02	0.02	0.02	0.02	0.02		
MEAN%	2%	0%	0%	0%	0%		

The behavior is very similar to the supply airflow. So as in previous case, both, the mean differences and the standard errors can be neglected.

Conclusions: The mean differences and the standard errors can be neglected, as they are within the uncertainty band.

2.4.2.3. Temperatures

Three different temperatures have been measured at the Air Handler Unit: leaving cooling coil temperature, return air temperature and entering cooling coil temperature

- **Supply Air Temperature. Test Defined Parameter**

The supply air temperature was defined as constant at 13.3ºC after the fan. The temperature leaving the cooling coil must be the supply temperature minus the increase caused by the supply fan. As table 2.20 shows, the mean temperature measured was 11.7ºC. All the models estimated accurately this temperature.

Table 2.20. Leaving cooling coil temperature (ºC)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERT.
dtmin	-0.32	-0.52	-0.02	-0.52	-0.51		**1%**
dtmax	1.22	1.22	1.62	1.12	1.13		
meandt	0.33	0.18	0.61	0.11	0.12		
min	11.90	11.70	12.30	11.80	11.81	10.68	0.11
max	12.10	11.90	12.40	11.80	11.81	12.32	0.12
mean	12.02	11.86	12.30	11.80	11.81	11.7	0.12
abmeandt	0.34	0.21	0.61	0.18	0.18		
rsqmeandt	0.40	0.28	0.65	0.23	0.24		
stderr	0.22	0.22	0.20	0.21	0.21		
stderr/mean	0.02	0.02	0.02	0.02	0.02		
MEAN%	3%	2%	5%	1%	1%		

The standard errors are very low for all the models, and can be neglected. Some of them (IDA+ICE and TRNSYS-TUD) estimated this temperature as accurate as the measurement.

Conclusions: The models are accurately estimating this temperature.

- **Return Air Temperature. Non Test Defined Parameter**

This parameter has not been defined in the exercise as an input value. The return air temperature must be a corrected mean value between the different rooms' temperature, increased by the return fan heat plus heat gain through the plenum (lights, roof, etc.). If the room temperatures are accurately predicted, this return air temperature must be also accurately predicted. Table 2.21 shows the exactness of these predictions.

Table 2.21 Statistical summary of return air temperature (ºC)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERT.
dtmin	0.74	-0.26	0.54	0.04	-0.23		**1%**
dtmax	2.99	1.89	2.69	2.20	1.92		
meandt	1.32	0.25	0.96	0.46	0.19		
min	23.20	22.20	23.00	22.50	22.23	20.31	0.2
max	23.60	22.40	23.00	22.51	22.24	22.46	0.2
mean	23.37	22.29	23.00	22.50	22.23	22.0	0.2
abmeandt	1.32	0.31	0.96	0.46	0.28		
rsqmeandt	1.38	0.41	1.01	0.56	0.37		
stderr	0.37	0.33	0.32	0.33	0.32		
stderr/mean	0.02	0.01	0.01	0.01	0.01		
MEAN%	6%	1%	4%	2%	1%		

The differences are mainly caused by overestimation on the heat of the return fan. Possible disagreements due to differences on the room temperatures will be analyzed in detail for each room.

The largest mean error is around 5%, given by DOE-IOWA and PROMETHEUS models, and the largest standard error is given by the DOE-IOWA model. Both errors are very small.

Conclusions: The errors can be neglected for all the models.

- **Entering Cooling Coil Air Temperature. Non Test Defined Parameter**

The entering cooling coil temperature is a consequence of the mixture of the return air and the outside air. If the outside and return air temperatures are properly predicted and so is the airflow, this temperature should be simulated accurately. Table 2.22 gives statistical summaries of cooling coil entering air temperature.

Table 2.22 Statistical summary of entering cooling coil temperature (ºC)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERT.
dtmin	0.05	-0.83	-0.23	-0.39	0.05		**1%**
dtmax	2.18	1.28	1.88	1.76	2.08		
meandt	0.58	-0.29	0.28	0.16	0.47		
min	21.00	20.20	20.90	20.55	21.06	20.624	0.21
max	24.50	23.50	24.00	24.07	24.22	23.566	0.24
mean	22.94	22.07	22.64	22.51	22.82	22.4	0.22
abmeandt	0.58	0.40	0.33	0.29	0.47		
rsqmeandt	0.69	0.46	0.46	0.39	0.56		
stderr	0.37	0.36	0.36	0.35	0.31		
stderr/mean	0.02	0.02	0.02	0.02	0.01		
MEAN%	3%	-1%	1%	1%	2%		

The greatest error is given by the DOE-IOWA model. This is caused by the overestimation of the return air temperature.

All the standard errors are very low and similar, most being within the uncertainty band.

Conclusions: The errors can be neglected for all the models.

2.4.2.4. Cooling Energy calculated at the Waterside. Non Test Defined Parameter

The cooling energy is a function of the airflow and the entering and leaving cooling coil temperatures. Those parameters are accurately predicted, so the cooling energy should be accurately estimated. There is only a possible difference caused by the latent heat exchange. This difference should not be very large and has not been controlled in this test.

It has been explained previously (see comparison between A and B system type) the two possible methods to calculate the cooling load (airside and waterside calculations).

Table 2.23 shows the results obtained considering the waterside. All the models overestimated the cooling loads. The closest estimation is given by the TRNSYS-TUD model and is 15% overestimated. This table shows a possible error or misinterpretation on the cooling loads measurements, which are too low at every hour. Figure 2.5 shows the simulations and measurement results for the cooling loads.

Table 2.23 Statistical summary of water-side energy balance (W)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERT.
dtmin	1353.1	910.2	973.2	-46.3	1804.2		8%
dtmax	4627.1	4569.1	4710.1	3765.3	5250.4		
meandt	2687.0	2482.5	2751.0	1775.4	3395.1		
min	10707.0	10132.0	10197.0	9137.3	10949.2	8493.6	679.48
max	17792.0	17614.0	17563.0	16758.5	18272.6	14207.5	1136.60
mean	14276.4	14071.8	14340.4	13364.8	14984.5	11589.4	927.15
abmeandt	2687.0	2482.5	2751.0	1776.2	3395.1		
rsqmeandt	2755.3	2575.1	2841.3	1923.6	3460.9		
stderr	609.7	684.4	710.5	740.4	671.7		
stderr/mean	0.04	0.05	0.05	0.06	0.04		
MEAN%	23%	21%	24%	15%	29%		

As Figure 2.5 shows, the measurements (the uncertainties are also showed) have the same behavior that the simulations but the mean value is different. If 2618 W is added to the measured cooling load, the results are much better, as Table 2.24 and Figure 2.6 shows. This value is three times higher that the uncertainty of the measurements.

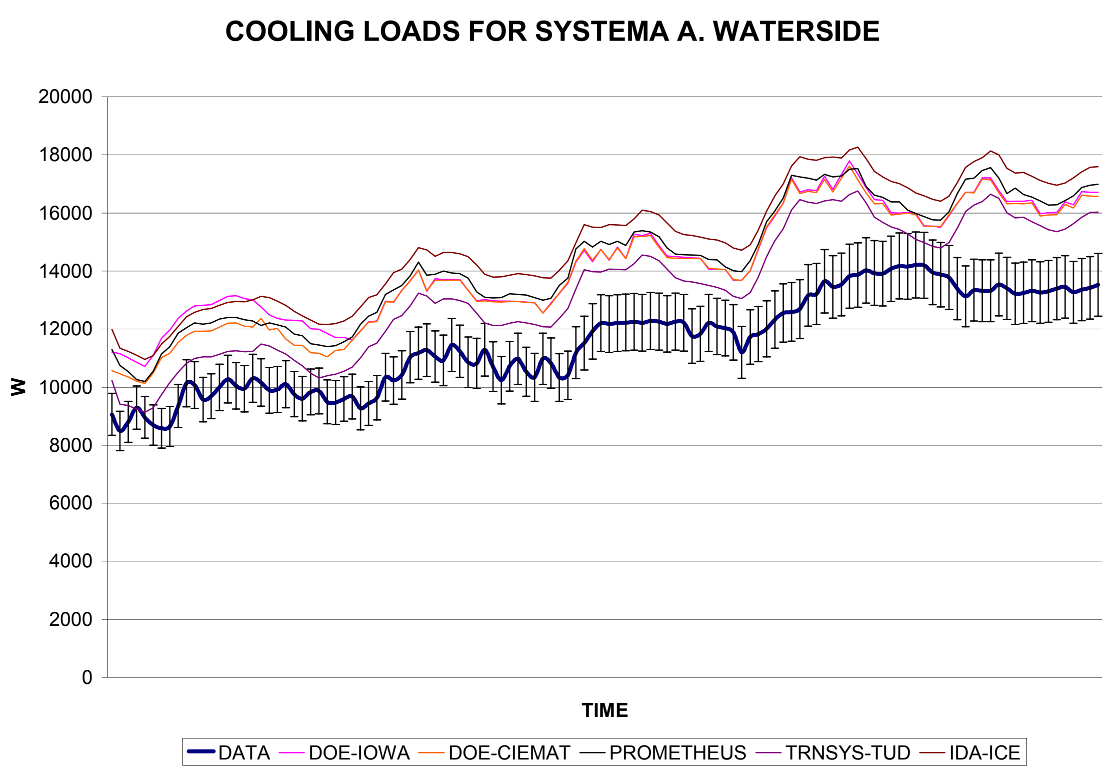

Figure 2.5 Waterside cooling loads

Table 2.24. Statistical summary of waterside corrected energy balance (W)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	CORRECTED
dtmin	-1265.15	-1707.97	-1644.97	-2664.53	-814.04	
dtmax	2008.94	1950.94	2091.94	1147.09	2632.18	
meandt	68.82	-135.74	132.80	-842.82	776.94	
min	10707.00	10132.00	10197.00	9137.27	10949.20	11111.75
max	17792.00	17614.00	17563.00	16758.50	18272.63	16825.67
mean	14276.38	14071.83	14340.37	13364.75	14984.51	14207.6
abmeandt	483.48	568.65	577.72	958.10	832.52	
rsqmeandt	613.54	697.78	722.80	1121.85	1027.02	
stderr	609.67	684.45	710.50	740.42	671.66	
stderr/mean	0.04	0.05	0.05	0.06	0.04	
MEAN%	0%	-1%	1%	-6%	5%	

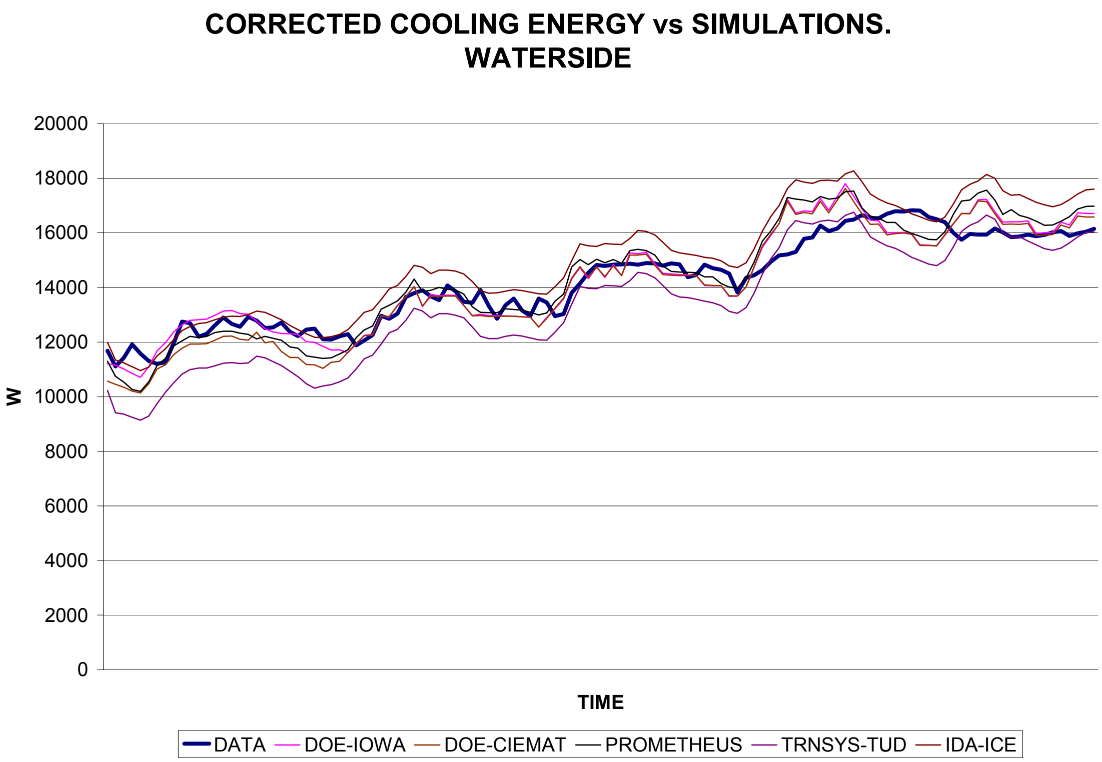

Figure 2.6 Corrected waterside cooling energy

This 2618 W difference could be caused by an airflow measuring error of 0.72 m^3/h or a temperature differences measuring error as small as 0.362 ºC.

Conclusions: Possible measuring errors of 0.362ºC might be causing cooling load disagreements of 20%. If this error is assumed, the results of all the models are close together and very similar (both, mean values and fast dynamics) to the corrected cooling loads.

2.4.2.5. Cooling Energy calculated at the Airside. Non Test Defined Parameter

Once the cooling energy considering the waterside has been analyzed, the cooling energy considering the airside is evaluated.

Table 2.25 shows the results obtained considering the airside. All the models overestimated the cooling loads. The closest prediction is given by the TRNSYS-TUD model and is 6% overestimated. This table shows the same problem as in the previous case: a possible error or misinterpretation on the cooling loads measurements, which are too low at every hour. Figure 2.7 shows the simulations and measurement results for the cooling loads. All the models predicted the cooling loads within the error band.

Table 2.25 Statistical summary of air-side energy balance (W).

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERT.
dtmin	-569.64	-1217.64	-1154.64	-2174.20	-323.71		**21.4%**
dtmax	4359.94	4301.94	4674.30	3577.49	5088.45		
meandt	1631.99	1427.43	1695.97	720.36	2340.11		
min	10707.00	10132.00	10197.00	9137.27	10949.20	9335.058	1997.70
max	17792.00	17614.00	17563.00	16758.50	18272.63	13918.055	2978.46
mean	14276.38	14071.83	14340.37	13364.75	14984.51	12644.4	2705.90
abmeandt	1664.45	1652.22	1829.84	1384.01	2349.61		
rsqmeandt	2089.64	2047.40	2261.66	1705.60	2765.89		
stderr	1305.07	1467.75	1496.25	1546.01	1474.45		
stderr/mean	0.09	0.10	0.10	0.12	0.10		
MEAN%	13%	11%	13%	6%	19%		

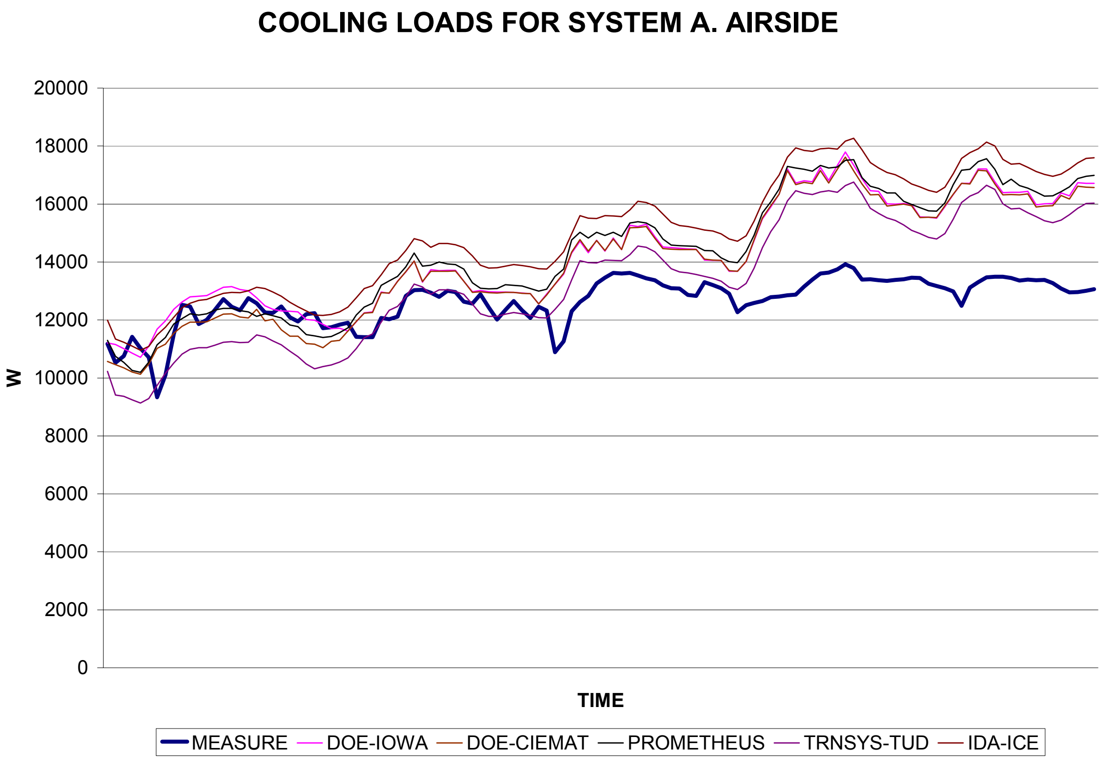

Figure 2.7 Corrected airside cooling energy

As Figure 2.7 shows, for the first two days, the simulations are very accurate. After that, some strange behavior on the measurements has been observed, the measurements are smaller than the simulations every time.

Conclusions: Possible-measurement errors on the cooling loads at the airside.

2.4.3. AHU B System Data

As it has been done for system A, the simulation results for the B Air Handling Unit will be compared with the measurements.

2.4.3.1. Supply Airflow. Test Defined Parameter

As for System A, the supply airflow has been defined as a constant air volume of 3516 m^3/h. All the programs simulated this condition quite accurately, as shown in Table 2.26.

Table 2.26 Statistical summary of supply airflow rate. (m^3/h)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERTAINTY
dtmin	32.62	31.62	31.62	33.12	32.11		**2%**
dtmax	166.08	165.08	165.08	166.58	165.58		
meandt	58.68	57.68	57.68	59.18	58.18		
min	3517.00	3516.00	3516.00	3517.50	3516.49	3350.918	67.0
max	3517.00	3516.00	3516.00	3517.50	3516.49	3484.384	69.7
mean	3517.00	3516.00	3516.00	3517.50	3516.49	3458.3	69.2
abmeandt	58.68	57.68	57.68	59.18	58.18		
rsqmeandt	60.13	59.15	59.15	60.62	59.64		
stderr	13.11	13.11	13.11	13.11	13.11		
stderr/mean	0.00	0.00	0.00	0.00	0.00		
MEAN%	2%	2%	2%	2%	2%		

The mean errors are around 2%, which is just within the uncertainty band. As in system A, the disagreements are mainly caused by the inconsistent behavior of the real case and the differences between the set point and the actual airflow.

The standard error is the same for all of the models because the real airflow is not constant, but it has small variations. The simulated models are strictly constant.

Conclusions: Both, the mean differences and the standard errors can be neglected.

2.4.3.2. Outside Airflow. Test Defined Parameter

As in System A, the outside airflow was defined as fixed at 680 m^3/h. All the models accurately predicted this value.

Table 2.27 Statistical summary of outside airflow rate (m^3/h).

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERTAINTY
dtmin	-5.69	-19.95	-698.54	-18.54	-18.13		**2%**
dtmax	144.21	129.95	-548.64	131.36	131.77		
meandt	13.80	-0.46	-679.05	0.95	1.37		
min	692.85	678.59	0.00	680.00	680.41	548.638	11.0
max	692.85	678.59	0.00	680.00	680.41	698.541	14.0
mean	692.85	678.59	0.00	680.00	680.41	679.0	13.6
abmeandt	13.96	5.64	679.05	5.45	5.44		
rsqmeandt	19.18	13.33	679.18	13.36	13.39		
stderr	13.32	13.32	13.32	13.32	13.32		
stderr/mean	0.02	0.02	#¡DIV/0!	0.02	0.02		
MEAN%	2%	0%	-100%	0%	0%		

The PROMETHEUS model had an input error and considered a 0 m^3/h outside airflow. This value should be checked.

For the other models, both types of errors, mean value and standard error, can be neglected. The standard error values are due to the non-predictable variations in the real airflow.

Conclusions: The PROMETHEUS model considered a 0 m3/h outside airflow. This value should be reviewed. The errors can be neglected for the other models.

2.4.3.3. Temperatures

- **Supply Air Temperature. Test Defined Parameter**

All the models are accurately predicting this temperature. Disagreements are smaller than 0.5ºC. Only PROMETHEUS estimated a supply air temperature outside of the uncertainty band. Table 2.28 gives the statistical summaries of supply air temperature from all participants.

Table 2.28 Statistical summary of supply air temperature (ºC)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERT.
dtmin	-0.59	-0.79	-0.32	-0.82	-0.81		**1%**
dtmax	1.12	1.02	1.52	1.02	1.03		
meandt	0.12	-0.09	0.40	-0.10	-0.09		
min	11.90	11.70	12.30	11.80	11.81	10.785	0.1
max	12.10	11.90	12.40	11.80	11.81	12.619	0.1
mean	12.02	11.81	12.30	11.80	11.81	11.9	0.1
abmeandt	0.24	0.22	0.43	0.22	0.22		
rsqmeandt	0.30	0.29	0.49	0.29	0.29		
stderr	0.27	0.27	0.28	0.28	0.28		
stderr/mean	0.02	0.02	0.02	0.02	0.02		
MEAN%	1%	-1%	3%	-1%	-1%		

Conclusions: The models are accurately predicting this temperature.

- **Return Air Temperature. Non Test Defined Parameter**

All the programs estimated accurately the return air temperature. The greatest error is given by the DOE-IOWA model and is only 1.34ºC. The error of all the other programs is lower than 1ºC. Table 2.29 gives the statistical summaries for return air temperature from all participants.

Table 2.29 Statistical summary of return air temperature (ºC)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERT.
dtmin	0.83	-0.17	0.57	0.37	-0.20		**1%**
dtmax	2.98	1.88	2.68	2.49	1.91		
meandt	1.34	0.27	0.97	0.78	0.20		
min	23.20	22.20	23.00	22.80	22.23	20.316	0.2
max	23.60	22.40	23.00	22.81	22.24	22.435	0.2
mean	23.37	22.30	23.00	22.81	22.23	22.0	0.2
abmeandt	1.34	0.30	0.97	0.78	0.26		
rsqmeandt	1.38	0.40	1.02	0.83	0.36		
stderr	0.33	0.30	0.30	0.30	0.30		
stderr/mean	0.01	0.01	0.01	0.01	0.01		
MEAN%	6%	1%	4%	4%	1%		

Conclusions: All the programs properly estimated the return air temperature.

- **Entering Cooling Coil Air Temperature. Non Test Defined Parameter**

All the models, except DOE-IOWA, are predicting almost exactly this temperature. This model is overestimating it by 3% (only 0.64ºC). This overestimation is the result of the errors in the return air temperature. Table 2.30 gives statistical summaries for cooling coil entering air temperature from all participants.

Table 2.30. Statistical summary of entering cooling coil temperature (ºC)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERT.
dtmin	-0.01	-0.83	-0.23	-0.37	0.05		**1%**
dtmax	2.21	1.31	1.91	1.88	2.11		
meandt	0.64	-0.23	0.33	0.30	0.52		
min	21.00	20.20	20.90	20.64	21.06	20.588	0.2
max	24.50	23.50	24.00	24.15	24.22	23.559	0.2
mean	22.94	22.07	22.64	22.60	22.82	22.3	0.2
abmeandt	0.64	0.39	0.37	0.35	0.52		
rsqmeandt	0.75	0.45	0.51	0.48	0.62		
stderr	0.39	0.39	0.39	0.38	0.34		
stderr/mean	0.02	0.02	0.02	0.02	0.01		
MEAN%	3%	-1%	1%	1%	2%		

All the standard errors are very low and similar.

Conclusions: Errors can be neglected.

2.4.3.4. Cooling Energy calculated at the Waterside. Non Test Defined Parameter

If the airflow and the entering and leaving cooling coil temperatures are being accurately predicted, the cooling energy has to be simulated accurately. But, as Table 2.31 shows, all the programs are overestimating the cooling loads. Their error is very close to the uncertainty band, so results can be considered as quite good.

Table 2.31 Statistical summary of waterside energy balance (W)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERT.
dtmin	400.6	181.1	366.0	-614.9	1141.1		8%
dtmax	3396.0	3378.0	3475.0	2393.2	3786.4		
meandt	1513.7	1349.6	1574.2	692.1	2217.2		
min	10709.0	10178.0	10198.0	9223.9	10949.8	9061.7	724.94
max	17798.0	17662.0	17564.0	16859.0	18277.1	15214.3	1217.14
mean	14281.1	14117.0	14341.6	13459.5	14984.6	12767.4	1021.39
abmeandt	1513.7	1349.6	1574.2	744.3	2217.2		
rsqmeandt	1620.3	1479.4	1691.4	940.6	2290.4		
stderr	577.9	605.9	618.8	636.9	574.4		
stderr/mean	0.04	0.04	0.04	0.05	0.04		
MEAN%	12%	11%	12%	5%	17%		

Figure 2.8 shows the waterside cooling loads for this AHU. As shown in System A, the measurements have the same behavior as the simulations but the mean value is different. The predictions are very close to the measurements if the error band is considered. In this case, the mean value of the standard error of the simulations is 1470W, so this is added to the measurements to correct them. Once again, the results are much better, as Table 2.32 and Figure 2.9 shows.

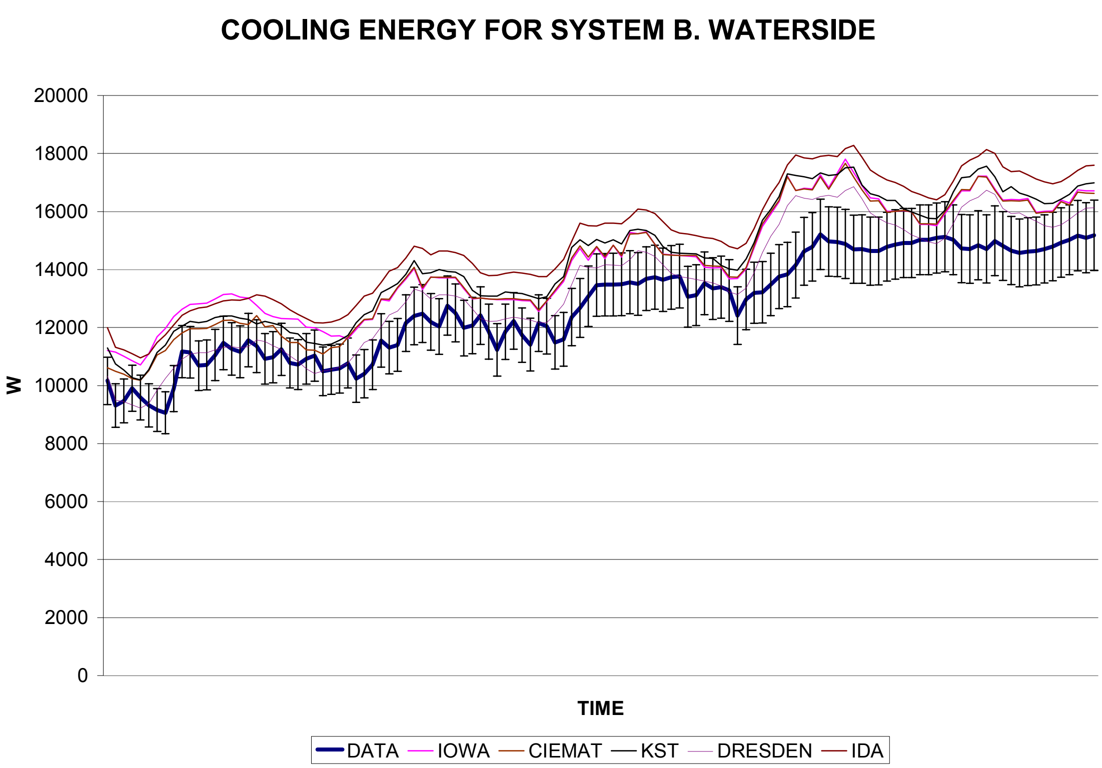

Figure 2.8 Waterside cooling energy for system B

Table 2.32. Statistical summary of waterside corrected energy balance (W)

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE
dtmin	-1068.81	-1288.24	-1103.39	-2084.24	-328.29	
dtmax	1926.61	1908.61	2005.61	923.86	2316.99	
meandt	44.33	-119.74	104.81	-777.24	747.84	
min	10709.00	10178.00	10198.00	9223.90	10949.76	10531.1
max	17798.00	17662.00	17564.00	16859.00	18277.06	16683.7
mean	14281.10	14117.03	14341.58	13459.53	14984.60	14236.8
abmeandt	456.82	509.43	506.29	868.56	769.53	
rsqmeandt	579.60	617.59	627.59	1004.86	942.95	
stderr	577.90	605.87	618.77	636.90	574.36	
stderr/mean	0.04	0.04	0.04	0.05	0.04	
MEAN%	0%	-1%	1%	-5%	5%	

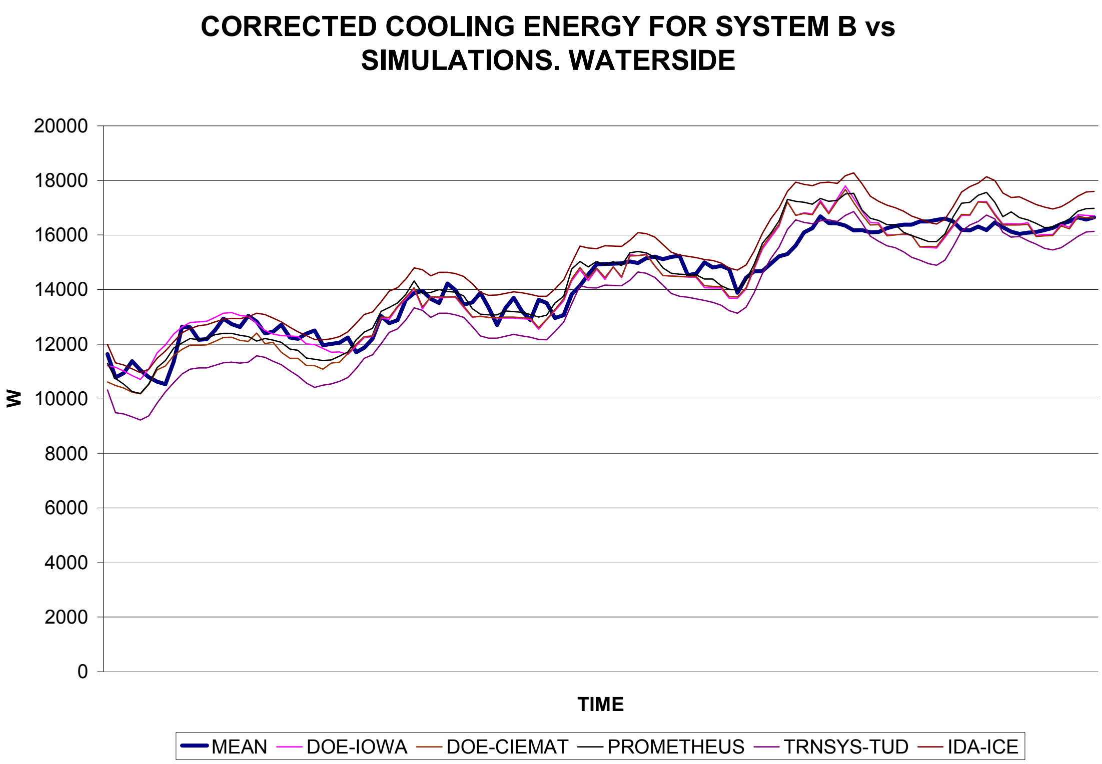

Figure 2.9 Corrected water side cooling energy

This 1470 W difference could be caused by a water flow measuring error of 0.4 m^3/h or a temperature difference measuring error as small as 0.4ºC.

Conclusions: Possible measurement errors of 0.2ºC might be causing cooling loads disagreements of 10%. Predictions are very close to measurements if the uncertainty band is considered.

2.4.3.5. Cooling Energy calculated at the Airside. Non Test Defined Parameter

Once the cooling energy considering the waterside has been analyzed, the cooling energy considering the airside will be evaluated.

Table 2.33 shows the results obtained considering the airside. The simulations are very accurate in some cases, such as the DOE-CIEMAT and PROMETHEUS cases. The errors in the other cases are always around 7%, which are also quite accurate. In fact, they are within the uncertainty band. As Figure 2.10 shows, for the first two days, the simulations are very accurate. Some strange behavior on the measurements are observed after the second day of simulation.

Table 2.33. Statistical summary of air-side energy balance (W).

	DOE-IOWA	DOE-CIEMAT	PROMETHEUS	TRNSYS-TUD	IDA-ICE	MEASURE	UNCERT.
dtmin	-78.20	-874.59	-757.39	-157.15	146.81		**21.4%**
dtmax	2047.31	1223.57	1575.17	1838.51	2198.01		
meandt	804.99	38.57	122.56	669.28	917.14		
min	10551.00	9962.00	10079.20	10360.33	10842.53	9676.025517	2070.67
max	14536.93	13712.40	13712.40	14480.02	14546.57	13110.97326	2805.75
mean	12795.04	12028.63	12112.62	12659.33	12907.20	11990.1	2565.87
abmeandt	807.47	386.85	380.70	678.45	917.14		
rsqmeandt	932.97	463.74	480.32	814.59	1009.18		
stderr	471.62	462.13	464.42	464.35	421.07		
stderr/mean	0.04	0.04	0.04	0.04	0.03		
MEAN%	7%	0%	1%	6%	8%		

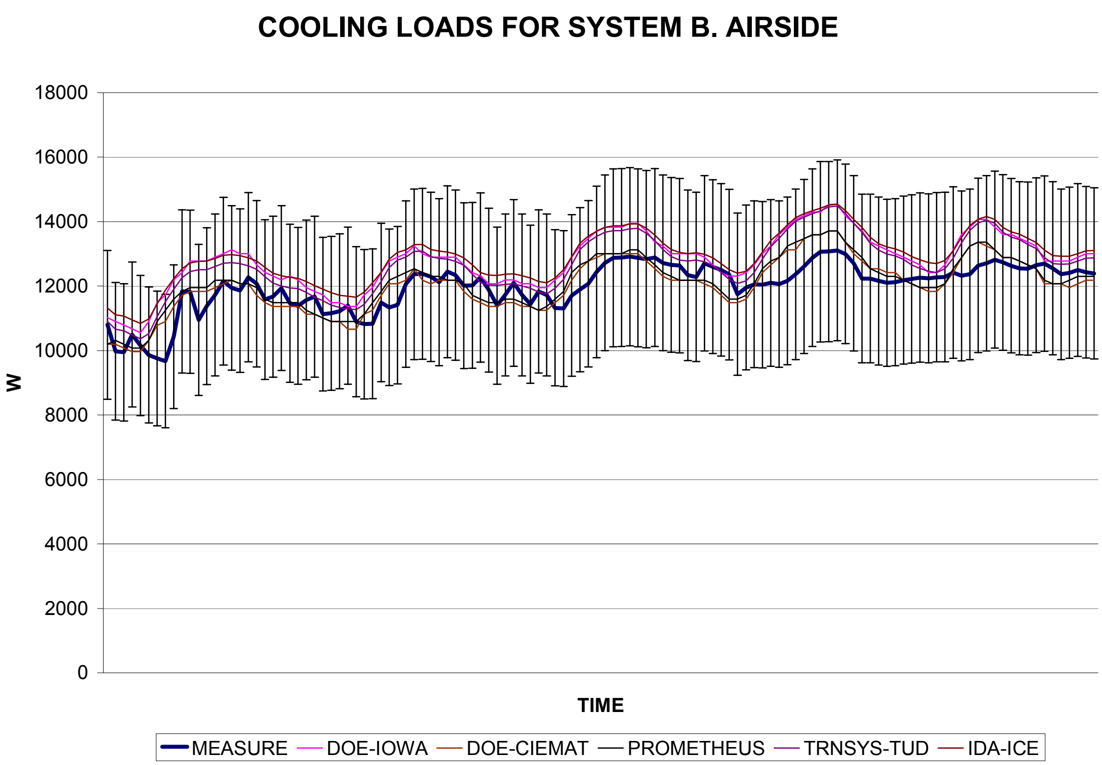

Figure 2.10 Airside cooling energy for System B

Conclusions: Possible measurement errors on the cooling loads on the waterside. Results on the airside are very accurate and are always within the uncertainty band.

2.4.4. Global Reheat Energy and Cooling Energy Supplied Into the Room. Non test Parameters

The behavior of each room has been analyzed and the results are presented in Appendix E. Questions might be answered by reading this appendix.

2.4.4.1. General Conclusions Common to Every Room

The analysis of the behavior of each room is presented in Appendix E. Some general conclusions, common for every room have been developed. To determine how these conclusions have been obtained, it is recommended to read the appendix.

- DOE-IOWA, DOE-CIEMAT and TRNSYS-TUD models accurately estimated mean values and fast dynamics. They only had some problems in estimating the large solar heat gains. Possible solar radiation gains modeling error or erroneous window specifications could be the cause.

- PROMETHEUS and IDA-ICE calculate higher solar gains. They showed larger errors in the mean values and fast dynamics.

2.4.4.2. Total Reheat Energy of System A

Each room of the System A have required some electrical reheat energy at each hour. The reheat energy demanded by all the rooms of the System A is presented in Table 2.34 and Figure 2.11.

Table 2.34. Electrical Reheat Energy demanded by the System A (W)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-1152.69	-1581.61	-386.61	-864.26	-412.49	
dtmax	2235.33	1871.99	3535.33	2619.58	3140.87	
meandt	-55.76	-20.55	1254.42	623.61	1030.27	
min	3064.00	3009.00	4889.00	4173.90	4805.59	2061.6
max	6791.00	6769.00	7361.00	7032.95	7195.49	7608.0
mean	5131.06	5166.28	6441.24	5810.43	6217.09	5186.8
abmeandt	532.22	645.51	1278.74	778.77	1070.72	
rsqmeandt	641.78	768.69	1622.76	1048.77	1404.12	
stderr	639.35	768.41	1029.46	843.23	953.99	
mean%	-1%	0%	24%	12%	20%	
stderr/mean	0.12	0.15	0.20	0.16	0.18	

The DOE-IOWA model presented a very good result with an error of only 1% (less that 60W for the building). Considering that the total floor area for the A room types is 100 m^2, the mean error is 0.6 W/m^2, which is obviously negligible. It slightly underestimated the large values and overestimated the small ones. The mean value predictions are very good and the fast dynamics are properly simulated.

DOE-CIEMAT model was very accurate (mean error is only 20 W, which means $0.2 W/m^2$) but presented a small error for low values, as Figure 2.11 shows. It showed the same kind of problem that DOE-IOWA model in the fast dynamics.

PROMETHEUS model had an overestimation of 24%, which is almost 1200 W for the building ($12.5 W/m^2$). It was caused by an overestimation of the low values.

TRNSYS-TUD model showed the same problem but was more accurate on high values. Its error is 12% (620 W<>$6.2 W/m^2$).

IDA-ICE behavior is very similar to PROMETHEUS. The mean error is 20% which means 1020 W ($10.2 W/m^2$).

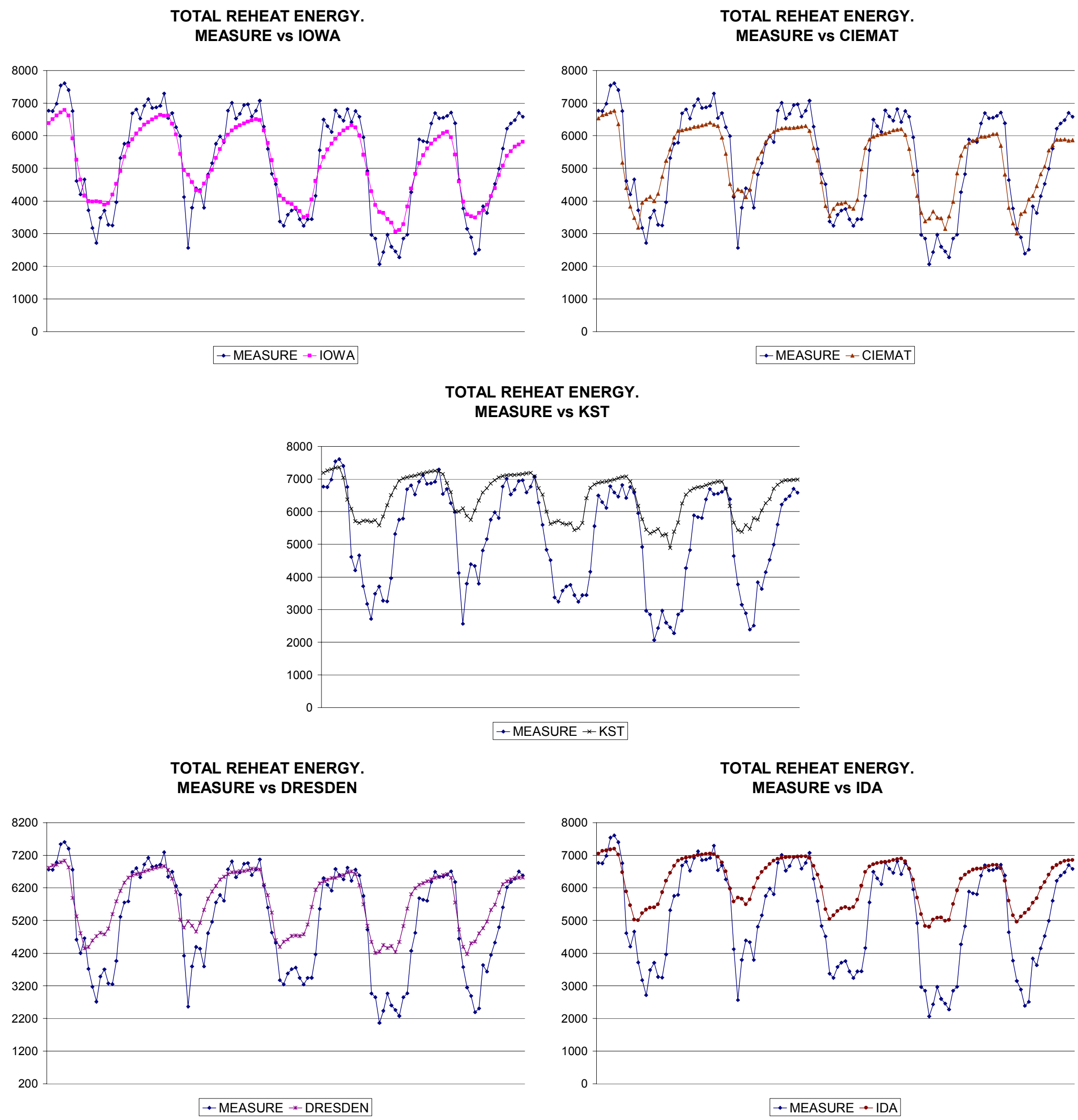

Figure 2.11 Comparison of measured reheat energy and simulation reheat energy. System A

2.4.4.3. Total Cooling Energy Supplied to A type room

The cooling energy supplied have been calculated as explained in Appendix E. Results obtained for System A are presented in Table 2.35 and Figure 2.12.

Table 2.35. Cooling energy supplied to the rooms by the System A (J)

	IOWA	CIEMAT	KST	DRESDEN	IDA-ICE	REAL
dtmin	-8056.10	-8512.87	-16579.75	-10366.49	-12568.62	
dtmax	3063.26	6109.82	-301.78	3536.03	1524.41	
meandt	-949.25	-804.22	-7238.52	-2845.09	-4224.51	
min	18643.32	19511.28	15270.48	18552.16	18068.13	16862.5
max	32427.36	32625.36	24169.68	28859.10	26684.00	37743.4
mean	24867.37	25012.40	18578.09	22971.52	21592.10	25816.6
abmeandt	1824.02	2177.33	7238.52	3164.85	4324.83	
rsqmeandt	2451.17	2798.68	8160.00	4161.42	5467.91	
stderr	2259.90	2680.64	3766.87	3036.92	3471.53	
mean%	-4%	-3%	-28%	-11%	-16%	
stderr/mean	0.09	0.10	0.15	0.12	0.13	

The DOE-IOWA model presented again a very good result with an error of only 4%. It had some problems on large values (solar gains) but accurately predicted fast dynamics.

DOE-CIEMAT model was very accurate (3% error) but presented the same type of error as the DOE-IOWA model.

PROMETHEUS model had an underestimation of 28%. It underestimated all the values but especially had some problems with dynamics and large values.

TRNSYS-TUD model showed an error of 11% due to an error similar to Iowa's and CIEMAT's but larger.

IDA-ICE model shows a behavior similar to TRNSYS-TUD but its errors are even larger.

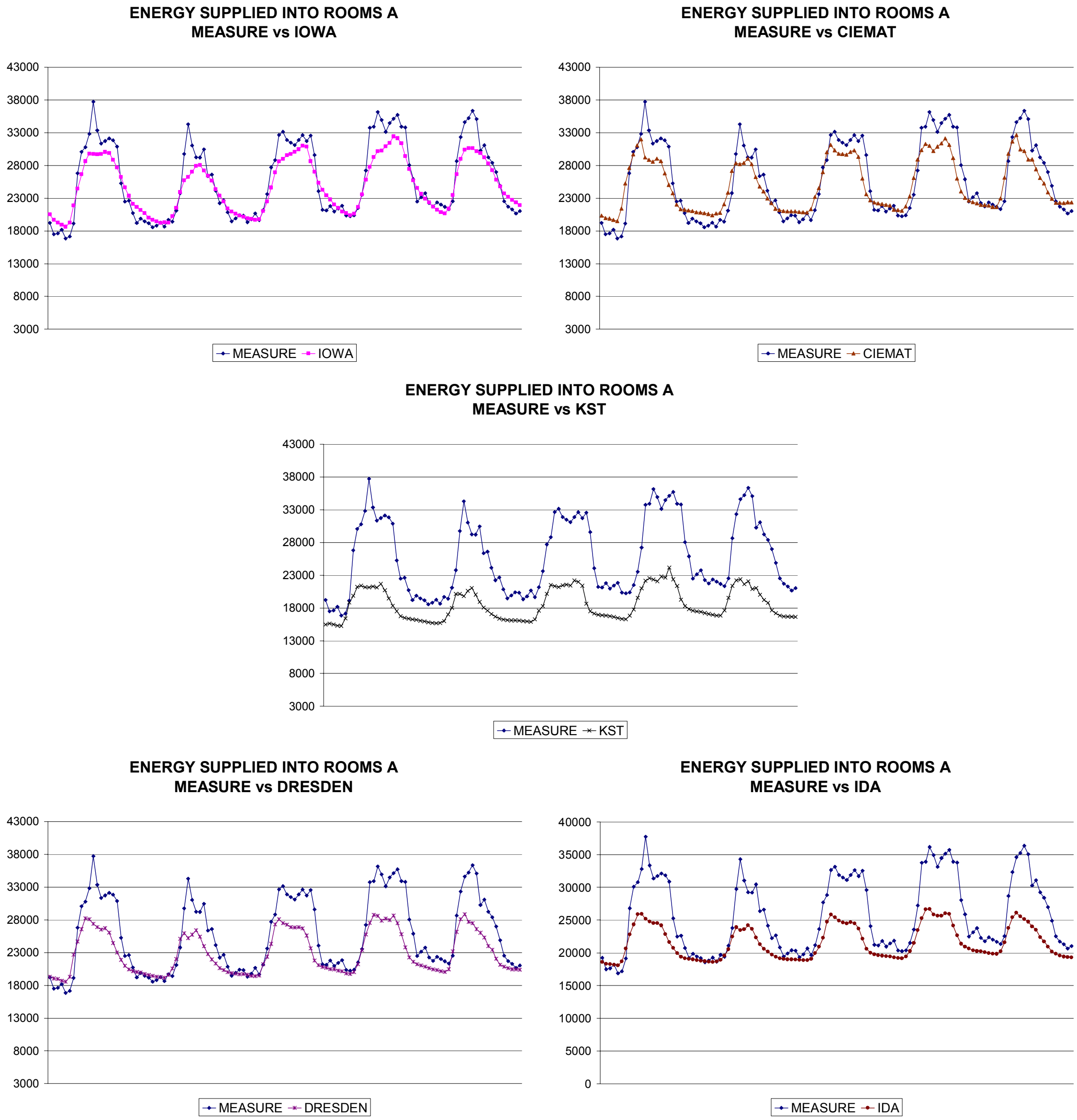

Figure 2.12 Comparison of energy supplied into Rooms A

2.4.4.4. Total Reheat Energy of System B

The reheat energy demanded by all the System B is presented in Table 2.36 and Figure 2.13.

Table 2.36 Electrical Reheat Energy demanded by the System B (W)

	IOWA	CIEMAT	KST	DRESDEN	IDA-ICE	REAL
dtmin	-797.59	-1299.63	-315.63	-452.01	-100.27	
dtmax	2282.06	2604.79	3612.79	3342.69	3671.34	
meandt	234.58	210.87	1082.01	892.96	1258.80	
min	3064.00	2946.00	4419.00	4202.25	4726.98	2036.6
max	6791.00	6715.00	6905.00	7002.80	7144.79	7075.1
mean	5131.06	5107.35	5978.48	5789.43	6155.28	4896.5
abmeandt	447.89	576.84	1114.92	926.16	1261.14	
rsqmeandt	598.26	744.40	1416.46	1177.27	1512.51	
stderr	550.35	713.91	914.12	767.20	838.51	
mean%	5%	4%	22%	18%	26%	
stderr/mean	0.11	0.15	0.19	0.16	0.17	

The DOE-IOWA model presented a very good result with an error of only 5% (less that 235W for all the building-2.35W/m^2). This model showed some small problems on the smallest values.

DOE-CIEMAT model was very accurate but (210W of mean error, which is 2 W/m^2). It presented a small error on the 4^{th} day (see figure). The error encountered in the 4^{th} day for all the programs have been caused by an uncontrolled heat gain on the interior B room during the test (see appendix 1A).

PROMETHEUS model had an overestimation of 22%, which is almost 1100W for all the building (11W/m^2). It was caused by an overestimation of the low values.

TRNSYS-TUD model showed the same problem but was more accurate on high values. Its error is 18% (890 W).

IDA-ICE model overestimated all the values and its error is 26%, which is 12W/m^2.

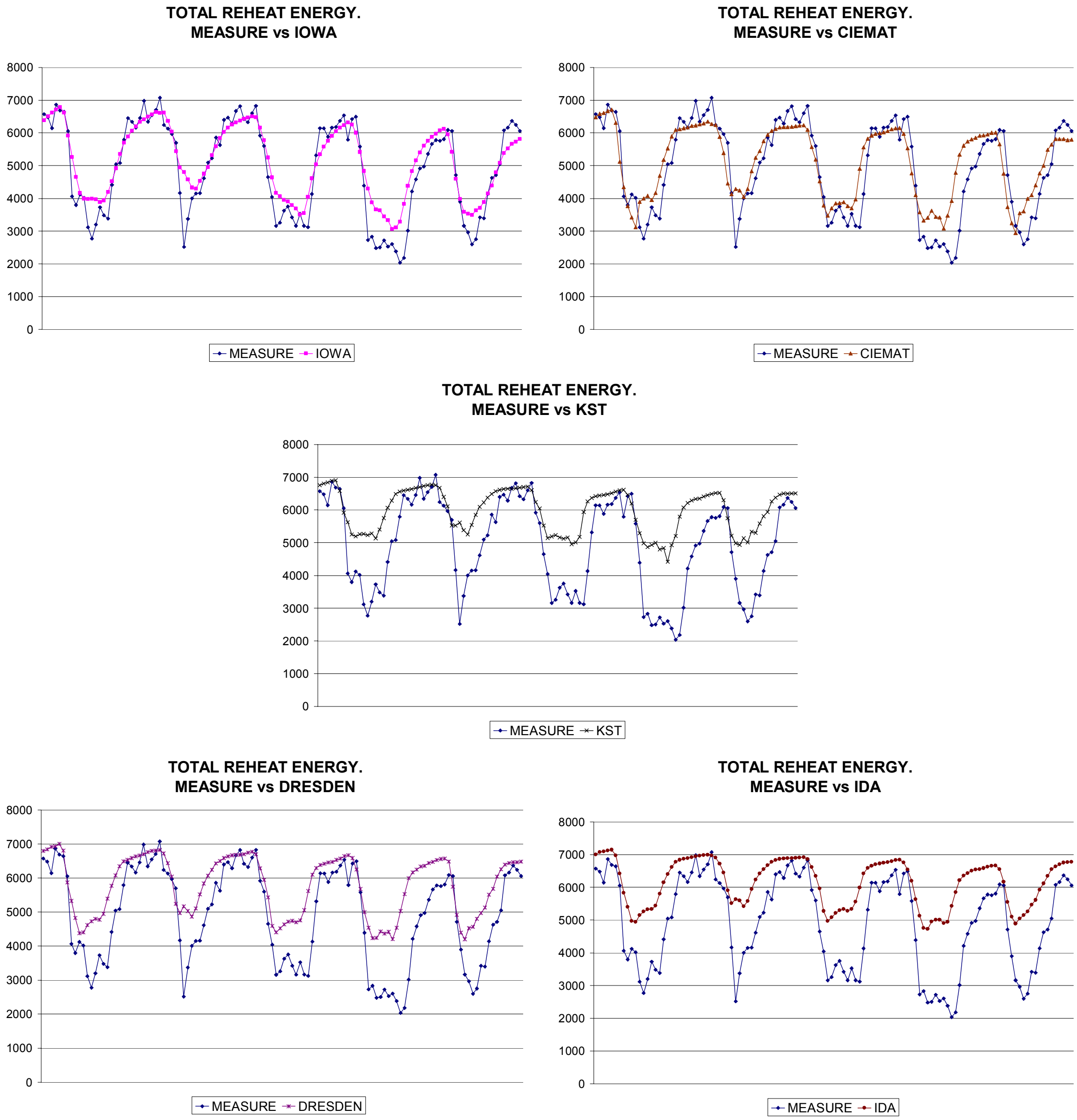

Figure 2.13 Comparison of total reheat energy for System B

2.4.4.5. Total Cooling Energy Supplied to B Type Room

The cooling energy supplied through system B is presented in Table 2.37 and Figure 2.14.

Table 2.37 Cooling energy supplied to the rooms by the System B (J)

	IOWA	CIEMAT	KST	DRESDEN	IDA-ICE	REAL
dtmin	-7978.45	-7846.21	-14895.01	-10431.12	-12274.78	
dtmax	2451.59	5672.82	722.82	3007.33	1270.01	
meandt	-1105.85	-509.60	-5725.68	-2922.11	-4154.98	
min	18643.32	19871.28	16912.08	18656.02	18250.65	17221.3
max	32427.36	33274.08	25861.68	28772.43	26966.99	37707.5
mean	24863.85	25460.10	20244.02	23047.59	21814.72	25969.7
abmeandt	1588.01	1844.97	5742.92	3095.27	4215.11	
rsqmeandt	2195.15	2392.36	6605.50	3933.94	5116.85	
stderr	1896.25	2337.46	3293.81	2633.85	2986.36	
mean%	-4%	-2%	-22%	-11%	-16%	
stderr/mean	0.07	0.09	0.13	0.10	0.11	

The DOE-IOWA model presented again a very good result with an error of only 4%. It shows the same problem that for System A; it had some problems on large values (solar gains).

DOE-CIEMAT model was also very accurate (2% error) and presents the same type of error as the DOE-IOWA model.

PROMETHEUS model had an underestimation of 22%. It had some problems with dynamics and large values.

TRNSYS-TUD model showed an error of 11% due to an error similar to Iowa’s and CIEMAT’s.

IDA-ICE´s behavior is similar to TRNSYS-TUD but it’s error is larger.

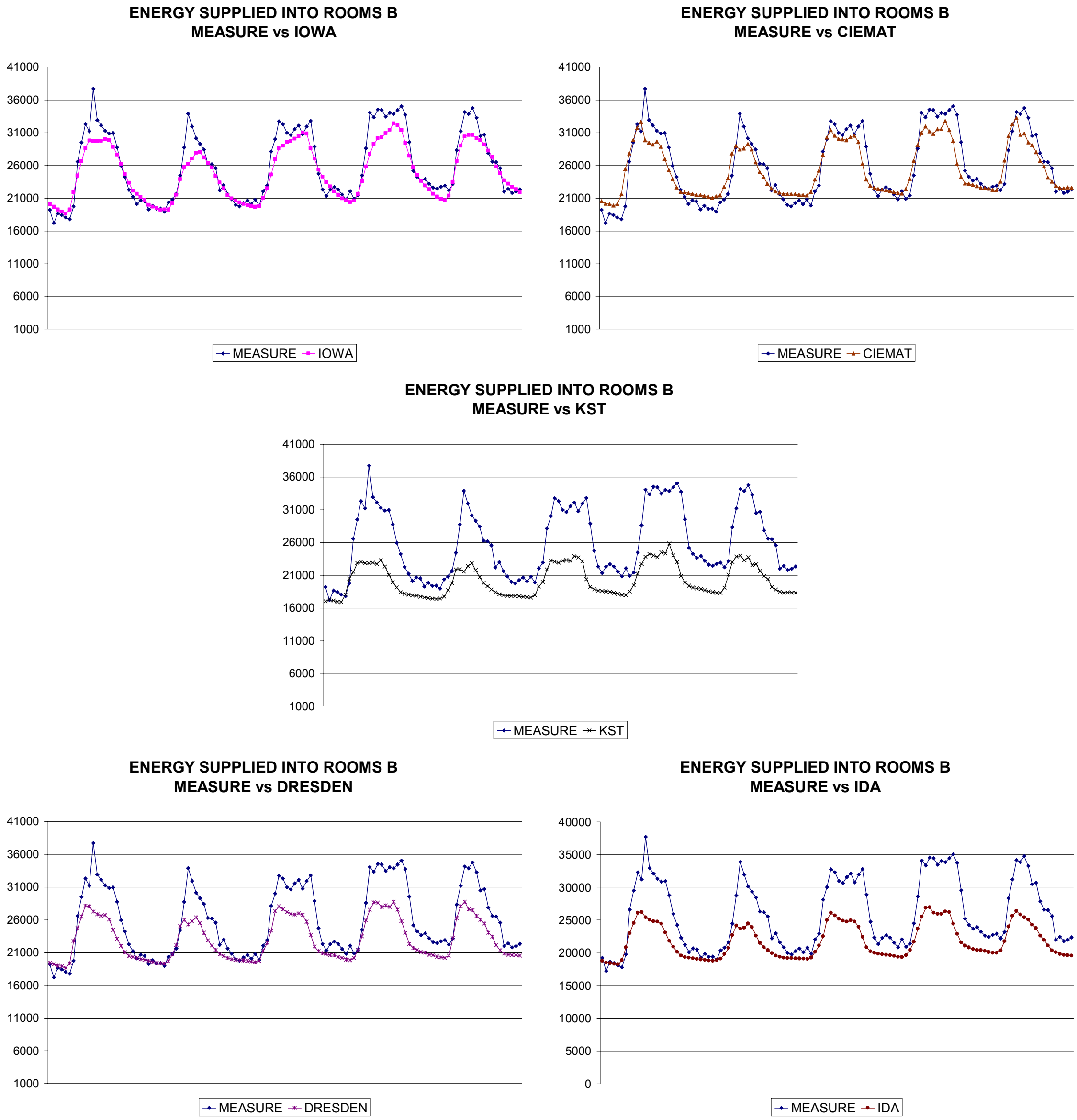

Figure 2.14 Comparison of energy supplied for System B

2.5. Discussion of the Results and Conclusions

The weather processor is very similar and accurate for every model tested. The main differences are due to different interpretation of the hour. Some models considered that the data were provided in solar time and made some corrections to consider the local standard time. Differences are very small and can be neglected.

The disagreement between measurements and simulations in the airflow is caused by the differences between actual control, where the control systems are not perfect, and theoretical control. Those differences can be neglected.

All the models accurately predicted the different temperatures on the AHU. Considering the accuracy of the predictions for the airflow and the temperatures, the cooling loads should be very well estimated. There are two possibilities for measuring the cooling load: waterside measures and airside measurements.

If the waterside measures are used, possible measuring errors of 0.362ºC for System A and 0.2ºC for Systems B might be causing cooling loads disagreements. If this error is assumed, the results of all the models are close together and very similar (both, mean values and fast dynamics) to the cooling loads.

If the airside measurements are used, the simulations are very accurate and agree with measurements, especially for System A.

After the room analysis of the simulations some conclusions can be drawn from the results:

- DOE-IOWA, DOE-CIEMAT and TRNSYS-TUD models accurately estimated mean values and fast dynamics. They only had some problems in estimating the large solar heat gains. Possible solar radiation gains modeling error or erroneous window specifications.

- PROMETHEUS and IDA-ICE calculate higher solar gains. They showed larger errors in the mean values and fast dynamics.

The simulation of the entire building and the comparison between measurements and predictions allow the following conclusions:

- DOE-IOWA and DOE-CIEMAT predictions are very accurate. They showed mean errors on the reheat energy around 2 W/m^2. They had some problems on the lowest values, slightly overestimating them.

- The other models were fairly accurate, obtaining errors around 10 W/m^2. They showed the same problem but their over-prediction of the low values is larger. This error could be caused by a misinterpretation on the losses through the floor.

IDA-ICE distinctly overestimates the reheat energy. This overestimation is due to an improperly modelled adjacent room relationship. See modeller report, Section 5.

Another point of view should be regarded:

A comparison of the results indicates that another behaviour should be considered. The results from DOE-CIEMAT and DOE-IOWA underestimate the high values by approximately about some 700-Watts. This value should be weighted in the same way as the low values. With this adjustment, TRNSYS-TUD and IDA-ICE accurately predict the high values, which are the starting values for the energy calculation. On that point, DOE-CIEMAT and DOE-IOWA already start on a lower level than the others.

If this point is not considered, the results of all the models were very accurate, especially for DOE-IOWA and DOE-CIEMAT which had errors smaller that 5%. All the models had small problems in estimating the higher cooling loads, probably due to an error in the solar gains simulation or in the window specifications.

3. SECOND EXERCISE. DYNAMIC CASE. (Variable Air Volume System with Scheduled Internal Loads)

3.1. Description of the Exercise.

This section contains information regarding the operating parameters and conditions used for a VAV test conducted at the Iowa Energy Center's Energy Resource Station as part of the empirical validation study for the International Energy Agency Task 22. The test was conducted over a four day period from February 5-8,1999.

Before running this test, a significant amount of work was done to improve the accuracy of the data obtained at the ERS. This included calibration of temperature sensors and air flow sensors. The participating experts found that in many cases a single-point measurement was insufficient to determine the temperature or volumetric flow rate of air. Water temperature sensors were calibrated as well as room air temperature sensors. Some of these measurements were off by as much as 3°F. Some differences in control parameters were observed between the test rooms. These were corrected. Duct leakage was addressed and an effort was made to minimize air movement between the test rooms and adjacent spaces. Tests were conducted to specifically look at the response of the test rooms. These tests included smoke tests, infrared imaging, blower door tests and thermal response tests. Although not perfect, the "A" and "B" test rooms match much better now than before and we have a great deal more confidence in the measured data than from previous tests.

For the test conducted, the "A" and "B" systems were operated in an identical manner. The "A" and "B" test rooms were operated utilizing Variable-Air-Volume ReHeat (VAVRH). Electric heating coils were used in the rooms to provide terminal heating. The air handling units were operated utilizing outdoor air economizer mode. The outdoor air flow rate was modulated to provide a discharge air temperature (after the supply fan) of 15.6 °C (or 60°F). The chiller was not used during the test, all cooling was accomplished with outdoor air. During the daytime of February 5, the outdoor air temperature rose to a value which resulted in the AHU discharge air temperature going above the set-point by about 2 degrees for one hour. The outdoor air temperature cooled down rapidly as the sun set and the outdoor air dampers were a little slow to respond. This resulted in the AHU discharge air temperature going below set-point by about 1 degree during the next hour. Other than this one time event, the economizer control maintained the AHU discharge air temperature within +/- 0.5 °C of set-point.

Another feature of this test was the use of thermostat schedules as well as scheduled internal sensible loads for the test rooms. The thermostats in the test rooms were programmed for a night setback temperature. The electric baseboard heaters in the test rooms were programmed to come on during the day to provide a scheduled internal load. The minimum outside air flow rates were also scheduled with the economizer. Although there is a minimum outside air flow rate schedule, outdoor air flow rates during the test never reached these minimum values. All of these schedules are summarized in Table 3.1. The lights in the test rooms were turned off. The thermostats in the rest of the ERS were not programmed for a setback schedule. The HVAC system that serves the remaining spaces at the ERS (i.e. computer room, classroom, etc.) was run to provide nearly constant temperature conditions in these spaces. The thermostats in the spaces adjacent to the test rooms were set at 22.2 °C and the temperature in the spaces adjacent to the test rooms remained fairly constant during the test. Specific adjacent space temperature data are provided in the file 990618adjtemp.dat. This file contains hourly temperature data.

3.1.1. Run Period and General Weather Conditions

This item is used to specify the initial and final dates of the desired simulation period and also the general conditions and location of the ERS facility.

- The dates for this test are February 5, 1999 through February 8, 1999.
- Weather data for Ankeny, Iowa is organized into the TMY format. In this file the measured data for the dates previously specified are included. This file is called "Ankeny.ia1" and is included with this report.
- Building Location
    - LATITUDE: 41.71 degrees North
    - LONGITUDE: 93.61 degrees West
    - ALTITUDE: 938.0 feet (285.9 m)
    - TIME-ZONE: 6, central time zone in U.S.
    - DAYLIGHT-SAVINGS: NO

3.1.2. Test Rooms Operation and Control Parameters

The following conditions apply to all of the test rooms. These conditions do not apply to the rest of the building where occupants may be present and lighting and window shading devices are used.

3.1.2.1. Internal Loads and General Room Conditions

A baseboard heater is installed inside each test room. The baseboard heaters were used to simulate internal loads in the test rooms for this test (additional information about the baseboard heaters is provided in the Appendix C). The internal loads scheduled during the test are shown in Table 3.1a

Besides the baseboard heaters, other general room characteristics must be considered:

- No lights or miscellaneous equipment other that the baseboards.
- No shading device on windows.
- No infiltration.

3.1.2.2. Room HVAC specifications

Each test room has its own thermostat and some HVAC specifications can be considered.

- **Thermostat Schedule**

Each test room has its own thermostat and the set point value is the same for all test rooms. The following values were used for both tests:

- Design heat temperature: 22.2 °C
- Design cool temperature: 22.7 °C
- Heat temperature schedule: see Table 3.1a
- Cool temperature schedule: see Table 3.1a
- Internal loads schedule: see Table 3.1a
- Dead-ban: 1.7 °C

Table 3.1a Set point temperature and internal loads schedules

Hour	Cooling set-point temperature (°C)	Heating set-point temperature (°C)	Internal loads (kW)
1-7	26.7	18.3	0
7-9	22.8	22.2	0
9-18	22.8	22.2	2
18-20	22.8	22.2	0
20-24	26.7	18.3	0

- **Room Airflow and Reheat Specifications**

The following airflow rates were specified for each test room:

- Exterior test rooms (east, south and west): max 1670 m^3/hr, min 765 m^3/hr

- Interior test rooms: max 1019 m^3/hr, min 467 m^3/hr
- The zone heat source installed are: 2 stage electric, max 3.34 kW (1.67 kW/stage) for exterior rooms and max 2 kW (1 kW/stage) for interior rooms

3.1.3. Air Handling Unit Operation and Control

Both AHU (A and B) were working in the same conditions to supply air to the four sets of test rooms.

3.1.3.1. Set Points and System Controls

The air handling system parameters were specified as follows:

- Supply air temperature: max 29.4 °C, min 15.6 °C
- Heating schedule: 24 hours available
- Cooling schedule: NOT available
- Cool control: supply air set point, 15.6 °C after the fan
- Preheat: NOT available
- Humidity control: NOT available
- Economizer: enabled
- Outside air control: temperature (supply air set point, 15.6 °C after the fan)

3.1.3.2. System Air and Fans

System airflow rates were specified as follows:

- Supply airflow: max 6116 m^3/hr
- Return air path: Plenum
- Minimum outside airflow: scheduled as shown in Table 3.1b

Table 3.1b Minimum outside air schedules

Hour	Minimum OA flow-rate (m^3/hr)
1-7	170
7-20	510
20-24	170

- Outside air control: Temperature
- Duct air loss: None
- Duct heat gain: 0.3 °C (increase)

The air handling unit fans were specified as follows:

- Supply air static pressure: 1.4 inch H_2O
- Fan schedule: Always on
- Supply Fan control: Duct static pressure of 1.4 inch H_2O
- Return Fan control: 90 % of supply fan speed
- Motor placement: In-Air flow
- Fan placement: Draw-Through

3.2. Participating Organizations

Five sets of results were developed with four different computer programs. The participating organizations and models are identified in Table 3.2.

Table 3.2. Participants

Notation	Program	Implemented By	Date of simulation/round
TRNSYS-TUD	TRNSYS-TUD (modified V.14.2)	University of Dresden Dresden, Germany	March 2000/3rd round
PROMETHEUS	PROMETHEUS	Klima System Technik Berlin, Germany	October 1999/2nd round
DOE-IOWA	DOE-2.1E	Iowa State University Ames, Iowa	October 1999/2nd round
DOE-CIEMAT	DOE-2.1E (V.088)	DOE-CIEMAT Madrid, Spain	June 2000/3rd round
IDA-ICE	IDA-ICE (V.2.11.06)	Hochschule Technik + Architektur Luzern, Switzerland	June 2000/3rd round

3.3. Comparison between A and B Room Type measurements

The measured results obtained for the A and B rooms were compared in order to find possible measurement errors.

For the test conducted, the pair of rooms, called "A" and "B", systems were operated in an identical manner. This should cause identical results for both room types and systems. In some cases, the measurements are different for each room type, and those differences are not negligible.

Before analyzing the accuracy of the model predictions, the errors associated with the measurements must be considered.

3.3.1. Systems Comparison

The first step comparing both room types measurements is to analyze the systems behavior. If the central system is supplying the air to each room at different temperatures, this should cause different reheat needs in each test room of a pair.

The parameters used for the analysis are as follows:

- A SYSTEM. Mean value for the A system.
- B SYSTEM. Mean value for the B system.
- B/A MEAN VALUE: Relation between both mean values

Table 3.3 shows the measured results.

Table 3.3. Comparison between system measurements

	SUPPLY AIR	TEMP ENTERING COIL	TEMP LEAVING COIL	OUTSIDE AIR	TEMP RETURN AIR
	m^3/h	ºC	ºC	m^3/h	ºC
A SYSTEM	2913	14.52	14.29	1477.5	21.15
B SYSTEM	2923	14.82	14.22	1707.7	20.89
B/A MEAN VALUE	100.33%	102.08%	99.45%	115.58%	98.77%

Those results show how both systems are operating in the same conditions. The supply air temperature, the temperatures entering and leaving the coil and the return air temperature are similar.

Only some small differences are found in the outside airflow for each room type. As the outside air is the method used by the Air Handling Unit to cool the return air, possible measuring errors can be analyzed by an energy balance on the economizer:

$$(m_{Supply} - m_{outside})h_{Return} + m_{outside}h_{outside} = m_{Supply}h_{in\ coil}$$

Applying the mean values of enthalpy and airflow to this equation, it can be concluded that the outside enthalpy is:

$$h_{outside} = \frac{m_{Supply}h_{in\ coil} - (m_{Supply} - m_{outside})h_{Return}}{m_{outside}}$$

Neglecting the variation of the humidity content, it can be assumed that:

$$t_{outside} = \frac{m_{Supply}t_{in\ coil} - (m_{Supply} - m_{outside})t_{Return}}{m_{outside}}$$

This value should be the same for both systems. The results obtained for these values are tabulated in Table 3.4.

Table 3.4. Entering outside temperature for each system.

	houtside
A SYSTEM	6.982
B PREDICTED	9.049
B/A MEAN VALUE	129.61%

The difference between both systems is almost 30%. Therefore, if the measurements are different in this percentage, the same error must be acceptable for the models.

After checking the facility, it was found that the difference was caused by a temperature increase in the outside air duct. The outside air is ducted through the mechanical room into the air handling units. This duct is not isolated, so there is a temperature increase of the air.

Once this problem has been detected, it has to be considered whenever the outside airflow measurement is considered during the analysis of the simulation results. The behavior of the economizers will be analyzed by comparing the results of the models.

Conclusions: The outside air measurement cannot be considered as a reliable measurement, because of a duct heat gain. Both room types should demand the same reheat energy.

3.3.2. Rooms Comparison

3.3.2.1. Interior Room

Table 3.5 shows the comparison between the A and B type Interior room temperatures, supply airflow and reheat energy.

Table 3.5. Interior rooms parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	21.0	516.2	-218.9
B ROOM	21.1	523.5	-218.0
B/A MEAN	100.2%	101.4%	99.6%

The results are very similar for both room types.

Conclusions: Disagreements in measurements are less that 2%.

3.3.2.2. East Room

Results of the comparison for the A and B East rooms are shown on Table 3.6.

Table 3.6. East rooms parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	20.5	770.5	-736.8
B ROOM	20.5	779.2	-726.2
B/A MEAN	100.2%	101.1%	98.6%

There are not considerable differences between both room types.

Conclusions: Disagreements in measurements are less that 2%.

3.3.2.3. South Room

Results of the comparison for the A and B rooms are shown on Table 3.7.

Table 3.7. South rooms parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	20.6	829.3	-615.6
B ROOM	20.6	830.0	-596.1
B/A MEAN	100.1%	100.1%	96.8%

There is no difference between both room types in the indoor temperature or the supply airflow. The B room presented a 3% less measured reheat energy that the A room.

Conclusions: For the South Rooms, disagreements are smaller than 4%.

3.3.2.4. West Room

Results of the comparison for the A and B rooms are shown on Table 3.8.

Table 3.8. West rooms parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	20.5	796.9	-832.1
B ROOM	20.5	790.0	-739.9
B/A MEAN	100.1%	99.1%	88.9%

As in the previous cases, no large differences exist between both cell types in the indoor temperature or the supply airflow. However, there are some differences on the reheat energy. Discrepancies as big as 100W have to be assumed.

To determine if those discrepancies are as important as they appear, the ES parameter is used as it has already been done in the previous test. For the other rooms, it has not been necessary because the measurements were so similar that the disagreements are obviously almost negligible. Table 3.9 shows the ES parameters for both West A and B.

Table 3.9. West room cooling energy

	SUPPLY AIR FLOW	ROOM TEMP.	ENTERING TEMP.	ES
	m^3/h	ºC	ºC	Wh
A ROOM	796.9	20.5	17.6	843.5
B ROOM	790.0	20.5	17.1	940.6
B/A MEAN	99.1%	100.1%	97.5%	111.5%

Conclusions: For the West Rooms, discrepancies such as 12% have been observed.

3.4. Comparison Between Experimental Results and Simulation Results

3.4.1. Weather Data

As in the first exercise, the weather data where provided in a TMY format. Each program has its own weather processor. Conclusions and differences on the weather processor have already been analyzed and are similar for every test case.

3.4.2. AHU System A Data

Before analyzing the room results, the simulation of the Air Handling Units will be considered.

3.4.2.1. Temperatures

Three different temperatures have been measured at the Air Handler Unit: supply air temperature leaving the cooling coil, return air temperature, and temperature entering the cooling coil. As the cooling coil was off, the temperature differences between the entering cooling coil and leaving cooling coil is neglected.

- **Supply Air Temperature. Test Defined Parameter**

The supply air temperature was defined as constant at 15.6 ºC. The temperature leaving the cooling coil must be the supply air temperature minus the temperature increase caused by the supply fan and the duct delta-t. As Table 3.10 shows, the mean temperature measured was 14.3 ºC.

The Iowa model presented a large error; 1.78 ºC underestimation. This error is caused by a fan reheat overestimation. As the cooling system in the Air Handling Unit is the economizer, this supply temperature error should also cause an outside air overestimation. Those temperature values should be reviewed.

The other models estimated this temperature very accurately. DOE-CIEMAT, TRNSYS-TUD and IDA-ICE models predicted this temperature almost exactly. The last model's prediction, PROMETHEUS, is accurate within 1°C.

Table 3.10. Leaving cooling coil temperature (ºC)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-3.10	-1.60	-1.20	-1.43	-1.50	
dtmax	1.30	1.30	1.60	1.25	1.30	
meandt	-1.78	-0.07	0.29	-0.07	-0.04	
min	12.30	13.90	14.50	13.90	14.20	12.9
max	15.70	15.50	15.80	15.70	15.60	15.9
mean	12.51	14.23	14.59	14.23	14.25	14.3
abmeandt	1.83	0.17	0.33	0.16	0.17	
rsqmeandt	1.88	0.30	0.41	0.28	0.31	
stderr	0.61	0.30	0.28	0.28	0.31	
stderr/mean	0.05	0.02	0.02	0.02	0.02	
MEAN%	-12%	0%	2%	0%	0%	

Conclusions: The DOE-IOWA model could shows an error on the fan heat gain estimations. The outside airflow would be overestimated.

- **Return Air Temperature. Non Test Defined Parameter**

The return air temperature must be a corrected mean value between the different rooms temperature. If the room temperatures are accurately predicted, this return air temperature must be also accurately predicted. Table 3.11 shows the exactness of these predictions.

Table 3.11. Return air temperature (ºC)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-1.90	-2.60	-2.30	-1.68	-2.36	
dtmax	1.70	0.10	2.60	1.13	0.69	
meandt	0.15	-0.75	0.37	-0.07	-0.54	
min	19.50	18.40	19.30	18.86	18.34	19.3
max	23.00	22.90	23.50	23.29	22.85	23.0
mean	21.30	20.40	21.51	21.07	20.61	21.1
abmeandt	0.64	0.75	0.65	0.53	0.69	
rsqmeandt	0.80	0.94	0.91	0.61	0.89	
stderr	0.78	0.58	0.83	0.61	0.71	
stderr/mean	0.04	0.03	0.04	0.03	0.03	
MEAN%	1%	-4%	2%	0%	-3%	

The largest mean error is 4% (0.75ºC). The largest standard error is given by the PROMETHEUS model. Both errors are very small.

Conclusions: The errors can be neglected for all the models.

- **Supply Airflow. Non Test Defined Parameter**

All the models are accurately predicting the supply airflow. The greatest mean errors is given by the IDA-ICE model (this error is very small, considering that it was a blind simulation). Table 3.12 shows the results obtained.

Table 3.12. Supply airflow by the AHU-A (m^3/h)

	IOWA	CIEMAT	KST	DRESDEN	IDA-ICE	REAL
dtmin	-1037.00	-908.00	-933.00	-838.89	-914.00	
dtmax	549.00	878.00	1102.00	356.28	1036.40	
meandt	-73.78	28.94	44.98	-28.29	58.34	
min	2761.00	2762.00	2762.00	2762.00	2761.55	2758
max	4054.00	4383.00	4607.00	4156.28	4298.97	4037
mean	2839.33	2942.05	2958.09	2884.83	2971.46	2913.1
abmeandt	88.47	67.98	103.46	63.38	130.98	
rsqmeandt	193.61	165.01	218.50	142.49	261.78	
stderr	179.00	162.45	213.82	139.65	255.19	
stderr/mean	0.06	0.06	0.07	0.05	0.09	
MEAN%	-3%	1%	2%	-1%	2%	

As this is a Variable Air Volume system, it is interesting to graphically analyze the variations on the supplied airflow. Figure 3.1 shows those results.

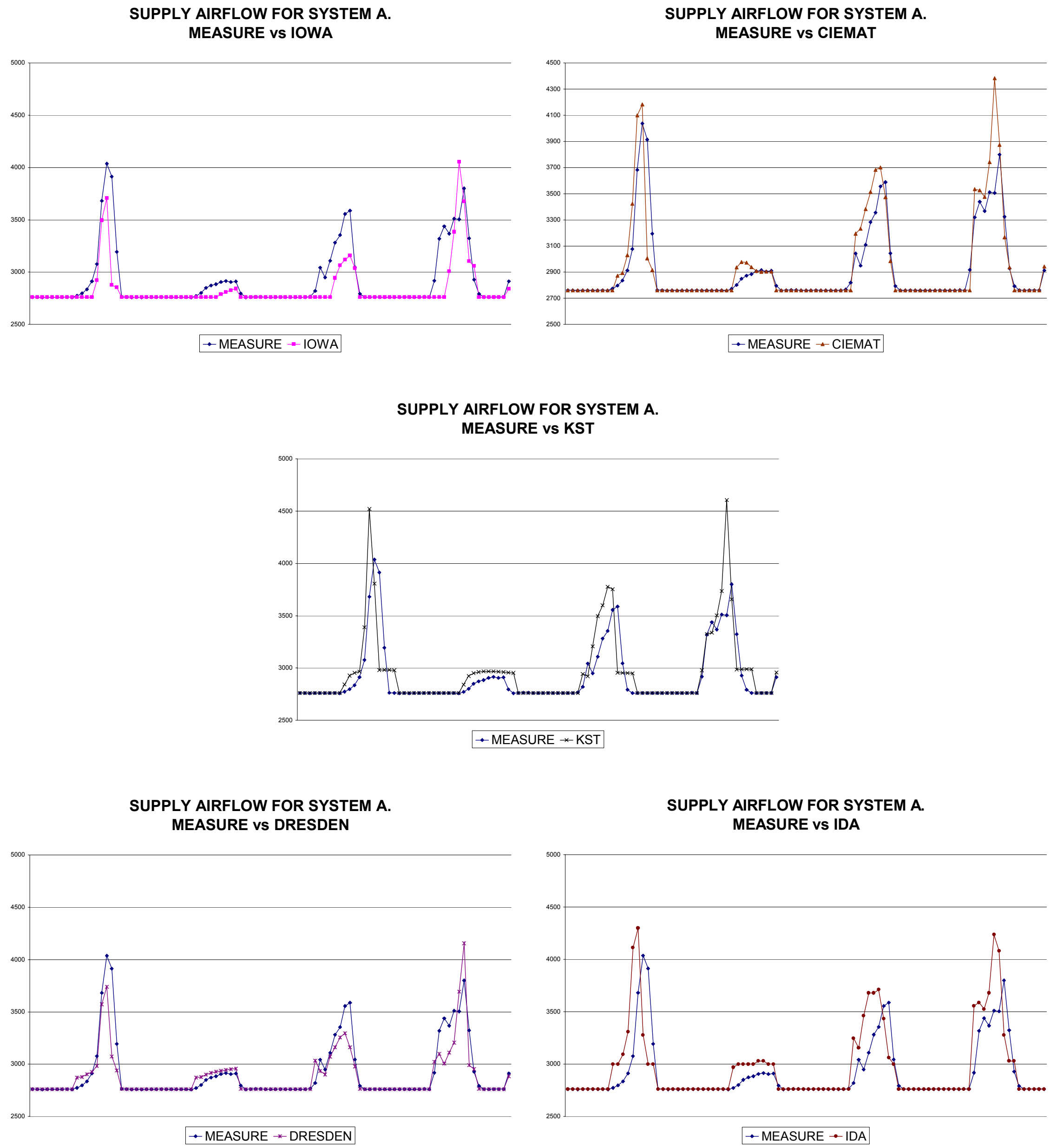

Figure 3.1 Supply air flow-rate for System A

Conclusions: All the models had errors smaller than 3%, which is negligible.

DOE-IOWA and TRNSYS-TUD models present small under-estimations on the large values of the supply airflow.

DOE-CIEMAT and PROMETHEUS models slightly overestimated the airflow when cooling loads are high.

IDA-ICE model might have a "daylight" saving time misinterpretation. It is one hour forward from the measurements. Another possible reason could be the program routine that builds an hourly report, based on variable time steps. See modeller report section 5.5.

- **Outside Air Flow. Non Test Defined Parameter**

The outside airflow is used in this case as the cooling medium. As it has been already explained, the measured airflow cannot be used as a good measurement. Instead of this measurement, a comparison analysis will be done.

As this is a variable air system, the supplied airflow is different for each model. Instead of the outside air, the parameter that will be analyzed is the OUTSIDE AIR RATIO (OAR) defined as:

$$OAR = \frac{\text{Outside air flow}}{\text{Supply air flow}}$$

Results for this parameter are shown in Table 3.13.

Table 3.13. Outside Air Ratio for AHU-A (fractional)

	IOWA	CIEMAT	KST	DRESDEN	IDA	MEASURE
dtmin	-0.12	-0.27	-0.24	-0.18	-0.27	
dtmax	0.29	0.18	0.18	0.17	0.26	
meandt	0.10	-0.04	-0.03	-0.01	-0.01	
min	0.36	0.22	0.23	0.24	0.25	0
max	1.00	1.00	1.00	1.00	1.00	1
mean	0.60	0.45	0.47	0.48	0.49	0.5
abmeandt	0.11	0.08	0.07	0.06	0.08	
rsqmeandt	0.12	0.09	0.09	0.07	0.10	
stderr	0.07	0.08	0.08	0.07	0.10	
stderr/mean	0.11	0.18	0.17	0.15	0.20	
MEAN%	21%	-8%	-5%	-3%	-1%	

All the models show very similar behavior, except for DOE-IOWA which is over predicting the outside airflow ratio and the IDA-ICE which underestimates it. In all other cases, the predictions are smaller than the measurements. This is caused by a heat gain on the ducts on the outside air duct from the inlet to the AHU. Those ducts are not isolated. The outside air is being heated, so the AHU will need more outside air to cool the supplied air. Because of this, it is recommended to analyze this parameter as a comparative test.

This disagreement between DOE-IOWA and the other models is a consequence of the error previously mentioned on the leaving cooling coil temperature. This temperature is underestimated in the DOE-IOWA model. Because of this, the DOE-IOWA model is not used to calculate the mean value of the OAR, which is shown in Table 3.14.

Table. 3.14. Outside Air Ratio for AHU-A, compared to the mean value (fractional)

	IOWA	CIEMAT	KST	DRESDEN	IDA	MEAN
dtmin	-0.02	-0.07	-0.09	-0.07	-0.12	
dtmax	0.22	0.03	0.07	0.14	0.10	
meandt	0.12	-0.02	0.00	0.01	0.02	
min	0.36	0.22	0.23	0.24	0.25	0
max	1.00	1.00	1.00	1.00	1.00	1
mean	0.60	0.45	0.47	0.48	0.49	0.5
abmeandt	0.13	0.02	0.01	0.02	0.03	
rsqmeandt	0.13	0.03	0.02	0.03	0.04	
stderr	0.05	0.02	0.02	0.03	0.04	
stderr/mean	0.08	0.04	0.05	0.06	0.07	
MEAN%	26%	-4%	-1%	2%	3%	

It is more interesting to analyze graphically the variations in the outside air ratio airflow. Figure 3.2 shows those results.

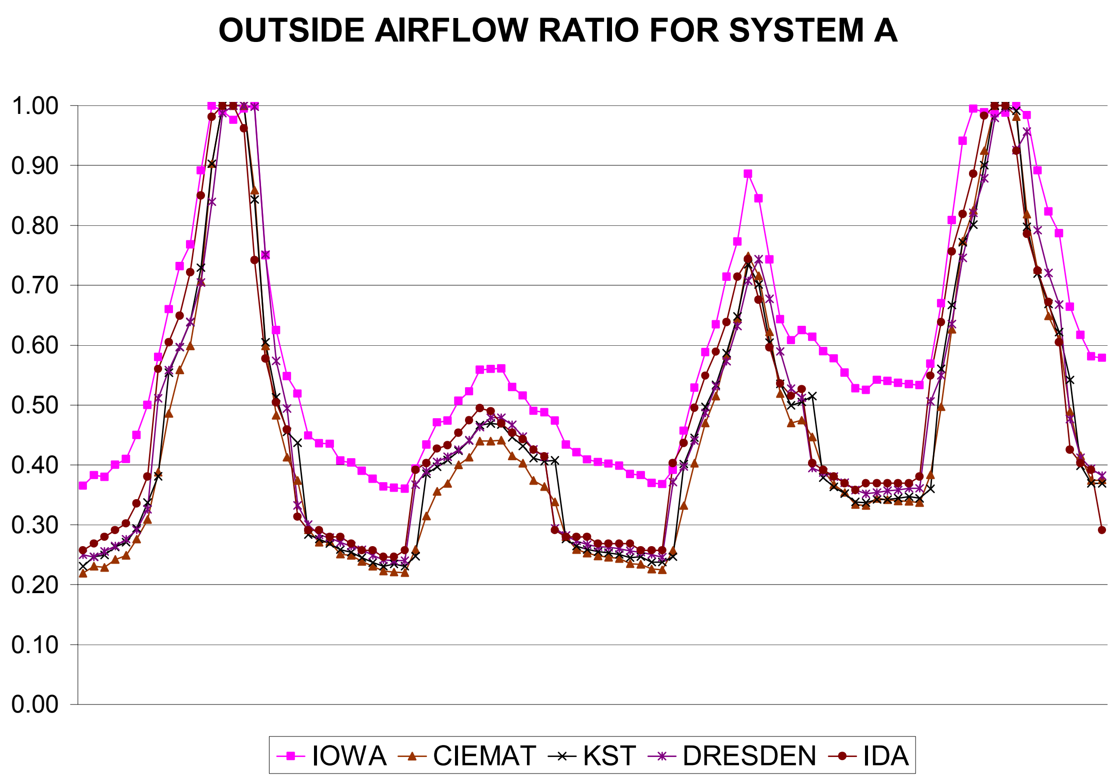

Figure 3.2 Outside air flow-rate ratio

Instead of a cooling coil, the Air Handling Unit is using the outside air to cool the return air. This is done by an economizer. To predict the outside air needed, all the models made an energy balance:

$$\left(m_{Supply} - m_{outside}\right)h_{Return} + m_{outside}h_{outside} = m_{Supply}h_{in\,coil}$$

A good parameter to analyze the differences between the models would be the outside enthalpy given by:

$$h_{outside} = \frac{m_{Supply}h_{in\,coil} - \left(m_{Supply} - m_{outside}\right)h_{Return}}{m_{outside}}$$

This value should be the same for all the programs. A good approximation to this value is:

$$t_{outside}^{approx} = \frac{m_{Supply}t_{incoil}^{approx} - \left(m_{Supply} - m_{outside}\right)t_{return}^{approx}}{m_{outside}}$$

The results obtained for this value are shown in Table 3.15. As it can be observed, all the models presented very similar behavior of the economizer, except for IDA-ICE which is considering always an outside air 1ºC warmer that the rest of the models. This point should be justified in the modeler report (Section 5.5) but could be caused by an intend to consider the heat gain in the outside air duct.

Table 3.15. Outside air Temperature considered on the economizer balance (ºC)

	IOWA	CIEMAT	KST	DRESDEN	IDA	MEAN
dtmin	-0.86	-1.13	-1.14	-1.04	-1.71	
dtmax	0.28	0.32	0.55	1.93	2.71	
meandt	-0.32	-0.41	-0.27	-0.10	1.09	
min	-1.15	-1.40	-1.38	-0.73	1.54	-0.6
max	15.56	15.60	15.80	15.70	15.60	15.6
mean	5.34	5.24	5.38	5.56	6.75	5.7
abmeandt	0.34	0.43	0.31	0.45	1.28	
rsqmeandt	0.39	0.50	0.37	0.57	1.43	
stderr	0.23	0.29	0.25	0.56	0.92	
stderr/mean	0.04	0.06	0.05	0.10	0.14	
MEAN%	-6%	-7%	-5%	-2%	19%	

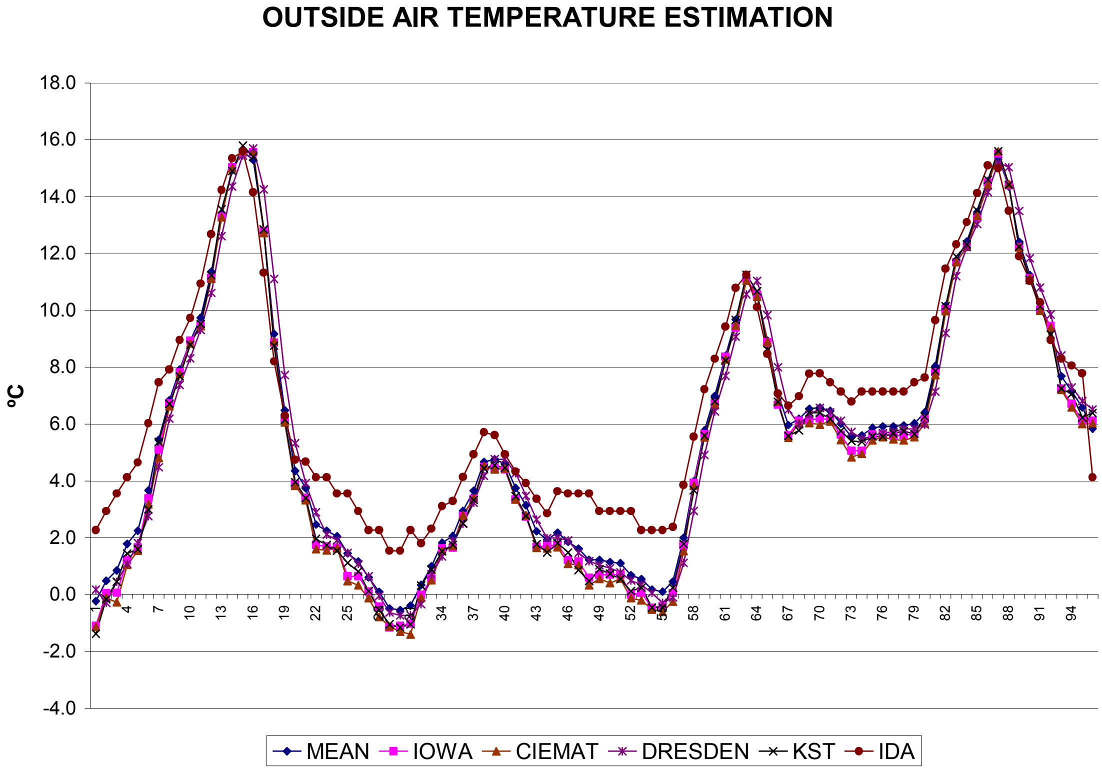

Figure 3.3 Outside air temperature estimation

Conclusions: All the models made very similar predictions and hypotheses for the economizer, except for IDA-ICE which considers a warmer outside air that the others (see modeler report, Section 5.5).

3.4.3. AHU System B Data

3.4.3.1. Temperatures

The same analysis procedure as was done for the system A will be used.

- **Supply Air Temperature. Test Defined Parameter**

The behavior of the models is similar to System A. As Table 3.16 shows, the mean temperature measured was 14.3 ºC.

The Iowa model had some problems to accurately predict this temperature. The Iowa model presented a large error; 1.75 ºC underestimation. This error should an outside air overestimation. The error could be caused by an overestimation on the fan heat. Those temperature values should be reviewed.

Table 3.16. Leaving cooling coil temperature (ºC)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-3.00	-1.50	-1.10	-1.33	-1.52	
dtmax	1.10	1.40	1.70	1.35	1.40	
meandt	-1.71	0.00	0.37	0.01	0.03	
min	12.30	13.90	14.50	13.90	14.20	12.8
max	15.70	15.50	15.80	15.70	15.60	16.0
mean	12.51	14.22	14.59	14.23	14.25	14.2
abmeandt	1.75	0.17	0.41	0.14	0.16	
rsqmeandt	1.80	0.29	0.47	0.27	0.31	
stderr	0.59	0.29	0.28	0.27	0.31	
stderr/mean	0.05	0.02	0.02	0.02	0.02	
MEAN%	-12%	0%	3%	0%	0%	

Conclusions**:** Almost all the models predict accurately the supply air temperature. The DOE-IOWA model presents the same problem as for System A.

- **Return Air Temperature. Non Test Defined Parameter**

The return air temperature must be a corrected mean value between the different rooms temperature. If the room temperatures are accurately predicted, this return air temperature must be also accurately predicted. Table 3.17 shows the exactness of these predictions.

Table 3.17. Return air temperature (ºC)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-1.60	-2.30	-2.10	-1.32	-2.06	
dtmax	1.80	0.30	2.90	1.37	0.89	
meandt	0.41	-0.48	0.63	0.21	-0.29	
min	19.50	18.40	19.30	18.91	18.34	19
max	23.00	22.90	23.50	23.28	22.85	23
mean	21.30	20.40	21.52	21.10	20.60	20.9
abmeandt	0.73	0.52	0.80	0.48	0.59	
rsqmeandt	0.90	0.74	1.02	0.60	0.72	
stderr	0.80	0.57	0.80	0.56	0.67	
stderr/mean	0.04	0.03	0.04	0.03	0.03	
MEAN%	2%	-2%	3%	1%	-1%	

The largest mean error is only 3% (0.63ºC, given by PROMETHEUS). The largest standard error is given by the DOE-IOWA and PROMETHEUS models. Both errors are very small.

Conclusions**:** The errors can be neglected for all the models.

- **Supply Airflow. Non Test Defined Parameter**

All the models, are accurately predicting the supply airflow. The largest error is only 3%, given by DOE-IOWA model. Table 3.18 gives statistical summary of supply air flow-rate from all participants.

Table 3.18. Supply airflow by the AHU-B (m^3/h)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-963.00	-758.00	-813.00	-727.30	-768.07	
dtmax	431.00	796.00	1051.00	450.04	990.47	
meandt	-83.44	26.66	38.85	-38.47	53.84	
min	2761.00	2762.00	2762.00	2762.00	2761.55	2759
max	4054.00	4419.00	4606.00	4163.04	4268.04	3818
mean	2839.33	2949.43	2961.63	2884.30	2976.61	2922.8
abmeandt	95.38	67.70	101.08	68.52	115.97	
rsqmeandt	217.37	164.51	214.44	159.95	234.03	
stderr	200.71	162.34	210.89	155.25	227.75	
stderr/mean	0.07	0.06	0.07	0.05	0.08	
MEAN%	-3%	1%	1%	-1%	2%	

Again, a graphical analysis is better that this numerical one.

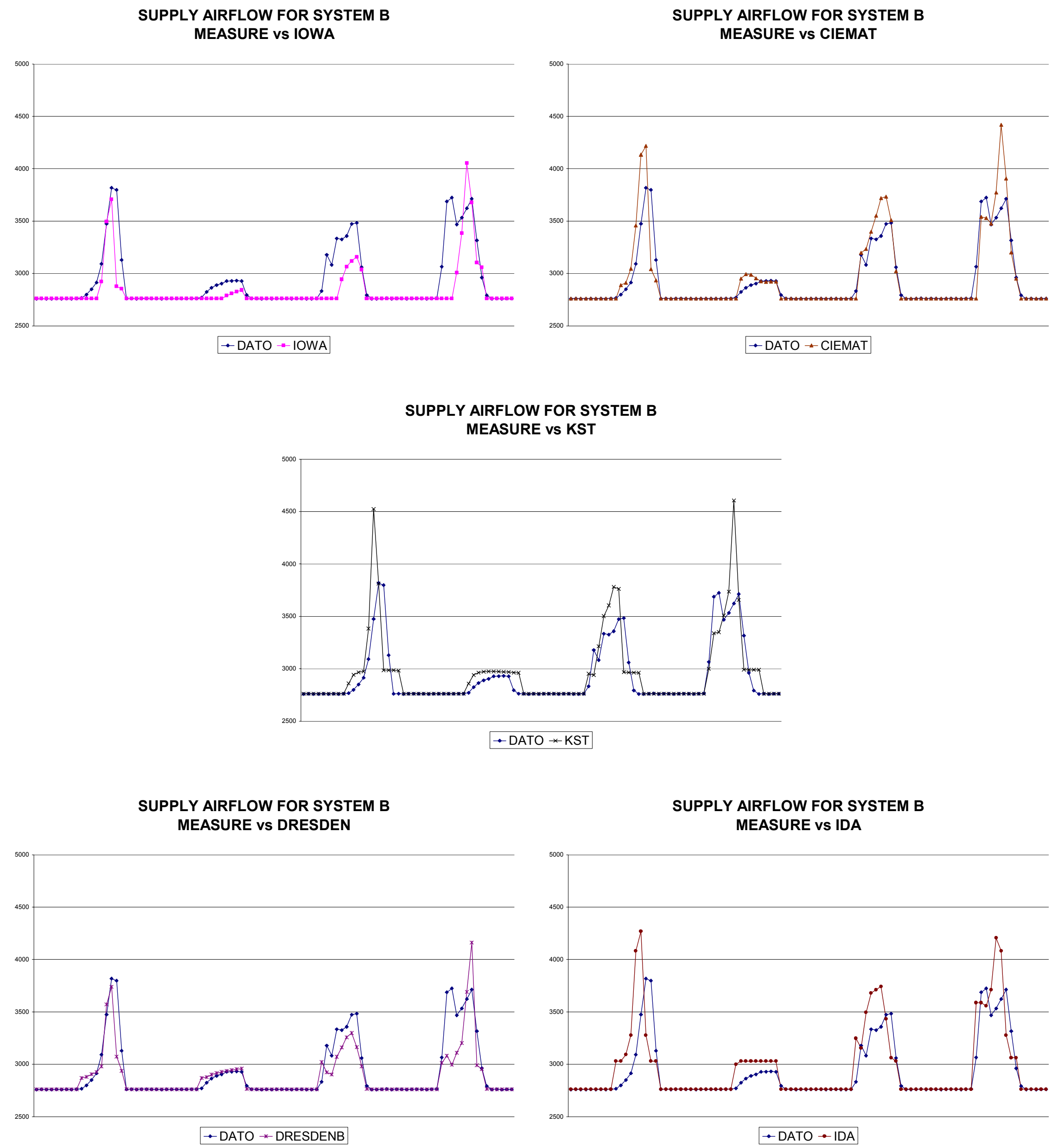

Figure 3.4 Supply air flow-rate for System B

Conclusions: All the models made very good predictions of the supply airflow.

DOE-IOWA model presents small under predictions on the supply airflow.

DOE-CIEMAT, TRNSYS-TUD, IDA-ICE and PROMETHEUS models lightly overestimated the airflow when cooling loads are rising.

- **Outside Airflow. Non Test Defined Parameter**

As has been explained for System A, there was an error in the measured value. The outside airflow will be analyzed as if this was a comparative test. The parameter that will be analyzed is the OUTSIDE AIR RATIO. Results for this parameter are shown in Table 3.19.

Table 3.19. Outside Air Ratio for AHU-B, compared to the mean value (fractional)

	IOWA	CIEMAT	KST	DRESDEN	IDA	MEAN
dtmin	-0.02	-0.07	-0.09	-0.07	-0.12	
dtmax	0.22	0.03	0.07	0.14	0.10	
meandt	0.12	-0.02	-0.01	0.01	0.02	
min	0.36	0.22	0.23	0.24	0.25	0.2
max	1.00	1.00	1.00	1.00	1.00	1.0
mean	0.60	0.46	0.47	0.48	0.49	0.47
abmeandt	0.12	0.02	0.01	0.02	0.03	
rsqmeandt	0.13	0.03	0.02	0.03	0.04	
stderr	0.05	0.02	0.02	0.03	0.04	
stderr/mean	0.08	0.04	0.05	0.06	0.07	
MEAN%	26%	-4%	-1%	2%	3%	

As was explained System A, the differences are due to the different supply air temperatures of the Iowa model.

For a better analysis of the economizer behavior, simulation the outside air prediction of each model can be used. This value should be the same for all the programs. The same behavior that for the A system have been observed.

Table 3.20. Outside air Temperature considered on the economizer balance (ºC)

	IOWA	CIEMAT	KST	DRESDEN	IDA	MEAN
dtmin	-0.88	-1.00	-1.16	-1.05	-1.74	
dtmax	0.28	0.32	0.54	1.93	2.67	
meandt	-0.33	-0.35	-0.29	-0.11	1.09	
min	-1.15	-1.22	-1.38	-0.74	1.54	-0.5
max	15.56	15.60	15.80	15.70	15.60	15.6
mean	5.34	5.32	5.37	5.55	6.75	5.7
abmeandt	0.35	0.37	0.33	0.45	1.26	
rsqmeandt	0.41	0.44	0.39	0.57	1.42	
stderr	0.24	0.26	0.26	0.56	0.91	
stderr/mean	0.04	0.05	0.05	0.10	0.13	
MEAN%	-6%	-6%	-5%	-2%	19%	

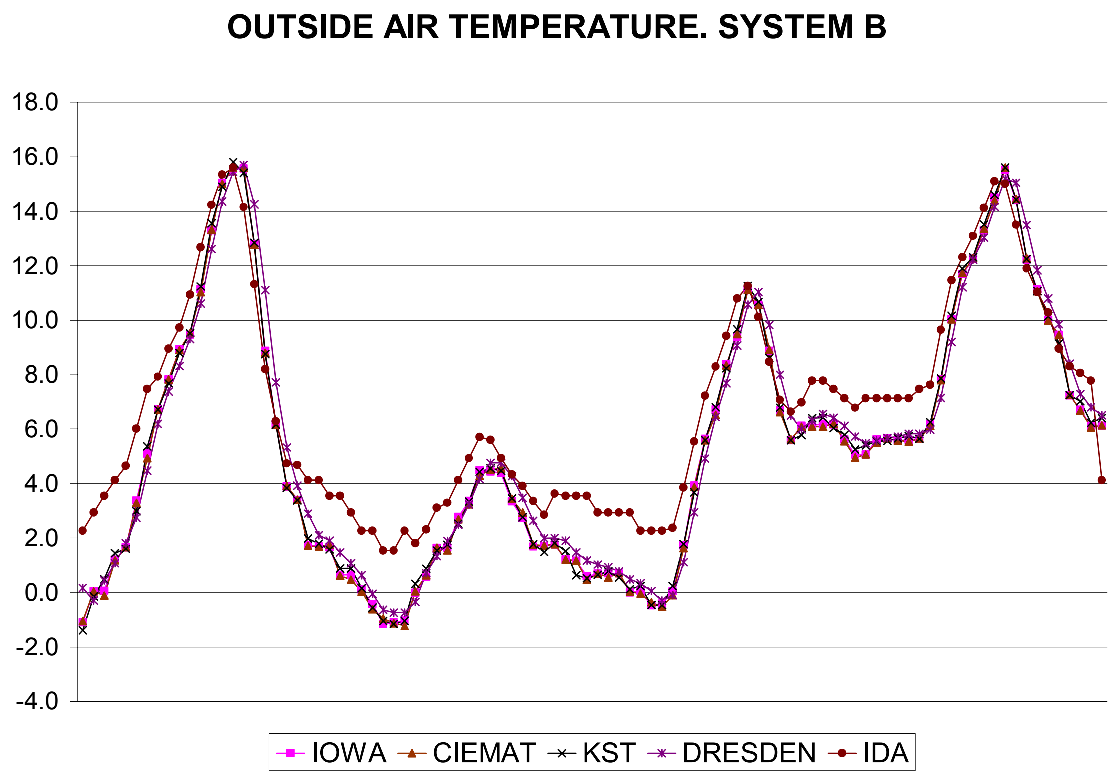

Figure 3.5 Outside air temperature for System B

Conclusions: All the programs estimated the same economizer behavior. Differences are close to 1% except for IDA-ICE, which is probably considering a fan heat (see modeler report, Section 5.5).

3.4.4. Global Reheat Energy and Cooling Energy Supplied Into the Room. Non Test Defined Parameters

The behavior of each room has been analyzed and it is presented in Appendix F. It is interesting to evaluate how big is the global error of the simulations.

3.4.4.1. General Conclusions Common to Every Room

An analysis of the behavior of each room is presented in Appendix F. Some general conclusions, common for every room are made as follows:

- All the models had some problems to estimate the first hour after the night setback of the thermostats.
- The predictions are very accurate, especially for the TRNSYS-TUD, IDA-ICE and DOE-CIEMAT models.
- DOE-IOWA, DOE-CIEMAT and TRNSYS-TUD models accurately estimated mean values and fast dynamics. They only had some problems to estimate the large solar heat

gains. Possible solar radiation gains modeling error or erroneous window specifications could be the cause.

- DOE-IOWA and PROMETHEUS models might be showing too much thermal inertia.

- IDA-ICE model might be one hour in advance from the measurements. The time shift for several hours could be also due to the program routine that builds an hourly report, based on variable time steps. See modeller report Section 5.5.

3.4.4.2. Total Reheat Energy of System A

The reheat energy demanded by the entire System A is presented in Table 3.21 and Figure 3.6

Table 3.21. Electrical Reheat Energy demanded by the System A (W)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-2578.00	-5213.00	-6562.00	-2577.60	-4947.32	
dtmax	3490.00	1873.00	875.00	1207.44	1428.51	
meandt	479.05	-63.59	-744.08	-194.55	-518.62	
min	0.00	0.00	0.00	0.00	-2.29	0.0
max	10195.00	7269.00	8044.00	9493.56	7327.66	11849.0
mean	2882.53	2339.89	1659.40	2208.93	1884.86	2403.5
abmeandt	898.41	873.84	972.96	482.19	841.24	
rsqmeandt	1274.24	1390.02	1855.34	768.90	1387.38	
stderr	1180.76	1388.56	1699.60	743.89	1286.80	
mean%	20%	-3%	-31%	-8%	-22%	
stderr/mean	0.49	0.58	0.71	0.31	0.54	

The DOE-IOWA model presented an overestimation of 20% (480W<>4.8W/m^2). It was caused by an overestimation of the low values and possibly too much thermal inertia.

DOE-CIEMAT model estimated very accurately the mean value (error of 0.6W/m^2), but it over predicting the low values and under predicting the high values.

PROMETHEUS model had an underestimation of 31%, which is almost 750W for the entire building (7.5W/m^2). It presented some problems with fast dynamics.

TRNSYS-TUD model estimated very accurately the fast dynamics. It only had some problems with the high values. The error is only 1.9W/m^2.

IDA-ICE model underestimated the low and the large values, but its error is not very large (5.2W/m^2).

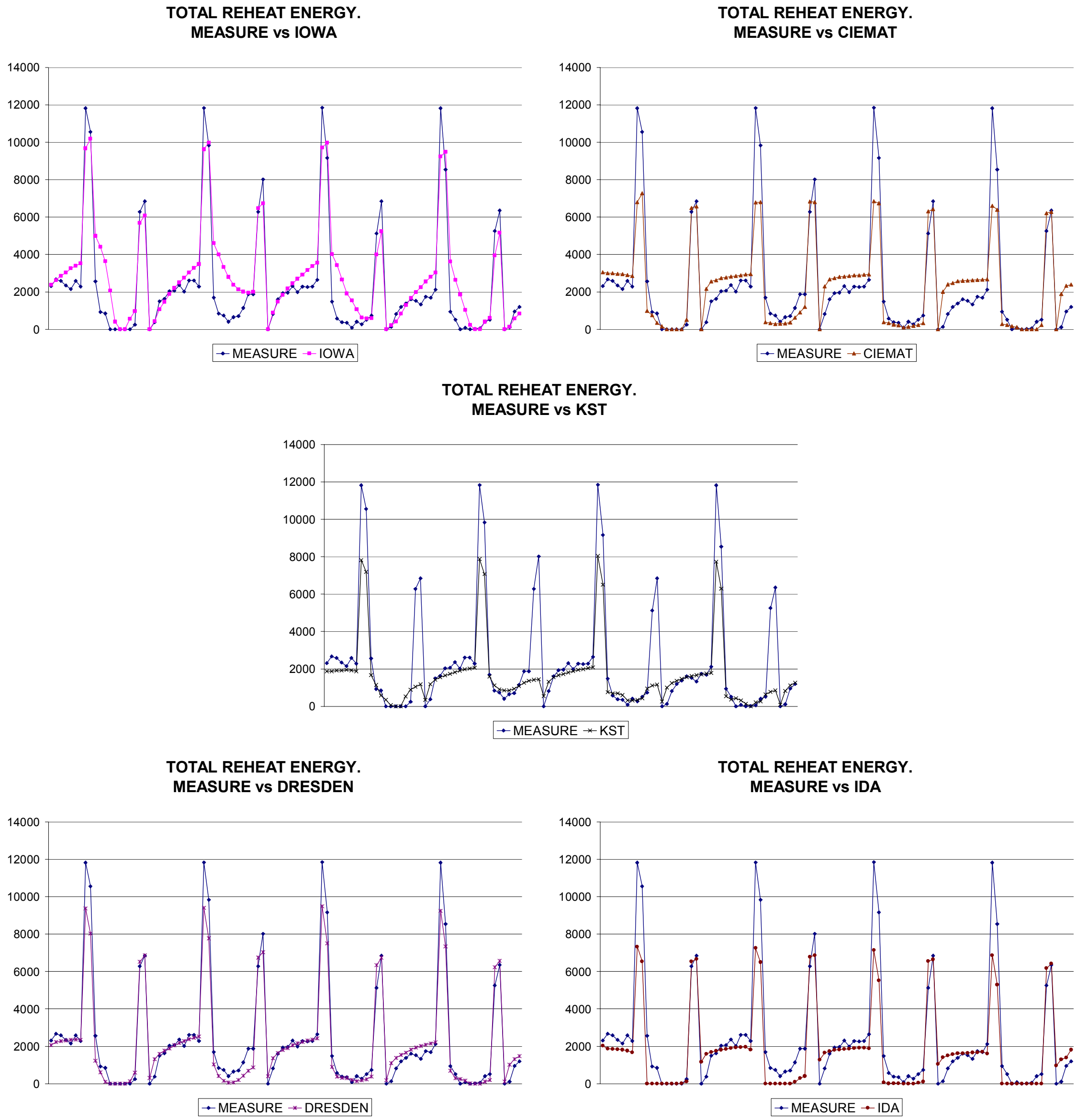

Figure 3.6 Total reheat energy for System A

3.4.4.3. Total Cooling Energy Supplied to A type room

The cooling energy supplied through System A is presented in Table 3.22 and Figure 3.7.

Table 3.22. Cooling energy supplied to the rooms by the System A (J)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-8391.48	-9047.40	-10146.60	-8305.54	-11547.92	
dtmax	22639.20	25807.56	25204.44	17476.68	26460.14	
meandt	3763.63	917.39	1433.71	319.84	2658.27	
min	-5638.44	-1366.44	-3768.96	-7917.33	297.54	-25033.3
max	37554.48	43043.40	37329.12	35277.77	39427.87	38236.2
mean	17615.60	14769.36	15285.68	14171.81	16510.24	13852.0
abmeandt	5854.84	4860.12	4968.12	3191.19	5177.52	
rsqmeandt	7224.05	6929.68	8241.40	4933.14	7683.00	
stderr	6166.19	6868.68	8115.74	4922.76	7208.47	
mean%	27%	7%	10%	2%	19%	
stderr/mean	0.45	0.50	0.59	0.36	0.52	

All the models showed an error whenever the system is turned on, in the morning.

The DOE-IOWA model presented a strange behavior in the mornings and had some problems on the dynamics on the evenings. It could be estimating too much thermal inertia.

DOE-CIEMAT model was very accurate (7% error). It predicted accurately the large and low values and also the fast dynamics.

PROMETHEUS model is very accurate in the mornings but shows too large a cooling load in the early afternoon. It is having also the same problem that the DOE-CIEMAT model has in the early mornings.

TRNSYS-TUD model is very accurate. It showed an error of 2%, estimating very accurately the fast dynamics. It predicted accurately the large and low values and also the fast dynamics. It shows the same problem that the DOE-CIEMAT has, but the error is smaller.

IDA-ICE model overestimated the large values and was not able to estimate the early mornings.

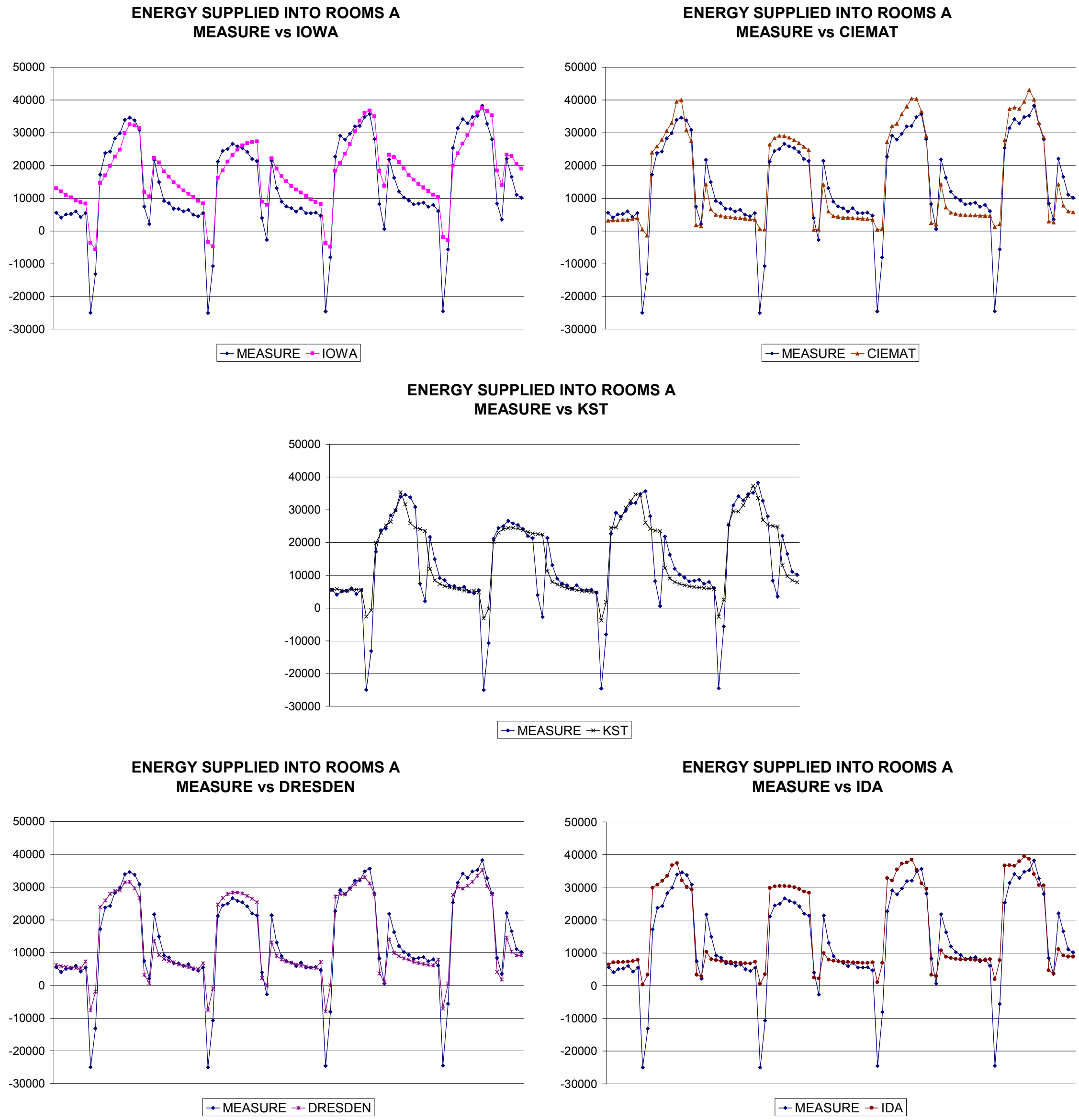

Figure 3.7 Total cooling energy supplied into Rooms A

3.4.4.4. Total Reheat Energy of System B

The reheat energy demanded by the entire System B is presented in Table 3.23 and Figure 3.8.

Table 3.23. Electrical Reheat Energy demanded by System B (W)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-2606.00	-5251.00	-6484.00	-2706.11	-5006.67	
dtmax	3546.00	2132.00	1170.00	1230.51	1578.95	
meandt	602.29	48.79	-574.43	-76.60	-401.76	
min	0.00	0.00	0.00	0.00	-2.29	0.0
max	10195.00	7262.00	8011.00	9486.81	7310.23	11877.0
mean	2882.53	2329.03	1705.81	2203.64	1878.48	2280.2
abmeandt	981.31	1046.75	962.26	589.94	861.82	
rsqmeandt	1306.69	1514.60	1848.85	847.07	1420.80	
stderr	1159.61	1513.82	1757.35	843.60	1362.81	
mean%	26%	2%	-25%	-3%	-18%	
stderr/mean	0.51	0.66	0.77	0.37	0.60	

The DOE-IOWA model presented an overestimation of 26% (602W<>6.02W/m^2). As in System A, it was caused by an overestimation of the low values.

DOE-CIEMAT model estimated very accurately the mean value (0.48W/m^2 error), but it slightly overpredicted the low values and underpredicted the high values.

PROMETHEUS model had an underestimation of 21%, which is almost 575W for the entire building. It presented some problems with fast dynamics.

TRNSYS-TUD model estimated very accurately the fast dynamics. It had a few problems with the high values. Its error is very similar to DOE-CIEMAT´s (0.7W/m^2).

IDA-ICE model underestimated the low and the large values, but its error is not very large (4W/m^2).

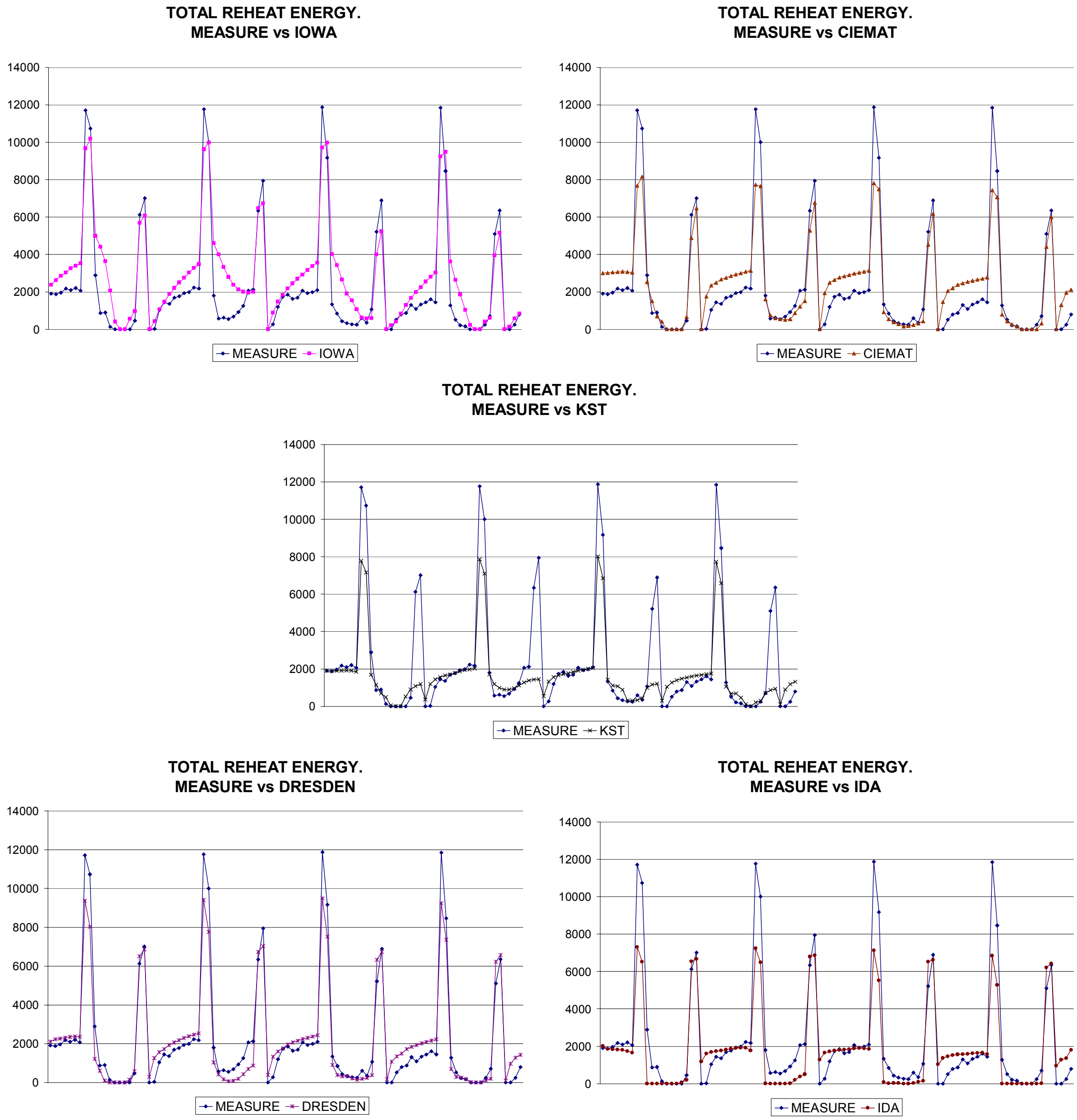

Figure 3.8 Total reheat energy for System B

3.4.4.5. Total Cooling Energy Supplied to B Type Room

The cooling energy supplied through System B is presented in Table 3.24 and Figure 3.9.

Table 3.24 Cooling energy supplied to the rooms by System B (J)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-12346.44	-10355.04	-11925.48	-8537.12	-12256.27	
dtmax	22782.24	25986.60	25249.32	17489.43	26716.04	
meandt	2859.49	192.49	43.54	-579.82	1776.19	
min	-5638.44	-1341.24	-3650.16	-8028.88	360.29	-24674.3
max	37554.48	43531.80	36105.24	35277.85	38853.36	37883.9
mean	17615.60	14948.60	14799.66	14176.29	16532.30	14756.1
abmeandt	5230.76	5532.49	6046.06	3724.59	5137.38	
rsqmeandt	6818.93	7360.35	8678.60	5182.38	7700.90	
stderr	6190.41	7357.83	8678.49	5149.84	7493.26	
mean%	19%	1%	0%	-4%	12%	
stderr/mean	0.42	0.50	0.59	0.35	0.51	

All the models show an error when the system is turned on in the morning.

The DOE-IOWA model presented a strange behavior in the mornings and had some problems on the dynamics on the evenings. It might have too much thermal inertia.

DOE-CIEMAT model was very accurate (1% error). The difference with the measurements was mainly caused when the system was turned on.

PROMETHEUS model predicted almost exactly the mean value. It predicted very well both, mean value and fast dynamics, but still shows the error in the early mornings.

TRNSYS-TUD model is very accurate. It showed an error of only 4%, estimating very accurately the fast dynamics.

IDA-ICE model overestimated the large values, which caused an overestimation on the mean value of 12%.

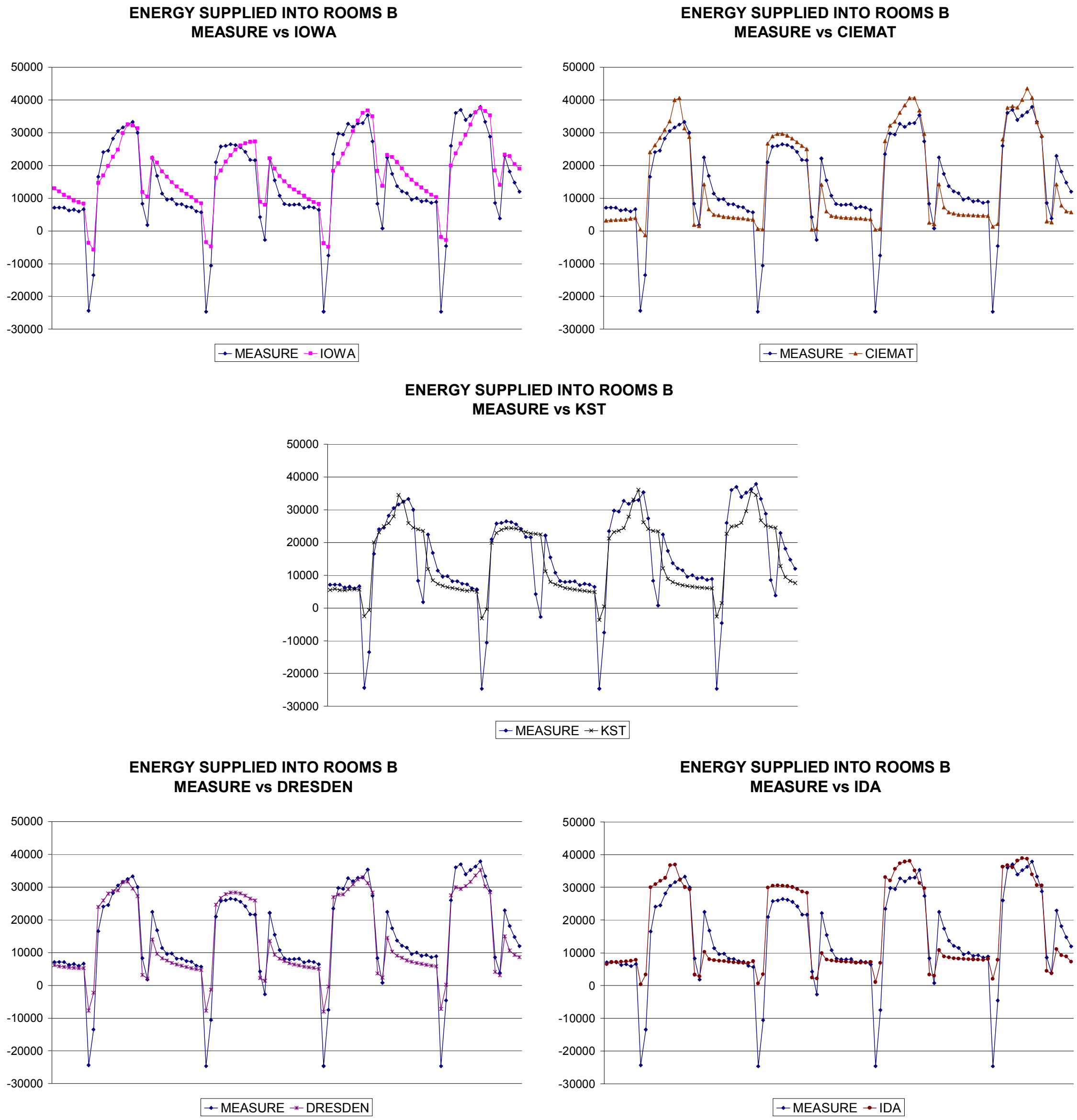

Figure 3.9 Total cooling energy supplied into Rooms B

3.5. Discussion of the Results and Conclusions

All the models accurately predicted the temperatures in the AHU. Only the DOE-IOWA model showed an error in the input, considering a different supply air temperature than the one defined in the test specifications.

As this is a Variable Air Volume case, the supply airflow is one of the most important parameters to be considered. All the models presented errors smaller that 3%, which is negligible.

- DOE-IOWA and TRNSYS-TUD models present small under-estimations on the large values of the supply airflow.

- DOE-CIEMAT and PROMETHEUS models slightly over-predicted the airflow for high cooling loads.

- IDA-ICE model might have a "daylight" saving time misinterpretation. It could be one hour forward from the measurements. The time shift for several hours could also be due to the program routine that builds an hourly report, based on variable time steps. See modeller report Section 5.5.

All the models simulated similar economizer behavior. Only IDA-ICE had some disagreements with the other models. The reason is properly explained in the modeler report, Section 5.5.

After the room analysis, the following conclusions can be made:

- All the models had some problems in estimating the first hour after the night setback of the thermostats.

- The predictions are very accurate, especially for TRNSYS-TUD, IDA-ICE and DOE-CIEMAT models.

- DOE-IOWA and PROMETHEUS models might be showing too much thermal inertia.

Some conclusions may also be drawn from the analysis of the simulation for the entire building:

- All the models showed an error when the system was turned on in the morning.

- The DOE-IOWA model presented a strange behavior in the mornings and had some problems on the dynamics on the evenings. It might have too much thermal inertia.

- DOE-CIEMAT, PROMETHEUS and TRNSYS-TUD models were very accurate. The differences are caused by the calculation error when the system is turned on.

- IDA-ICE model lightly over-predicted the large values.

Simulation results showed good agreement with measurements for all the models. Some of them were very accurate and made a good prediction of real behavior.

4. THIRD EXERCISE. VERY DYNAMIC CASE (Variable Air Volume System with Variable Internal Loads and Scheduled System)

4.1. Description of the Exercise

This section contains information regarding the operating parameters and conditions used for a VAV test conducted at the Iowa Energy Center's Energy Resource Station as part of the empirical validation study for the International Energy Agency Task 22. The test was conducted over a five day period from June 12-16,1999.

For this test, the "A" and "B" test rooms were operated utilizing Variable-Air-Volume ReHeat (VAVRH). Electric heating coils were used in the rooms to provide terminal heating. The air handling units were operated without utilizing outdoor air economizer mode. The air handling units were operated with 100 % return air. The chiller was available throughout the test, but the system supply and return fans were scheduled to be off during the unoccupied period.

Another feature of this test was the use of thermostat schedules as well as scheduled internal sensible load for the test rooms as shown in Table 4.1. The thermostats in the test rooms were programmed for a night setback temperature. The electric baseboard heaters in the test rooms were programmed to come on during the day to provide a scheduled internal load. The schedule is also shown in Table 4.1. Table 4.2 gives the values of baseboard heater power for the different stages. The lights in the test rooms were turned off. The thermostats in the rest of the ERS were programmed for a constant set point schedule. Hence, the temperature in the spaces adjacent to the test rooms remained fairly constant during the test. The thermostats in the spaces adjacent to the test rooms were set at 22.7 °C. Specific adjacent space temperature data are provided in the file 990612adjtemp.dat. This file contains hourly temperature data.

4.1.1. Run Period and General Weather Conditions

This item is used to specify the initial and final dates of the desired simulation period and also the general conditions and location of the ERS facility.

- The dates for this test are: June 12, 1999 through June 18,1999.
- Weather data for Ankeny, Iowa is organized in a TMY format. In this file the measured data for the dates previously specified are included. This file is called "Ankeny.ia1" and is included with this report.
- Building Location
    - LATITUDE: 41.71 degrees North
    - LONGITUDE: 93.61 degrees West

- ALTITUDE: 938.0 feet (285.9 m)
- TIME-ZONE: 6, central time zone in U.S.
- DAYLIGHT-SAVINGS: YES

4.1.2. Test Room Operation and Control Parameters

4.1.2.1. Internal Loads and General Room Conditions

A baseboard heater is installed inside each test room. The baseboard heaters were used to simulate internal loads in the test rooms for this test (additional information about the baseboard heaters is provided in the Appendix C). The internal loads scheduled during the test are shown in Table 4.1a, and the baseboard heating capacity is shown in Table 4.2.

Besides the baseboard heaters, other general room characteristics must be considered:

- No lights or miscellaneous equipment other that the baseboards.
- No shading device on windows.
- No infiltration.

4.1.2.2. Room HVAC specifications

Each test room has its own thermostat and some HVAC specifications can be considered.

- **Thermostat Schedule**

Each test room has its own thermostat and the set point value is the same for all test rooms. These values were used for both tests.

- Design heat temperature: 22.2 °C
- Design cool temperature: 22.8 °C
- Heat temperature schedule: see Table 4.1
- Cool temperature schedule: see Table 4.1
- Internal loads schedule: see Table 4.1
- Dead-ban: 1.7 °C

Table 4.1 Set point temperature and internal loads schedules

Hour	Cooling Set-point temperature (°C)	Heating set-point temperature (°C)	Internal loads (stage of BB heat)
1-7	32.2	12.8	0
7-12	22.8	22.2	2
12-13	25.0	17.8	1
13-18	22.8	22.2	2
18-24	32.2	12.8	0

Table 4.2 Baseboard heater power for different stages

Rooms	Stage 1 (W)	Stage 2 (W)
East A	0.900	0.880
East B	0.875	0.845
South A	0.885	0.875
South B	0.870	0.875
West A	0.855	0.845
West B	0.885	0.885
Interior A	0.865	0.880
Interior B	0.915	0.900

- **Room Airflow and Reheat Specifications**

The airflow rates were specified for each test room .

- Exterior test rooms (east, south and west)

  Unoccupied : max 0 m^3/hr, min 0 m^3/hr
  Occupied : max 1699 m^3/hr, min 765 m^3/hr

- Interior test rooms

  Unoccupied : max 0 m^3/hr, min 0 m^3/hr
  Occupied : max 934 m3/hr, min 459 m3/hr

- The installed zone heat source is as follows: 2 stage electric, max 3.34 kW (1.67 kW/stage) for exterior rooms and max 2 kW (1 kW/stage) for interior rooms.

4.1.3. Air Handling Unit Operation and Control

Both AHUs (A and B) were working in the same conditions to supply air to the four sets of test rooms.

4.1.3.1. Set Points and System Controls

The air handling system parameters were specified as follows.

- Supply air temperature.

  East-A : max 22.5 °C, min 13.2 °C
  South-A : max 23.1 °C, min 13.7 °C
  West-A : max 24.0 °C, min 13.5 °C
  Interior-A : max 22.6 °C, min 13.3 °C

- Heating schedule: NOT available
- Cooling schedule: 24 hours available
- Cool control: supply air set point, 13.3 °C after the fan
- Preheat: NOT available
- Humidity control: NOT available
- Economizer: NOT available
- Outside air control: NOT available

4.1.3.2. System Air and Fans

System airflow rates were specified as follows:

- Supply air flow: max 6116 m^3/hr
- Return air path: Plenum
- Minimum outside air flow : NOT available
- Outside air control: NOT available
- Duct air loss: None
- Duct heat gain: 0.3 °C (increase)

The air handling unit fans were specified as follows:

- Supply air static pressure: 1.4 inch H_2O
- Fan schedule: OFF between midnight and 7 hours

  ON between 7 and 18 hours

OFF between 18 and 24 hours

- Supply Fan control: Duct static pressure of 1.4 inch H_2O
- Return Fan control: 90 % of supply fan speed
- Motor placement: In-Air flow
- Fan placement: Draw-Through

4.2. Participating Organizations

Five sets of results were developed with four different computer programs. The participating organizations and models are identified in Table 4.3.

Table 4.3 Participants

Notation	Program	Implemented By	Date of simulation/round
TRNSYS-TUD	TRNSYS-TUD (modified V.14.2)	University of Dresden Dresden, Germany	March 2000/3rd round
PROMETHEUS	PROMETHEUS	Klima System Technik Berlin, Germany	October 1999/2nd round
DOE-IOWA	DOE-2.1E	Iowa State University Ames, Iowa	March 2000/3rd round
DOE-CIEMAT	DOE-2.1E (V.088)	DOE-CIEMAT Madrid, Spain	June 2000/4rd round
IDA-ICE	IDA-ICE (V.2.11.06)	Hochschule Technik + Architektur Luzern, Switzerland	June 2000/4rd round

4.3. Comparison between A and B room type measurements

As has been done in previous exercises, the measured results obtained for the A and B rooms are compared in order to find possible measurement errors.

For the test conducted, the pair of rooms, called “A” and “B”, systems were operated in an identical manner. This should cause identical results for both room types and systems. In some cases, the measurements are different for each room type, and those differences are not negligible.

Before analyzing the accuracy of the model predictions, the errors associated with the measurements must be considered.

4.3.1. Systems Comparison

The first step in comparing both room type measurements is to analyze the systems behavior. If the central system is supplying the air to each room at different temperatures, this should cause different reheat needs in each test room of a pair.

The parameters used for the analysis are the same as in the previous exercises.

Table 4.4 shows the measured results.

Table 4.4. Comparison between system measures

	SUPPLY AIR	TEMP ENTERING COIL	TEMP LEAVING COIL	OUTSIDE AIR	TEMP RETURN AIR	COOLING
	m^3/h	ºC	ºC	m^3/h	ºC	W
A SYSTEM	1339	21.46	15.78	63.5	22.95	4215.4
B SYSTEM	1324	21.34	15.81	35.4	22.69	4451.3
B/A MEAN VALUE	98.87%	99.44%	100.20%	55.70%	98.84%	105.59%

These results show how both systems are operating in the same conditions. The supply air temperature, the temperatures entering and leaving the coil and the return air temperature are very similar.

The outside airflow was supposed to be zero. The measurements have to be assumed as measuring errors or air leakage and can be neglected.

The cooling energy disagreements are only 5%. As for the outside airflow, it has been found that the minimum value is not zero, small values have been measured when the system is off. These values have to be assumed as measurement errors. If the measurement errors of the cooling energy for low water flows are corrected, the differences remain at 5%. When the cooling energy is below 450W, it will be assumed as a zero, because it might be a measurement error.

Conclusions: Both systems are operating in the same conditions. Differences are smaller than 5%.

4.3.2. Rooms Comparison

4.3.2.1. Interior Room

Table 4.5 shows the comparison between the A and B type interior room temperatures, supply airflow and reheat energy.

Table 4.5. Interior Room Parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	22.4	299.1	14.4
B ROOM	22.2	332.3	24.4
B/A MEAN	98.8%	111.1%	169.4%

Big differences exist between both room types. This is caused by errors on sensors for very small measurements, as it happened with the outside airflow.

To solve this effect, two corrections will be done: 1) the low values of supply airflow (values lower that minimum value defined on the VAV box) are assumed as zero, and 2) reheat energy lower that 10W is also assumed as zero. Considering these two effects, the results obtained are shown in Table 4.6.

Table 4.6. Corrected interior room parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	22.4	241.9	12.9
B ROOM	22.2	224.9	23.0
B/A MEAN	98.8%	93.0%	178.2%

The temperature and supply airflow differences are very low. The reheat disagreements are very high if the relative error is considered, but it is a very low absolute error (only 10W).

The parameter defined previously, called ES will be used again to confirm if both room types are having the same thermal behavior. Table 4.7 shows this parameter and that the disagreements are less that 8%, which could be caused by the adjacent room temperatures.

Table 4.7. Interior room cooling energy

	SUPPLY AIR FLOW	ROOM TEMP.	ENTERING TEMP.	ES
	m^3/h	ºC	ºC	Wh
A ROOM	241.9	22.4	11.9	1925.7
B ROOM	224.9	22.2	11.8	1781.3
B/A MEAN	93.0%	98.8%	99.9%	92.5%

Conclusions: For the Interior Rooms, disagreements are 8%.

4.3.2.2. East Room

Results of the comparison for the A and B East Rooms are shown on Table 4.8.

Table 4.8. East Room parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	22.2	485.1	62.3
B ROOM	22.4	527.2	29.3
B/A MEAN	100.8%	108.7%	47.1%

As in previous exercises, these measurements must be corrected. After this correction, the results obtained are shown in Table 4.9

Table 4.9. Corrected East room parameters.

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	22.2	385.9	61.5
B ROOM	22.4	382.4	29.0
B/A MEAN	100.8%	99.1%	47.2%

As for the Interior Room, the best option to check the thermal behavior is to review the ES parameter.

Table 4.10. East room cooling energy

	SUPPLY AIR FLOW	ROOM TEMP.	ENTERING TEMP.	ES
	m^3/h	ºC	ºC	Wh
A ROOM	385.9	22.2	11.5	3083.4
B ROOM	382.4	22.4	11.9	2957.0
B/A MEAN	99.1%	100.8%	103.6%	95.9%

Conclusions: The disagreements for the temperature and the supply airflow are very low, less than 1%, while the errors on reheat energy are 36W. The ES disagreements are smaller that 5%.

4.3.2.3. South Room

Results of the comparison for the A and B rooms are shown on Table 4.11.

Table 4.11. South room parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	22.2	458.4	202.6
B ROOM	22.1	474.8	186.7
B/A MEAN	99.8%	103.6%	92.2%

Once again those measures must be corrected.

Table 4.12. Corrected south room parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	22.2	349.5	201.9
B ROOM	22.1	349.7	186.3
B/A MEAN	99.8%	100.1%	92.3%

Finally, the ES parameter is shown in Table 4.13.

Table 4.13. South room cooling energy

	SUPPLY AIR FLOW	ROOM TEMP.	ENTERING TEMP.	ES
	m^3/h	ºC	ºC	Wh
A ROOM	349.5	22.2	10.3	3032.9
B ROOM	349.7	22.1	10.6	2954.5
B/A MEAN	100.1%	99.8%	102.7%	97.4%

Conclusions: No important differences exist between both room types in the indoor temperature or the supply airflow. The reheat disagreements are only 15W. The ES behavior is almost identical.

4.3.2.4. West Room

Results of the comparison for the A and B rooms are shown on Table 4.14

Table 4.14. West room parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	22.7	499.2	154.6
B ROOM	22.8	499.3	120.2
B/A MEAN	100.5%	100.0%	77.7%

Which, once they have been corrected, as shown in Table 4.15:

Table 4.15. Corrected West room parameters

	TEMPERATURE	SUPPLY AIR FLOW	REHEAT
	ºC	m^3/h	W
A ROOM	22.7	356.8	162.4
B ROOM	22.8	356.8	127.1
B/A MEAN	100.5%	100.0%	78.3%

The ES parameter is shown on Table 4.16.

Table 4.16. West room cooling energy

	SUPPLY AIR FLOW	ROOM TEMP.	ENTERING TEMP.	ES
	m^3/h	ºC	ºC	Wh
A ROOM	356.8	22.7	10.6	3005.3
B ROOM	356.8	22.8	11.1	2899.1
B/A MEAN	100.0%	100.5%	104.2%	96.5%

Conclusions: No important differences exist between both room types in the indoor temperature or the supply airflow. The reheat disagreements are only 35W. The Es parameter shows disagreements lower that 4%.

4.4. Comparison Between Experimental Results and Simulation Results

4.4.1. Weather Data

As for the first case, the weather data where provided in a TMY format. Each program has its own weather processor. Conclusions and differences on the weather processor have already been analyzed and presented.

4.4.2. AHU A System Data

Before analyzing the rooms results, the results for the Air Handling Units will be considered.

4.4.2.1. Temperatures

Three different temperatures have been measured at the Air Handler Unit: temperature leaving the cooling coil, return air temperature, and temperature entering the cooling coil.

- **Supply Air Temperature. Test Defined Parameter**

The supply air temperature was defined as constant at 13.3ºC. The temperature leaving the cooling coil must be the supply air temperature minus the temperature increase caused by the supply fan and the duct delta-t. As Table 4.17 shows, the mean temperature measured was 12ºC.

All the models estimated accurately this temperature.

Table 4.17. Leaving cooling coil temperature (ºC)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-12.07	-0.89	-1.00	-1.70	-2.19	
dtmax	14.80	0.47	0.97	0.27	0.00	
meandt	1.86	-0.07	0.50	-0.23	-0.72	
min	0.00	0.00	0.00	0.00	0.00	0.0
max	14.80	12.80	12.60	11.80	11.31	13.5
mean	13.89	11.96	12.53	11.80	11.31	12.0
abmeandt	4.03	0.20	0.64	0.30	0.72	
rsqmeandt	5.97	0.27	0.67	0.51	0.85	
stderr	6.02	0.27	0.70	0.52	0.90	
stderr/mean	0.43	0.02	0.06	0.04	0.08	
MEAN%	15%	-1%	4%	-2%	-6%	

Conclusions: The errors can be neglected. DOE-IOWA model shows an input error, it is considering a different supply air temperature (probably around 15.5ºC). It should be checked. It should overestimate the supply airflow and underestimate the reheat needs and the cooling loads on the AHU.

- **Return Air Temperature. Non Test Defined Parameter**

The return air temperature must be a corrected mean value between the different rooms' temperature. If the room temperatures are accurately predicted, this return air temperature must be also accurately predicted. Table 4.18 shows the exactness of these predictions.

Table 4.18. Return air temperature (ºC)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-22.98	-2.35	-3.05	-1.52	-2.47	
dtmax	24.40	0.26	1.28	0.46	0.20	
meandt	0.82	-0.61	0.00	-0.28	-0.55	
min	0.00	0.00	0.00	0.00	0.00	0.0
max	24.40	22.80	24.30	22.84	22.63	23.3
mean	23.53	22.11	22.72	22.43	22.17	22.7
abmeandt	4.98	0.62	0.55	0.38	0.56	
rsqmeandt	9.95	0.79	0.89	0.49	0.77	
stderr	9.95	0.83	0.89	0.51	0.80	
stderr/mean	0.42	0.04	0.04	0.02	0.04	
MEAN%	4%	-3%	0%	-1%	-2%	

The largest mean error is given by the DOE-IOWA model and is around 4%, which means less that 1ºC. The largest standard error is given also by this model.

All the models showed errors smaller than 1ºC.

Conclusion: The errors can be neglected for all the models.

- **Supply Airflow. Non Test Defined Parameter**

All the models, except the DOE-IOWA, are accurately predicting the supply airflow. The error in the DOE-IOWA model is consistent with the supply air temperature defined. Those errors are in overestimating the supply air temperature. Table 4.19 shows the results obtained.

Table 4.19. Supply airflow for AHU-A (m^3/h)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-3223.15	-371.03	-501.68	-391.33	-333.98	
dtmax	3246.00	184.34	217.74	39.98	285.45	
meandt	349.05	-60.89	-82.36	-92.23	-55.40	
min	0.00	0.00	0.00	0.00	0.00	0.0
max	4212.00	3293.00	3207.00	3229.12	3154.64	3420.4
mean	3270.62	2860.67	2839.20	2829.33	2866.17	2921.6
abmeandt	933.20	91.72	107.91	95.96	87.15	
rsqmeandt	1367.12	119.73	177.10	123.91	121.61	
stderr	1375.20	122.51	180.55	130.00	123.89	
stderr/mean	0.42	0.04	0.06	0.05	0.04	
MEAN%	12%	-2%	-3%	-3%	-2%	

As this is a Variable Air Volume system, it is interesting to graphically analyze the variations in the supplied airflow. Figure 4.1 shows those results.

Figure 4.1 Supply air flow-rate for System A

Conclusion: DOE-IOWA model presents large over predictions in the supply airflow. This is caused by the supply air temperature over estimation. The other models accurately predicted the airflow, with errors less that 3%.

- **Cooling Coil. Non Test Defined Parameter**

Cooling load simulated and measured are given by Table 4.20

Table 4.20. Cooling load for AHU-A (Wh)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-9957.08	-482.50	-1469.45	-1034.85	-919.27	
dtmax	10108.00	2074.16	2795.15	1486.32	3078.07	
meandt	1456.11	671.02	725.56	150.60	1153.76	
min	0.00	0.00	0.00	0.00	0.00	0.0
max	13448.00	12316.00	11790.00	10955.60	11613.81	11166.6
mean	10573.33	9788.24	9842.78	9267.83	10270.98	9117.2
abmeandt	3198.34	734.87	993.34	507.21	1248.14	
rsqmeandt	4365.83	866.15	1160.98	623.55	1402.99	
stderr	4409.76	912.19	1201.50	626.85	1486.75	
stderr/mean	0.42	0.09	0.12	0.07	0.14	
MEAN%	16%	7%	8%	2%	13%	

Once again, the DOE-IOWA model shows an error of approximately 15%. It also shows an input error in the schedules of the system. It is turned on and off one hour before it was supposed to be.

DOE-CIEMAT and PROMETHEUS results are very similar. They slightly overestimated the large values.

TRNSYS-TUD model predicted very accurately the mean value and the fast dynamics.

IDA-ICE model shows an almost constant behavior on the cooling loads. It is less sensitive to the solar radiation or other weather conditions than the other models.

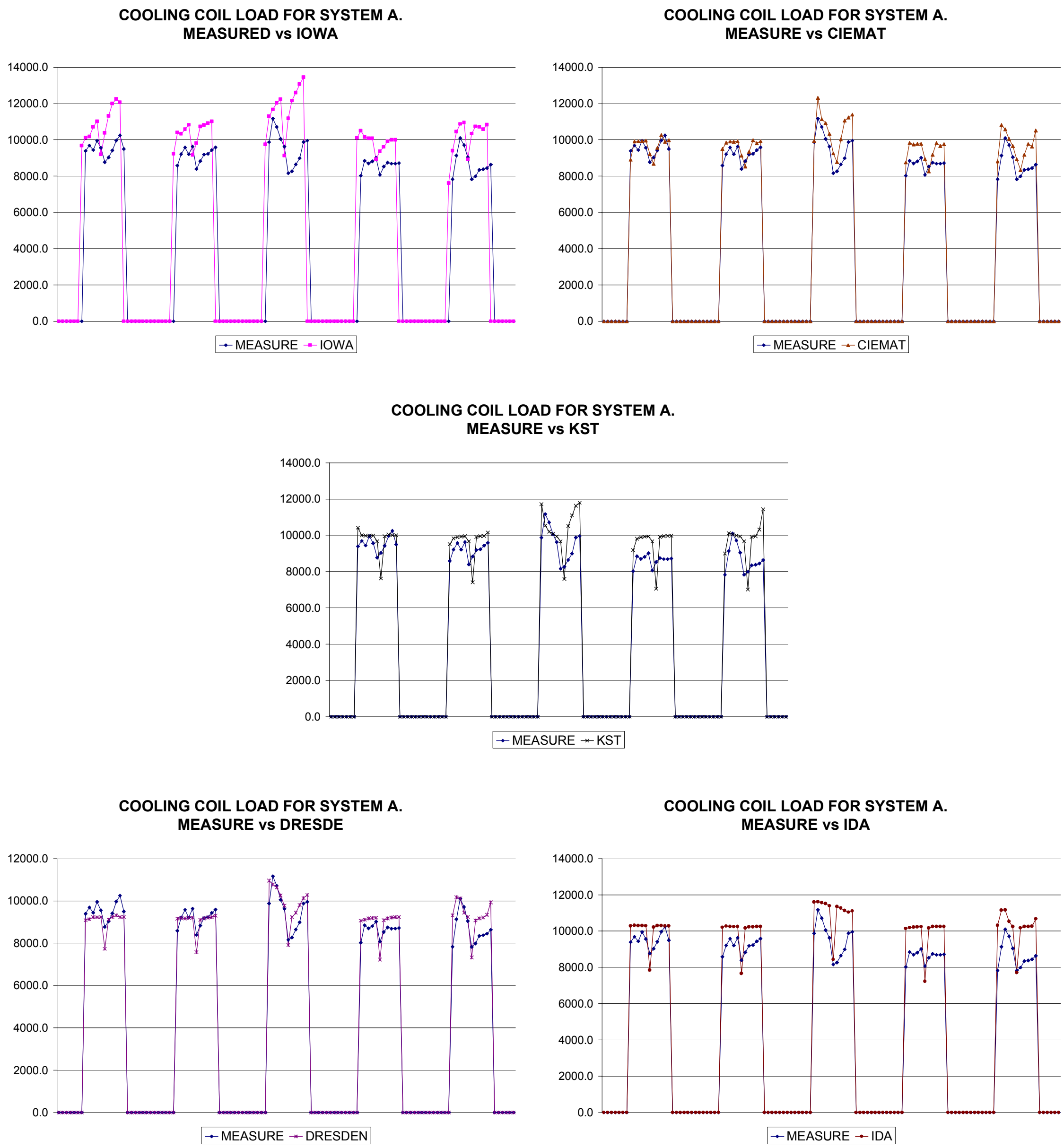

Figure 4.2 Cooling coil load for System A

The difference between the measurement value and the mean value could be caused by an water flow measuring error of 0.18 m^3/h or a temperature difference measuring error on the water side of 0.14ºC, which could be caused by a temperature measurement error of 0.7ºC.

Conclusions: DOE-IOWA and IDA-ICE models showed important disagreements.

The other models, especially TRNSYS-TUD made good predictions of the cooling load. The errors are within the measurement error band.

4.4.3. System B

4.4.3.1. Temperatures

- **Supply Air Temperature. Test Defined Parameter**

The supply air temperature was defined as constant at 13.3ºC. The temperature leaving the cooling coil must be the supply air temperature minus the temperature increase caused by the supply fan. As Table 4.21 shows, the mean temperature measured was 12.2ºC.

Only the DOE-IOWA model had problems in estimating this temperature. None of the other programs had larger errors than 1ºC.

Table 4.21 Leaving cooling coil temperature (ºC)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-12.22	-1.00	-1.12	-1.82	-2.31	
dtmax	14.80	0.26	0.80	0.08	0.00	
meandt	1.72	-0.21	0.36	-0.37	-0.86	
min	0.00	0.00	0.00	0.00	0.00	0.0
max	14.80	12.80	12.60	11.80	11.31	13.6
mean	13.89	11.96	12.53	11.80	11.31	12.2
abmeandt	3.92	0.26	0.53	0.38	0.86	
rsqmeandt	5.95	0.33	0.57	0.58	0.97	
stderr	6.00	0.34	0.59	0.60	1.04	
stderr/mean	0.43	0.03	0.05	0.05	0.09	
MEAN%	14%	-2%	3%	-3%	-7%	

Conclusions: The errors can be neglected.

DOE-IOWA model should review their leaving cooling coil temperature.

- **Return Air Temperature. Non Test Defined Parameter**

The return air temperature must be a corrected mean value between the different room temperatures. If the room temperatures are accurately predicted, this return air temperature must be also accurately predicted. Table 4.22 shows the exactness of these predictions.

Table 4.22. Return air temperature for the AHU-B (ºC)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-23.06	-2.11	-2.91	-1.11	-2.25	
dtmax	24.40	0.15	1.36	0.55	0.04	
meandt	0.79	-0.61	-0.02	-0.06	-0.57	
min	0.00	0.00	0.00	0.00	0.00	0.0
max	24.40	22.80	24.30	23.09	22.63	23.2
mean	23.53	22.13	22.72	22.68	22.17	22.7
abmeandt	4.95	0.62	0.56	0.30	0.57	
rsqmeandt	9.96	0.77	0.88	0.40	0.77	
stderr	9.96	0.81	0.88	0.40	0.81	
stderr/mean	0.42	0.04	0.04	0.02	0.04	
MEAN%	3%	-3%	0%	0%	-3%	

The largest mean error is given by the DOE-IOWA and DOE-CIEMAT, but they are smaller that 1ºC. All the models accurately predict this temperature.

Conclusion: The errors can be neglected for the other models.

- **Supply Airflow. Non Test Defined Parameter**

Table 4.23 shows the results for the supply airflow in System B

Table 4.23. Supply airflow for AHU-B (m^3/h)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-3156.23	-261.40	-393.69	-345.47	-268.97	
dtmax	3246.00	225.92	254.72	108.50	288.06	
meandt	381.96	-13.69	-49.45	-70.81	-21.36	
min	0.00	0.00	0.00	0.00	0.00	0
max	4212.00	3301.00	3207.00	3173.99	3154.64	3276
mean	3270.62	2874.96	2839.20	2817.84	2867.29	2888.7
abmeandt	959.28	71.57	91.85	82.79	69.20	
rsqmeandt	1369.92	101.38	146.36	111.35	96.18	
stderr	1379.57	101.54	147.87	115.37	96.61	
stderr/mean	0.42	0.04	0.05	0.04	0.03	
MEAN%	13%	0%	-2%	-2%	-1%	

The results are very good. Only the DOE-IOWA model presented relatively large errors caused by an input error on the supply air temperature.

The other models are very accurate in their prediction. The DOE-CIEMAT, TRNSYS-TUD and PROMETHEUS models predicted almost exactly the measured supply airflow.

As this is a Variable Air Volume system, it is interesting to graphically analyze the variations on the supplied airflow. Figure 4.3 shows those results.

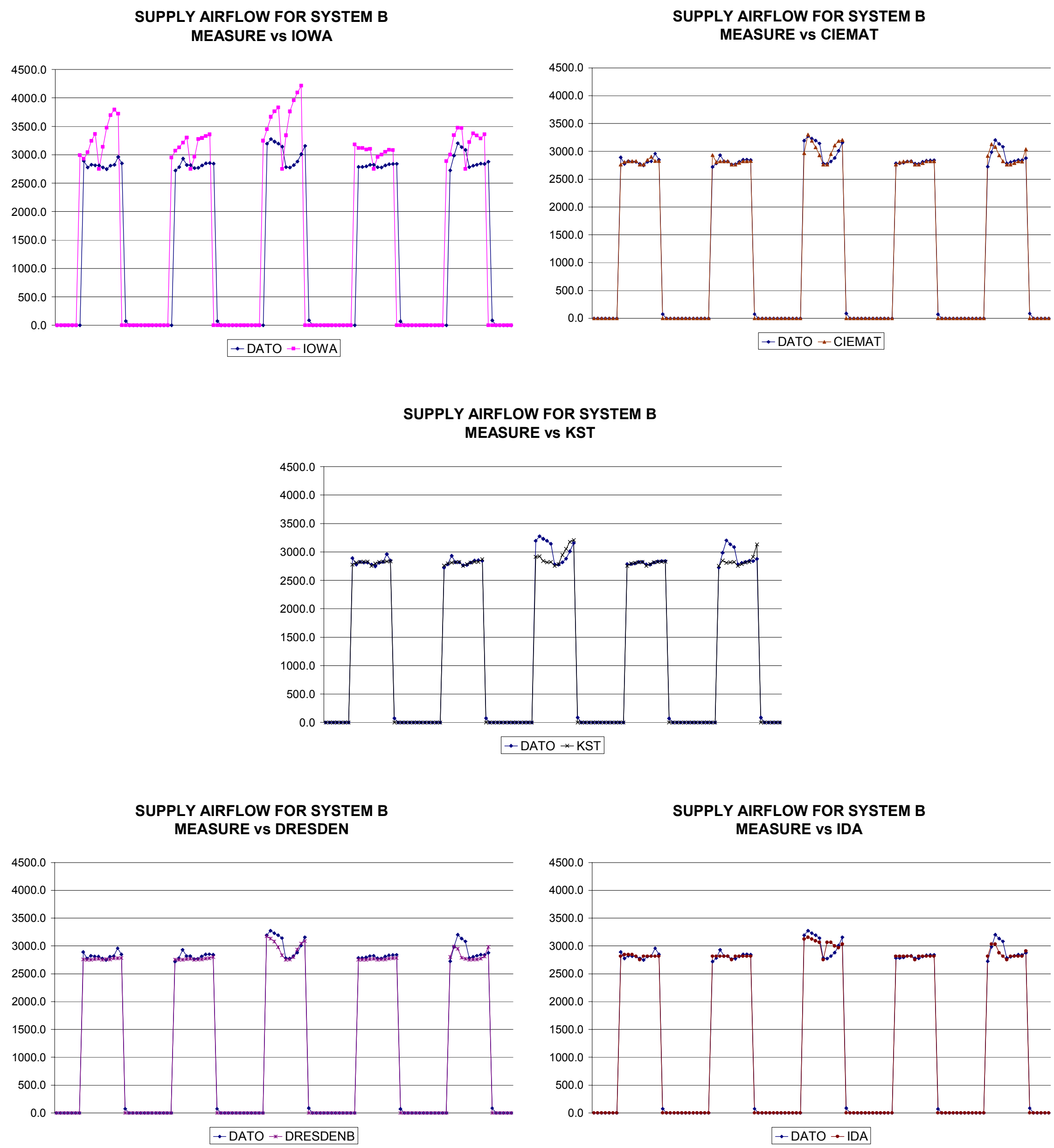

Figure 4.3 Supply air flow-rate for System B

Conclusion**:** DOE-IOWA model presents over predictions on the supply airflow. This is consistent with the previous error of the supply air temperature overestimation. The thermostat and system schedules must be checked. The system is working one hour forward.

PROMETHEUS IDA-ICE TRNSYS-TUD and DOE-CIEMAT present a very good behavior. All of them accurately predicted the airflow, with only small errors.

- **Cooling Coil. Non Test Defined Parameter**

Cooling load simulated and measured are given by Table 4.24

Table 4.24. Cooling load for AHU-B (Wh)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-10613.03	-1341.75	-1984.10	-1471.25	-1308.74	
dtmax	10108.00	1593.71	2312.81	1111.02	2525.41	
meandt	927.44	217.75	258.17	-377.76	628.16	
min	0.00	0.00	0.00	0.00	0.00	0.0
max	13448.00	12376.00	11966.00	10804.60	11632.15	11356.4
mean	10573.33	9863.64	9904.05	9268.13	10274.05	9645.9
abmeandt	2904.49	487.13	765.16	608.15	860.15	
rsqmeandt	4330.49	620.95	948.64	730.36	1025.53	
stderr	4348.51	627.85	955.01	747.91	1059.93	
stderr/mean	0.41	0.06	0.10	0.08	0.10	
MEAN%	10%	2%	3%	-4%	7%	

Results are very good for DOE-CIEMAT, PROMETHEUS and TRNSYS-TUD models.

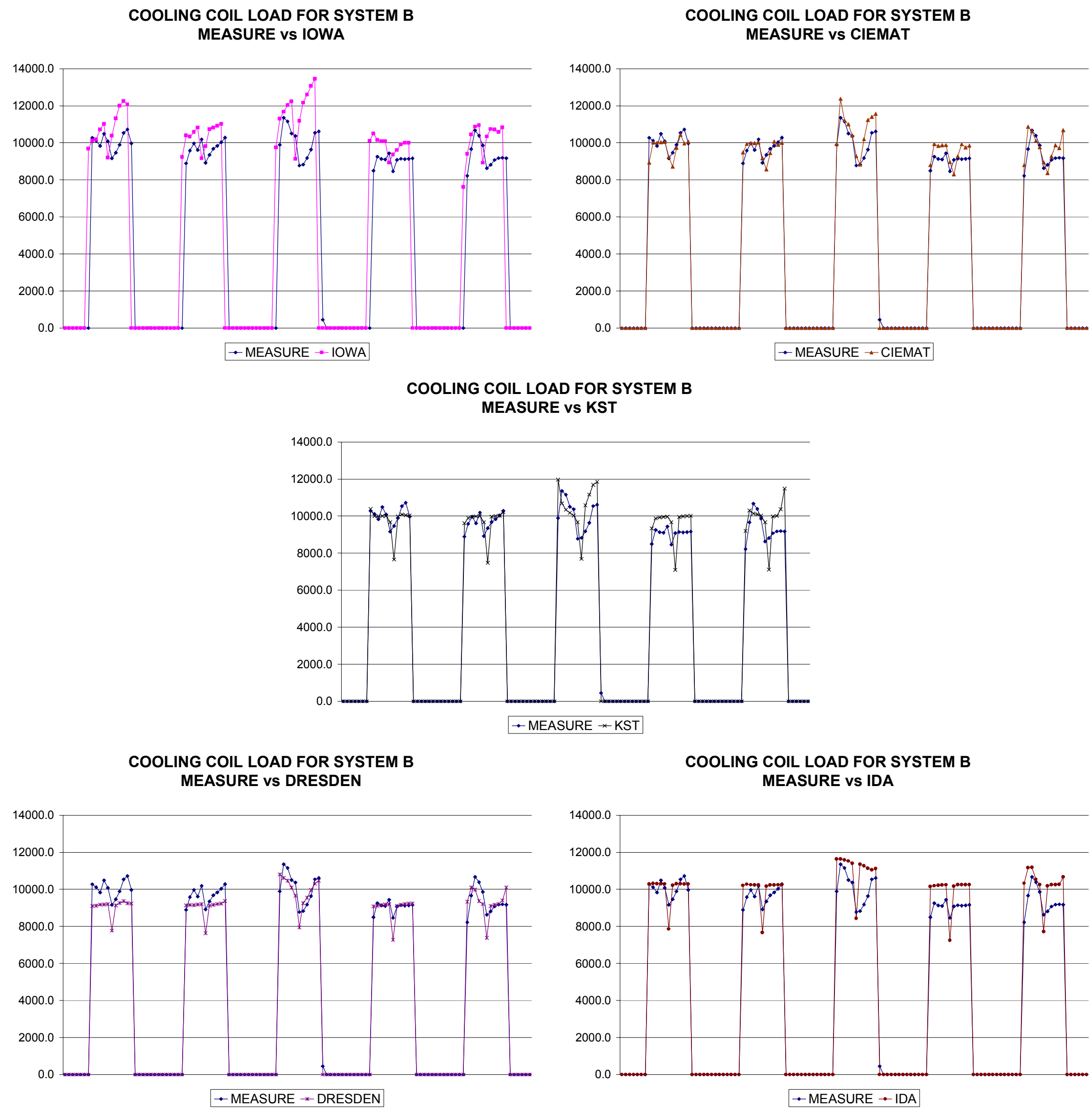

Figure 4.4 Cooling coil load for System B

Conclusions: DOE-IOWA model overestimated the cooling loads by a 10%. It is turned on and off one hour before it is specified for the test.

DOE-CIEMAT model accurately predicted the mean values and fast dynamics. It presented a small overestimation of the large values.

PROMETHEUS model predicted accurately the mean values, but it under-predicted the low values and over-predicted the large ones.

TRNSYS-TUD model presented a 4% underestimation error due to an under-prediction on the high values.

IDA-ICE model shows an error of 7% and its dynamic behavior seems to be very constant.

4.4.4. Global Reheat Energy and Cooling Energy Supplied Into the Room. Non Test Defined Parameters

The behavior of each room has been analyzed and it is presented in Appendix G. It is interesting to evaluate how big is the global error of the simulations.

4.4.4.1. General Conclusions Common to Every Room

An analysis of the behavior of each room is presented in Appendix G. Some general conclusions, common for every room, are made as follows:

- All the models had some problems in estimating the fast dynamics.
- All the models overestimated the effect of the midday setback. All the models showed less thermal inertia than the measurements.
- The Iowa model overpredicted the supply airflow due to an error on the input of the supply air temperature. It is one hour forward.
- All the other models accurately predicted the supply airflow.
- All the models accurately estimated the cooling loads.

4.4.4.2. Total Reheat Energy of System A

The reheat energy demanded for the entire System A is presented in Table 4.25 and Figure 4.5.

Table 4.25. Electrical Reheat Energy demanded by the System A (W)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-4607.55	-3808.55	-2741.55	-2815.60	-3007.87	
dtmax	1595.00	1904.00	1494.74	1168.14	1209.74	
meandt	-810.45	8.08	134.08	-230.08	-55.63	
min	0.00	0.00	0.00	0.00	-3.49	0.0
max	1595.00	2091.00	1994.00	1861.95	1669.68	4677.6
mean	146.89	965.42	1091.42	727.26	901.71	957.3
abmeandt	895.57	625.08	550.69	464.98	508.12	
rsqmeandt	1221.66	935.52	749.76	682.66	730.81	
stderr	1269.60	935.53	751.94	689.68	731.20	
mean%	-85%	1%	14%	-24%	-6%	
stderr/mean	1.33	0.98	0.79	0.72	0.76	

The DOE-IOWA model presented an underestimation of 85%. This means an error of $8W/m^2$ for the entire building.

DOE-CIEMAT model accurately estimated the mean value; it shows an error of 8W for the entire building ($0.08W/m^2$).

PROMETHEUS results are also very accurate; the error is around $1W/m^2$.

TRNSYS-TUD model accurately estimated the fast dynamics. It only had some problems with the high values.

IDA-ICE results are also very accurate; the error is only $0.5W/m^2$.

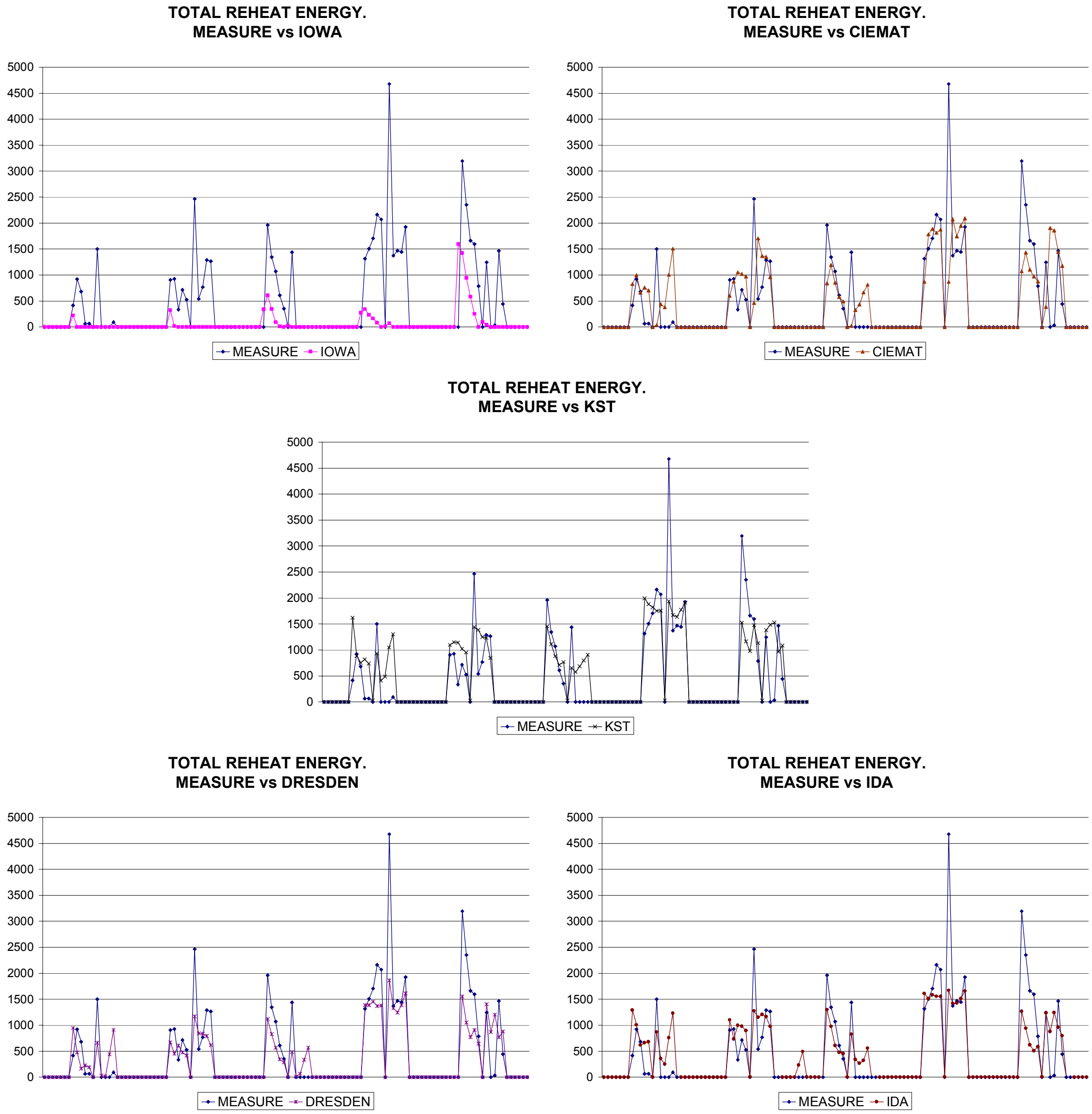

Figure 4.5 Total reheat energy for System A

4.4.4.3. Total Cooling Energy Supplied to A Type Room

The cooling energy supplied through System A is presented in Table 4.26 and Figure 4.6.

Table 4.26. Cooling energy supplied to the rooms by the System A (J)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-40469.41	-10923.12	-11960.16	-6928.87	-6061.31	
dtmax	30253.20	15251.52	9773.38	12728.22	16184.92	
meandt	1111.35	-1288.96	-4110.28	-203.53	1640.19	
min	0.00	0.00	0.00	0.00	0.00	0.0
max	48218.64	39071.04	35652.96	38464.41	40309.39	42191.3
mean	33988.93	31588.62	28767.31	32674.06	34517.77	32877.6
abmeandt	9393.51	4045.00	5085.36	3029.33	3497.05	
rsqmeandt	14324.62	5437.01	6107.47	3985.58	4757.86	
stderr	14332.45	5464.72	6353.97	3986.52	4808.99	
mean%	3%	-4%	-13%	-1%	5%	
stderr/mean	0.44	0.17	0.19	0.12	0.15	

The DOE-IOWA model was very accurate but it is one hour in advance from the measurements.

DOE-CIEMAT model was also very accurate (4% error) but slightly underestimated the large values.

PROMETHEUS model slightly underestimated the cooling loads.

TRNSYS-TUD model is very accurate; its error is only 1% with a very good estimation of the fast dynamics.

IDA-ICE model accurately estimated the mean value, but slightly overestimated the large values.

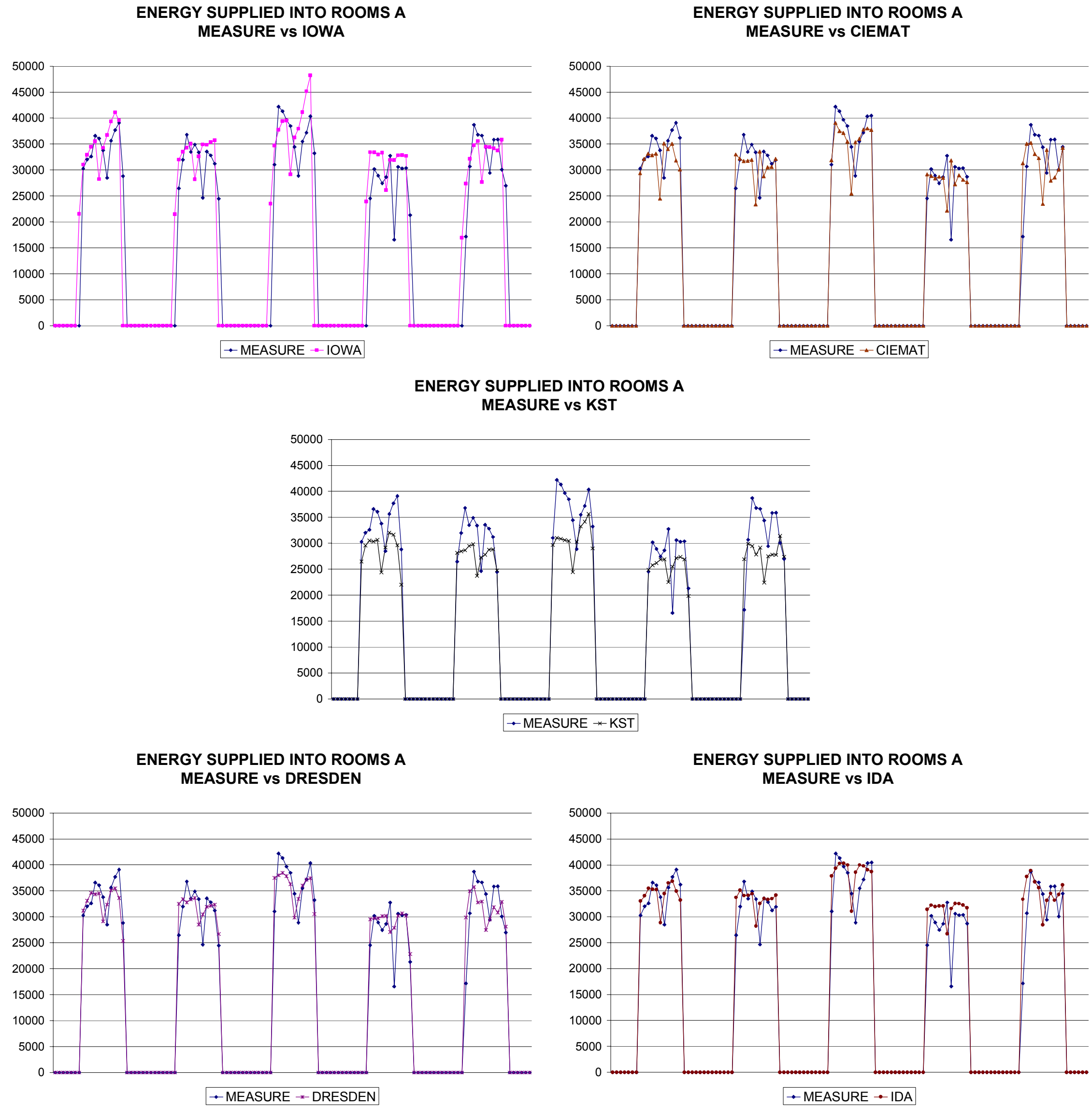

Figure 4.6 Total cooling energy supplied into System A

4.4.4.4. Total Reheat Energy of System B

The reheat energy demanded by all the System B is presented in Table 4.27 and Figure 4.7.

Table 4.27. Electrical Reheat Energy demanded by the System B (W)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-4195.84	-3406.84	-2318.84	-2463.69	-2601.32	
dtmax	1563.00	1870.00	1483.00	801.60	961.31	
meandt	-665.69	142.97	300.33	-117.42	95.51	
min	0.00	0.00	0.00	0.00	-3.53	0.0
max	1563.00	2075.00	1977.00	1800.15	1662.52	4263.8
mean	131.71	940.36	1097.73	679.98	892.90	797.4
abmeandt	746.27	638.62	544.64	392.06	438.83	
rsqmeandt	1036.04	879.72	695.51	573.93	587.43	
stderr	1074.22	881.83	707.20	576.11	588.84	
mean%	-83%	18%	38%	-15%	12%	
stderr/mean	1.35	1.11	0.89	0.72	0.74	

The DOE-IOWA model presented an error of 83%; this is only 6.65W/m^2 for the entire building.

DOE-CIEMAT model accurately estimated the mean value (error of 143W) for the entire building, 1.43W/m^2.

PROMETHEUS model was very accurate (3W/m^2 error for the entire building).

TRNSYS-TUD model accurately estimated the fast dynamics. It only had some problems with the high values (1.17W/m^2 error)

IDA-ICE model made almost perfect estimations. The error is smaller that 1W/m^2.

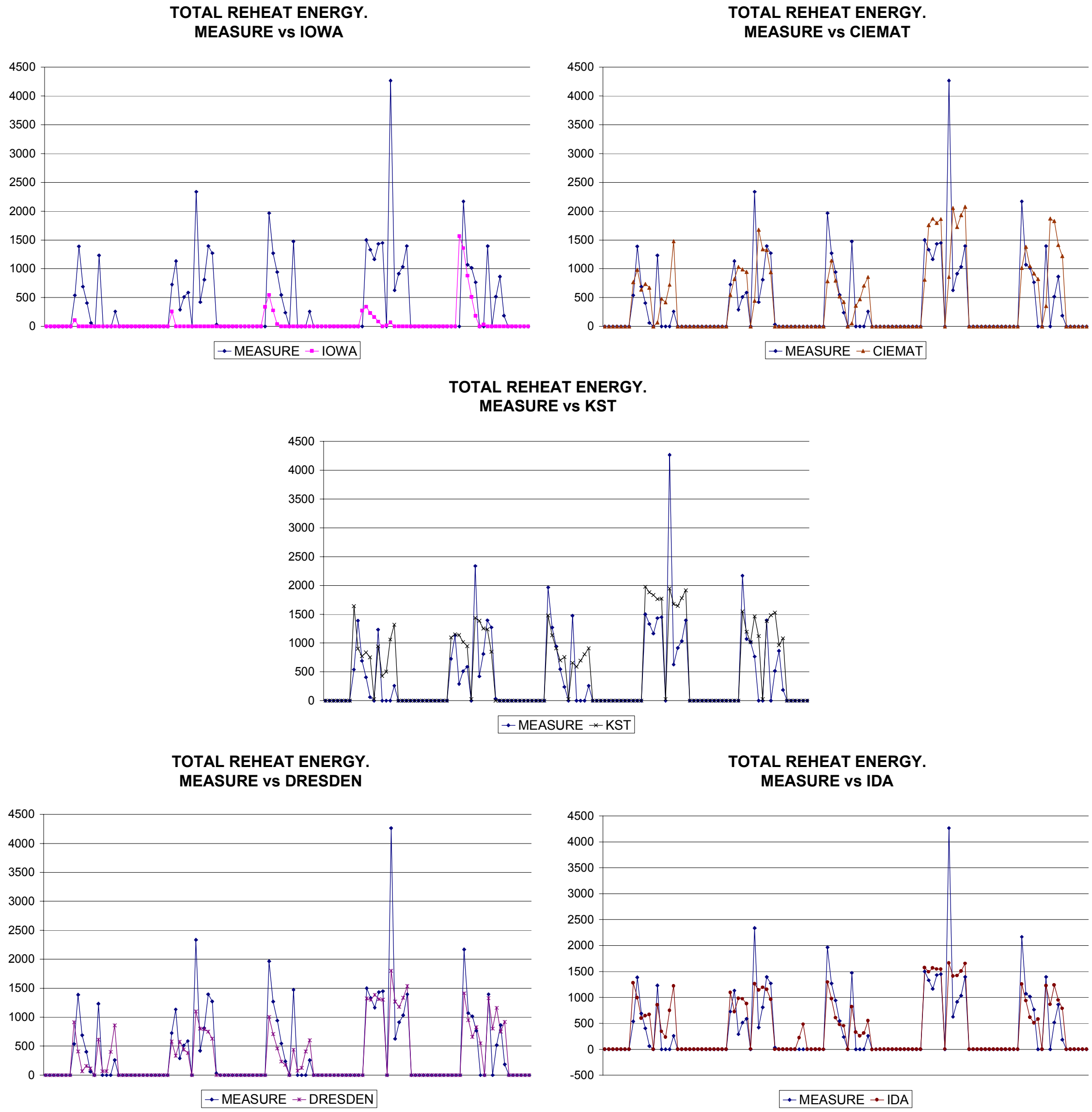

Figure 4.7 Total reheat energy for System B

4.4.4.5. Total Cooling Energy Supplied to B Type Room

The cooling energy supplied through System B is presented in Table 4.28 and Figure 4.8.

Table 4.28. Cooling energy supplied to the rooms by the System B (J)

	IOWA	CIEMAT	KST	DRESDEN	IDA	REAL
dtmin	-38387.60	-10830.20	-11904.44	-6915.77	-6004.68	
dtmax	29613.96	14511.00	8190.84	10670.28	14228.91	
meandt	795.81	-224.42	-1601.06	122.92	988.58	
min	0.00	0.00	0.00	0.00	0.00	0.0
max	44203.08	39361.08	35811.60	37899.56	40313.71	39277.1
mean	15642.98	14622.75	13246.11	14970.09	15835.75	14847.2
abmeandt	4329.37	1643.84	2035.41	1244.25	1549.14	
rsqmeandt	9572.28	3332.43	3642.55	2507.24	3111.17	
stderr	9539.14	3324.87	3271.81	2504.22	2949.93	
mean%	5%	-2%	-11%	1%	7%	
stderr/mean	0.64	0.22	0.22	0.17	0.20	

The DOE-IOWA model is very accurate but presented a small over-prediction of the high values, and also had an error in the schedules.

DOE-CIEMAT model was very accurate (2% error). It accurately estimated the mean value and the fast dynamics.

PROMETHEUS model shows a small under-prediction of the high values.

TRNSYS-TUD model is very accurate. It showed an error of 1%, accurately estimating the fast dynamics.

IDA-ICE model is very accurate but showed a small over-prediction of the cooling loads.

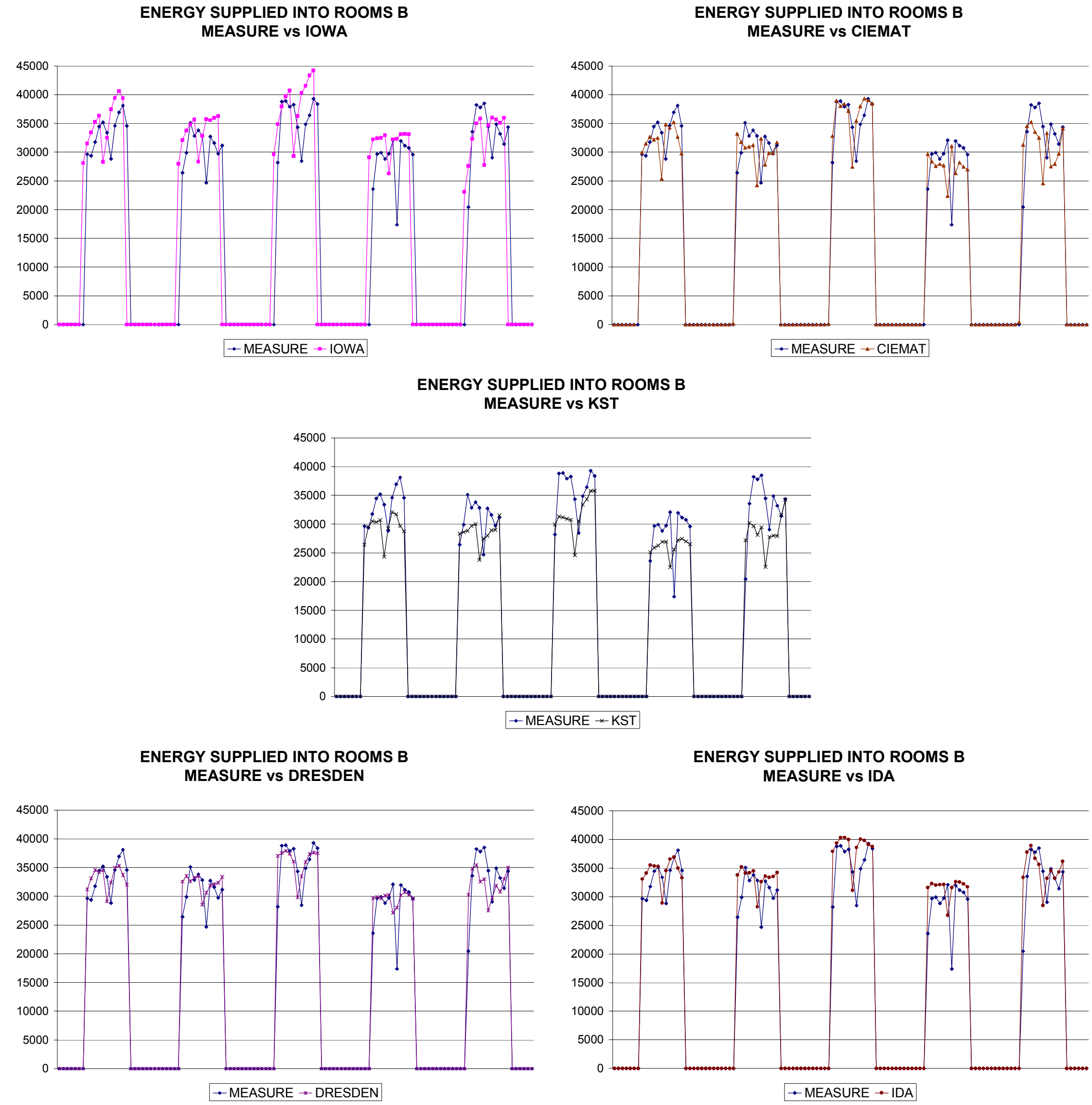

Figure 4.8 Total cooling energy for System B

4.5. Discussion of the Results and Conclusions

All the models accurately predicted the temperatures on the AHU. Only the DOE-IOWA model showed an error on the input, similar to previous exercise. It should overestimate the supply airflow and underestimate the reheat needs and the cooling loads on the AHU.

As a Variable Air Volume case, the supply airflow of the AHU is carefully considered. All the models presented errors smaller that 3%, which is negligible. Only DOE-IOWA showed larger errors, which is consistent with the error previously described in the supply air temperature.

An analysis of the cooling loads of the AHU showed how only DOE-IOWA and IDA-ICE models had non-negligible disagreements. IDA-ICE's dynamic behavior seems to be very constant. The other models, especially TRNSYS-TUD, made good predictions of the cooling load. The errors are within the measuring uncertainty band.

The rooms analysis resulted in the following conclusions:

- All the models had some problems in estimating the fast dynamics.
- All the models overestimated the effect of the midday setback. All the models might be showing less thermal inertia than measurements.
- The Iowa model over-predicted the supply airflow due to an error on the input of the supply air temperature. It is one hour forward.
- All the other models accurately predicted the supply airflow.
- All the models accurately estimated the cooling loads.

In the cooling load calculation, IDA-ICE shows an over-prediction cooling demand for the Interior and South Rooms. For the Interior Rooms, this behaviour is likely due to a problem of the specification of the neighbouring room connection.

For the South Room, overestimation may be due to overestimating the solar heat gains. The examination of the weather data shows that only the third day is a clear day. The overestimated cooling energy demand occurs exactly on this day. A possible reason could be an error in the solar data or location (calculation of the solar position). Some problems in the zone model could be excluded because the East and West Rooms show accurate results.

All the models made very good predictions on the global building needs. The errors are lower than 5% in almost every case. Only PROMETHEUS showed some disagreements and its underestimation is around 10%. In all the other cases, the mean value and fast dynamics were accurately predicted.

5. Modeler Reports

This section of the report presents the Modeler Reports from the individual and organizations that participated in this IEA Task 22 empirical validation exercise. These reports address any information that the modelers with to share which explain or clarify their model validation results. After the “blind” portion of the validation exercise, modelers were allowed to revise their models and results for “legitimate” modeling reasons. These legitimate modeling reasons include incorrect inputs, misinterpretation of output requirements, faulty or deficient algorithms found because of disagreement with the measured data, or other “legitimate” reasons that were not arbitrary to simply obtain closer agreement to the measured data.

A Modeler Report is not available for PROMETHEUS because the participating organization, Klima System Technik, had to withdraw from the work prior to completing all rounds of the validation exercises, and preparation of the final report.

5.1. DOE-IOWA

The main objective of this report is to describe the modeling strategy used for the empirical validation exercises developed at the Iowa Energy Resource Station (ERS) by Iowa State University, Ames, Iowa. The program used was DOE-2.1E which was run on a PC.

The LOADS model was developed for the matched set of test rooms at the ERS. Building construction documents were used to obtain details about the wall, roof and slab construction layers as well as the windows. In the IOWA model, the partition walls separating the test rooms from the remainder of the ERS were modeled as adiabatic.

No particular problems were encountered using DOE-2 to model the ERS. Many of the default values provided by the program were used when specific values were not available.

5.2. DOE-CIEMAT

The main objective of this report is to describe the modeling strategy used for the empirical validation exercises developed a the Iowa Energy Resource Station (ERS) by CIEMAT, Madrid, Spain.

The program used is the DOE-2.1E version 088. The DOE-2 program is a set of 4 different subprograms, LOADS, SYSTEMS, PLANT and ECONOMICS. The LOADS program simulator calculates the hourly heating and cooling loads considering:

1. A constant space temperature for the room.
2. A constant temperature for the unconditioned spaces.

The SYSTEM program adjusts the LOADS results by considering the temperature for each space every hour and the actual temperature for each room.

Some model assumptions besides the ERS specifications have been made and are explained in this report.

Climate

The DOE-2 program has a weather processor to convert the TMY weather files to a bin file, which can be interpreted by the program.

The weather data is provided in TMY format, which compiles the data, by solar standard time. The DOE-2 program does not consider the difference between the solar time and the local standard time. The input data is provided at solar time, which is different than local standard time. This causes some small differences between DOE-2 and measurements.

The ground temperature is calculated by the weather processor, considering a soil difussivity of 0.02 m^2/day.

Construction. Walls, ceiling, floor.

The material and layers compositions have been defined from the description section. All the input data required by DOE-2 which have not been defined have been set to default values. Such is the case of infrared emissivities, absortance, etc.

Adjacent Rooms

The adjacent rooms' temperatures have been registered at the test period. Those measured values have been considered for the simulation by defining a variable thermostat strategy and an ideal HVAC equipment. This method has been applied to the office, reception, vestibule, mechanical equipment room, etc.

This way only the test rooms are completely simulated and the neighbor conditions are completely controlled.

Local Reheat

The DOE-2 program defines the reheat coils capacity by defining the temperature of the air rising through it. The calculation of this temperature's increase has been hand calculated by using:

$$\Delta t = \frac{Q(W)}{C\left(m^3/h\right)\rho\left(kg/m^3\right)C_p\left(kJ/kg{}^\circ C\right)}$$

Conclusions

No important difficulties have been encountered to develop the models. The DOE-2 program simplifies very much the data input for this kind of simulation.

The results have been close enough to the measurements to consider them as good estimations, as the general report concludes.

5.3. PROMETHEUS

As previously stated, a Modelers Report is not available for PROMETHEUS.

5.4. TRNSYS-TUD

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Background

All simulation runs were conducted with TRNSYS-TUD, a modified 14.2-Version of TRNSYS.

The building model

The basis for the building model are a set of architecture drawings of the real building in combination with a manual containing information about the properties of wall layers and windows. It is a very labour-intensive procedure to create the model if each detail of the construction should be taken into consideration. This might also be a source of error.

But it must be done just once for all the test cases.

As expected and as it is typically for simulation problems the dynamic of the building model differs a little from the behaviour of the real building because theoretical properties of wall layers were given in the specifications. That is why temperature peaks as well as floating profiles are not fully identical to measuring data. The building model was not really validated.

The chiller model

The calculation of the cooling energy demand requires the calculation of both sensible and latent loads. Therefore, a simplified model of the chiller according to the following equation was implemented into the simulation program.

$$\dot{Q}_{total} = \dot{Q}_{sens} + \dot{Q}_{lat}$$
$$= \dot{m}_{air}\left(c_{p,L}\Delta\vartheta_L + \Delta_V H \Delta x\right)$$

$\dot{m}_{air}$ air mass flow
$c_{p,L}$ heat capacity of air
$\Delta\vartheta_L$ temperature decrease
$\Delta_V H$ enthalpy of vaporization
Δx humidity decrease

In order to be able to calculate the latent cooling loads the humidity of zone air has to be known.

Validation

The efficiency of the chiller is a parameter of the chiller model. The model can be validated by changing the value of this parameter. To adapt the parameter the measurement data from the first test (September 1997) have been used because all information about flow rates, entering and leaving temperatures from both water side and airside were given. This test was not a *blind test* and in this way the cooling energy calculated by our model was compared to the measurements (Fig.1; Fig.2).

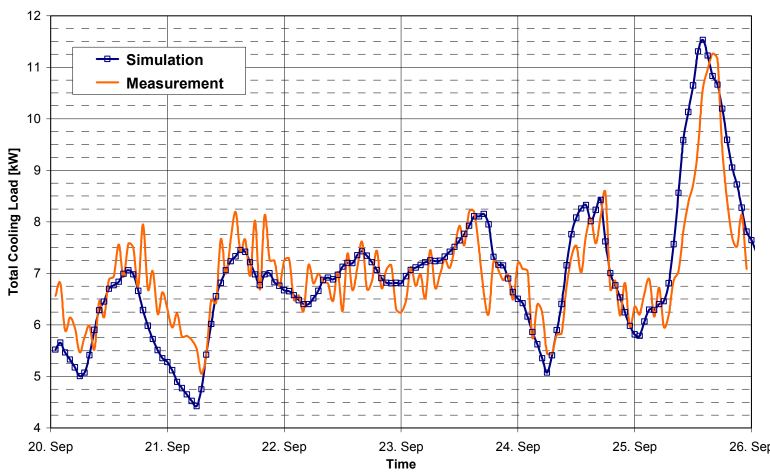

Fig.1: Total Cooling Load System A, Test No. 1 (September 1997)

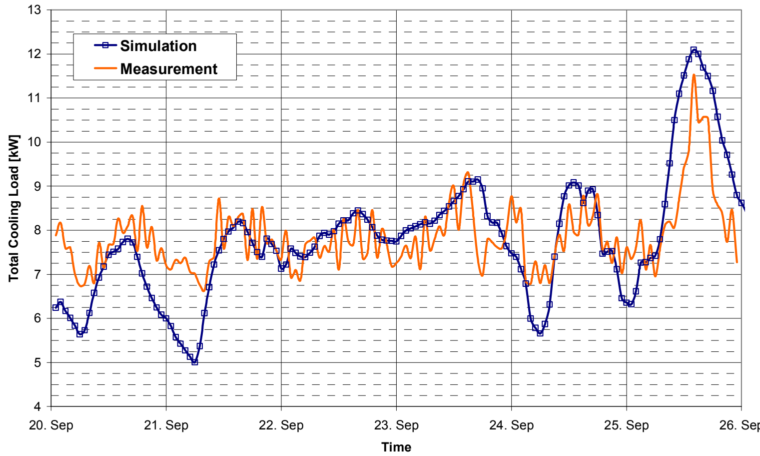

Fig. 2: Total Cooling Load System B, Test No. 1 (September 1997)

Test cases

At the ERS several tests were conducted within the *Task22* activities. For each test case, a specification that contains information about control, heat sources and almost everything which is important to know to start the simulation, was available. All the tests were marked by:

- Set points for the room temperatures (fixed or time dependent),
- Set point for supply air temperature.

The first simulation run always was a real *blind test*. The re-runs were conducted with knowing the measurement data. The simulation results did not match the measurements at all points, because of

- Simulation model does not represent the real facts exactly,
- Measurements are flawed partly or measuring devices were not validated (is done now)
- Information are not available or inaccurate (heat output of baseboard heaters)

After the measured data are known, it is very easy to change the parameters to fit the simulation results to the measurement data. It is a kind of art and it depends on the skill of the modeller to create a appropriate good model. But simulation should be a tool for *blind tests*.

From my point of view, there is one thing of interest: Which assumptions to facilitate the models does the modeller do, either he wants to do or he has to do, and how does it influence the simulation results? Which of these assumptions are useful for future simulation runs?

For instance: The return air is not taken directly from the room but from the plenum. The room temperature is fixed at a set point. But who can really simulate the plenum temperature profile? For the simulation a constant plenum heat gain was defined (Fig.3; Fig.4). That is why the difference between simulated plenum and room air temperature is not changing and differs from the measurement. As shown in figures 5 and 6, this fact does not matter if the total cooling energy is considered. As a reference value for the cooling energy consumption of the building, the waterside measurements are more accurate than the airside data.

Some more simplification of the model concern the duct system, the air handling units, the cooling coil, the economizer and the fan, which are represented by just a few equations. The waterside of the chiller was not considered anyway just for the reason of comparing the simulation results. Duct heat gains as well as gains due to the fans were assumed to be constant all over the time.

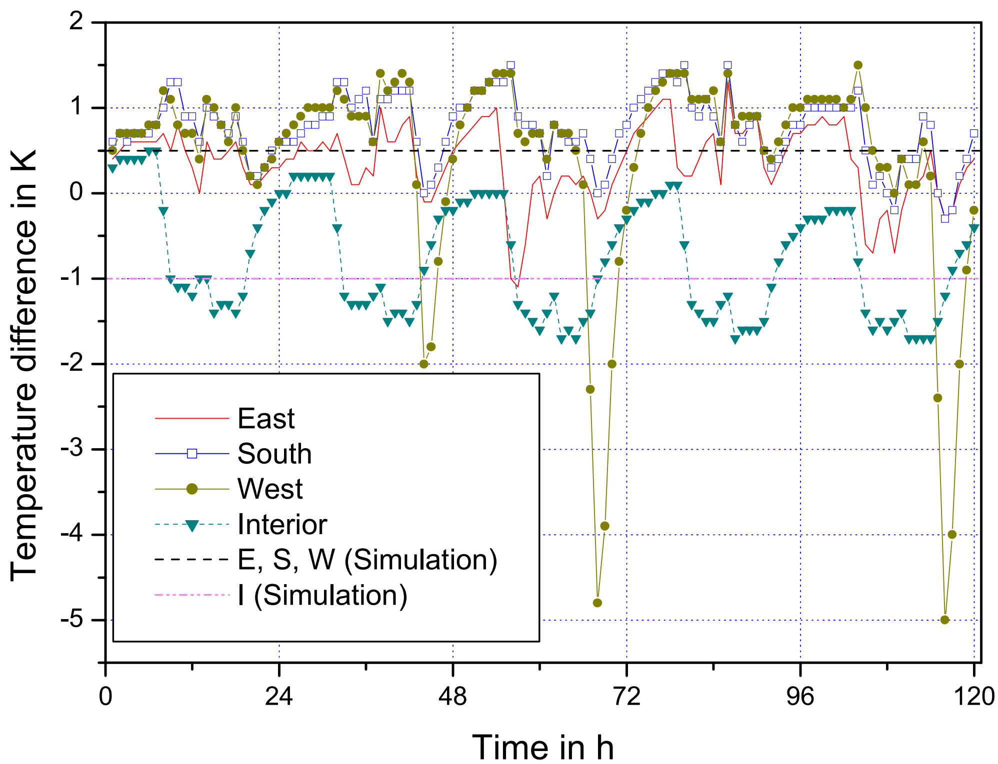

Fig.3: Temperature differences Plenum - Room; June VAV Test, System A

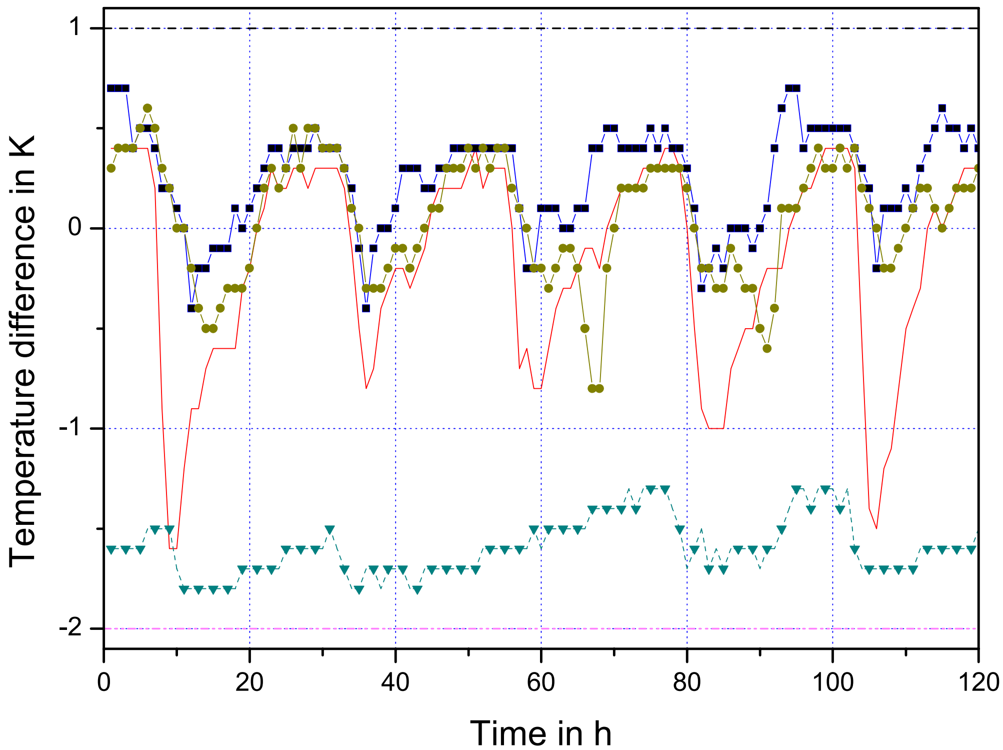

Fig.4: Temperature differences Plenum - Room; June CAV Test, System B

On account of the wide range of measuring data available from the ERS, it is possible to analyse a lot of things: temperatures, air mass flows, energy rates. Thus, it is possible to check simulation data versus measurements. Doing it this way, simulation software and/or the model can be improved (because a bug was detected during simulation runs).

Conclusions

These facts were confirmed to me:

- It is difficult to get good measurement results equivalent to simulation results. Simulation is more accurate then experimental data.
- A lot of assumptions have to be made because it is not possible to model each detail of a system.
- In spite of simplifications, the model is good enough to calculate cooling energy demand.

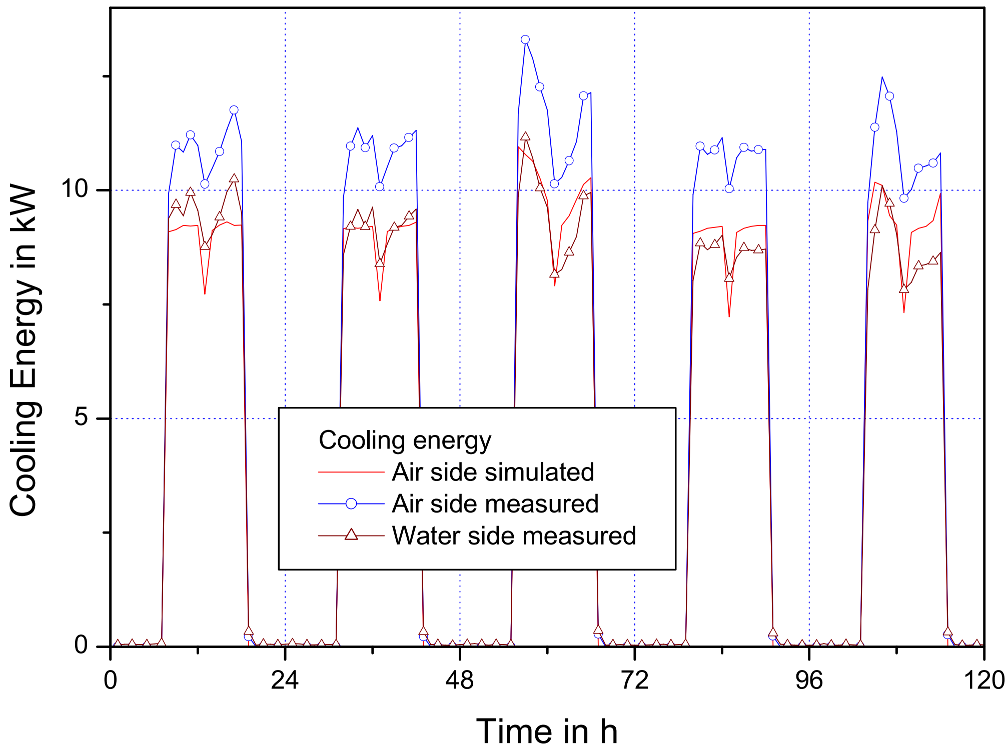

Fig.5: Cooling Energy in System A; June VAV Test

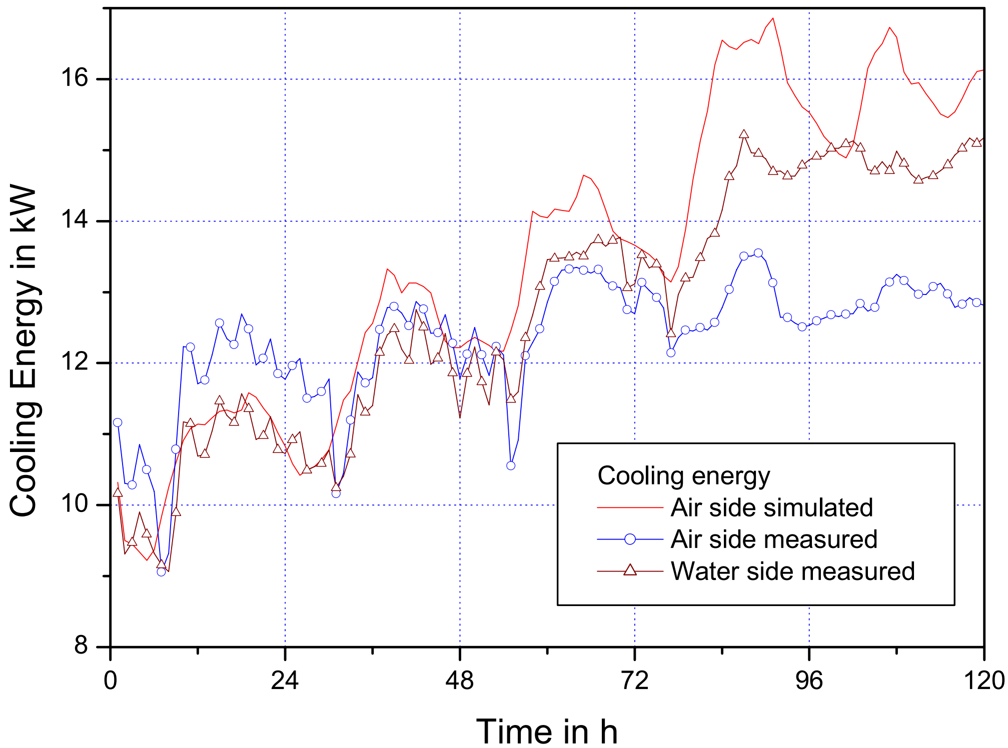

Fig.6: Cooling Energy in System B; June CAV Test

5.5. IDA-ICE

IDA-ICE Version 2.11.06

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Introduction

This report describes the modelling strategy used for the empirical validation of the Iowa Energy Resource Station (ERS).

Model assumptions made in addition to the ERS specification, and modelling difficulties that occurred are noted. Input and output files are saved on electronically.

The program version used is IDA-ICE, 2.11.06. IDA-ICE ICE is a simulation tool developed and distributed by EQUA Simulation Technology Group (formerly BRIS DATA), Stockholm, Sweden. The program is based on modular components describing a building and its HVAC system. The models are written in equation-based NMF code. The IDA-ICE simulation environment contains a special application, called IDA-ICE ICE – Indoor Climate and Energy. All tests described in this document have been simulated with ICE.

Climate

IDA-ICE contains a translator routine to convert TMY files to program specific inputs.

It is used the Ankeny.tmy file. The time coordinates of a weather file are always used as clock time for the simulated location. This would effect that the comparison between measure data and IDA-ICE output show some differences. Another reason for some differences is due to the output file format.

Neither the climate processor nor the translator is able to calculate wet bulb temperatures. That is the reason why these results are missing in the report. To take into account the humidity fraction, the relative humidity is specified in the climate file.

For all calculations, the air density is set as a constant value. Assumed value for the ERS location is 1.164 kg/m3.

Construction

Walls, ceiling, floor.

The material parameter where adopted from the description section. For the surface specification, the following values had to be set:

a) Interior Wall

Long wave emissivity	0.82	- gypsum board
Short wave reflectance	0.50	- gypsum board

b) Ceiling

Long wave emissivity	0.82	- gypsum board
Short wave reflectance	0.75	- gypsum board

c) Floor

Long wave emissivity	0.90	- carpet
Short wave reflectance	0.50	- carpet

d) Exterior Wall

Long wave emissivity	0.90	- white surface
Short wave reflectance	0.30	- white surface

Heat transfer coefficient

Inside and outside heat transfer coefficient are calculated automatically for every time step. Depending on wind speed, surface temperature and solar incidence.

Window proprieties

Glazing

The window properties are specified as follows:

Double Pane for the test rooms

U-Value	= 3.1 W/m2K
Internal and external emissivity	= 0.9
Total shading Coefficient	= 0.85 (Double Pane Reference)
Short Wave shading Coefficient	= 0.73 (Double Pane Reference)

Shading

No external shading required.

Zone Model

The test rooms are modelled as a shoebox with six surfaces. Each surface has one neighbour room with its room temperature. The floor with its 1m-insulation layer is set adiabatic. The walls between the test rooms where set adiabatic. The ceiling adjoins the plenum. The neighbour rooms and the plenum zones are not modelled. Only the measured temperatures where used as boundary conditions.

This restriction has the effect that the heat flow from the test room to the adjacent room is to high. This behaviour is due to the program limits. IDA-ICE has an option for interior walls to set the surface temperature of the other side of the test room. The test specification only describes the room temperatures. For all test cases, it was assumed that the surface temperature of the adjacent room is equal to the room temperature. For interior rooms this assumption is correct. For exterior rooms, the surface temperature will be much higher during the daytime due to the distributed solar radiation. This is the reason why the reheat energy in the CAV cases for the exterior rooms is higher than the measurement. An alternative solution would be to define all interior walls as adiabatic ones. The reheat energy would be reduced about 190 W or 10% from the max. Reheat energy. See Diagram 1.

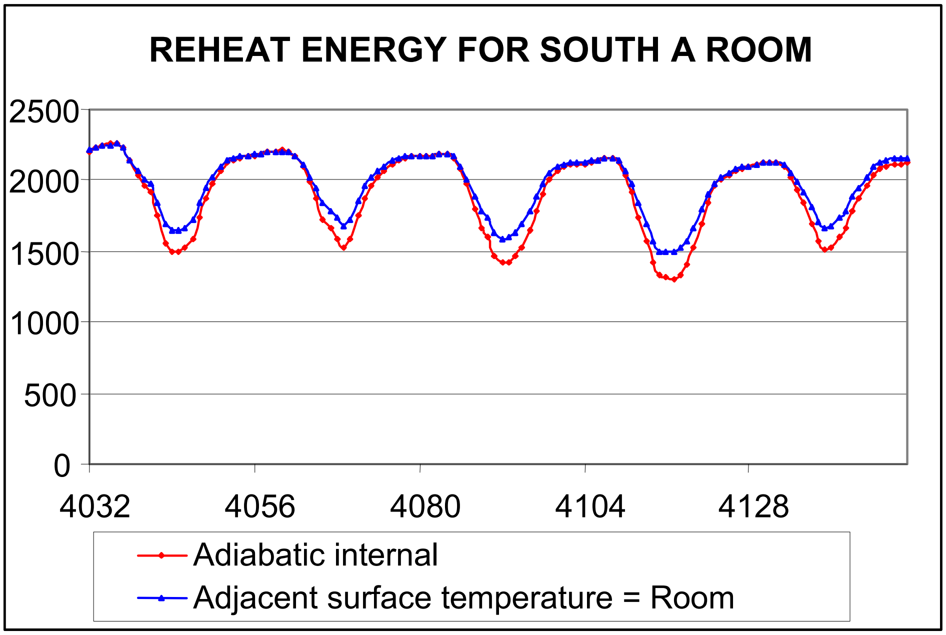

Diagram 1

HVAC-System

Air handling unit

Two system A and B for the three test cases were defined like the description propose. The two systems were simulated separately due to the following program restrictions:

- It is not possible to define two air-handling units without a lot of model effort on the advanced level. The flexibility to modify and rebuilt the model would be restricted too much.

- Two air handling units effects a non-proportional longer simulation run time.

Local Reheat

Each test room has a local reheat coil. In the IDA-ICE model, a water radiator was inserted in each room. To get only a convective heat exchange, the surface of the radiator was reduced to 0.0001 m^2 . The reheat coils are controlled individually for each room.

Zone Control

To specify the three different control strategies, some restrictions and modifications had to be done:

Volume flow

In all NMF models which are available in the air handling unit operates only with mass flow. The given values from the test specifications where translated in kg/m3. The results from the output file had to be reconverted into volume flow again. On that reason the output values from the volume flow doesn't correspond exactly to the other results.

In the variable case, the mean value for the supply airflow varies about 8m3/h. That difference corresponds to 0.0025 kg/s. The IDA-ICE report only give results with 2 decimal places. The difference between the results occurs due to these rounded values.

Temperature rising by the fan

The fan model in IDA-ICE allows only a constant temperature rising during ventilation time. For the variable cases, a mean value from the measured temperature differences was inserted.

Test case	Supply Fan dT	Return Fan dT
CAV	1.5	1.0
VAV	1.2	1.0
VVAV	1.8	1.0

Constant outside air fraction

To run the CAV case, the "MIXBXCTR" model had to be modified. It was the economizer function that had to be switched off during the calculation. This modification was made in the NMF source code. It was built a new NMF component, called "MIXBOX".

Output

IDA-ICE produce a number of predefined output files. For all test cases, a specific output file was defined to get all needed values.

Some results had to be converted to the specified units. Due to this recalculation some small differences in the results were occurred.

Deviation of output data's

IDA-ICE uses a variable time step integration scheme and in conjunction with weather input files with fixed hourly time steps and printed ICE reports (also with fixed hourly time steps but for the preceding hour) seemingly strange results can occur. E.g. The direct normal solar radiation.

The figure 1 below represents a single problem variable; say the outdoor air temperature in various possible representations. (The location of time steps and shape of solution curve is not quite realistic but that is of minor consequence here.)

All PRN files (input as well as output) are interpreted by IDA-ICE as containing *instantaneous* values of a variable, i.e. no integration over a certain time period is done or assumed. When the PRN-file happens to be generated from measured weather data, which usually represents mean values over some period, the given value should be the average *around* the current time point.

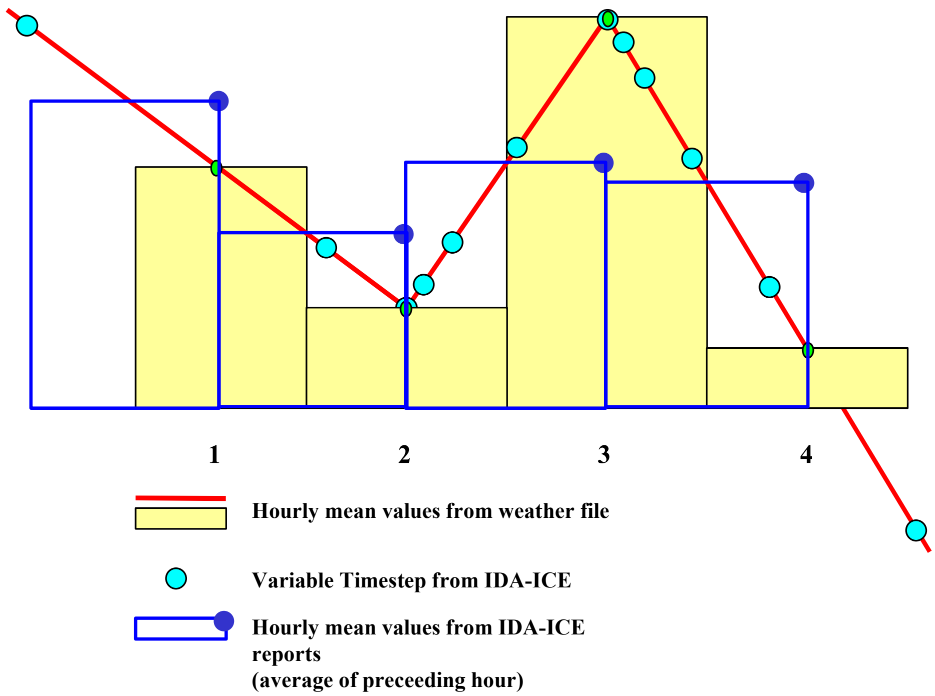

Figure 1

However, in the results post processing in IDA-ICE, the values are presented as the average over the *preceding* hour. In a case such as in the example above, this can lead to quite different results. The existence of this different convention in the same environment may be unfortunate but it is an artifact of other considerations, which are not fully presented here (c.f. average of a month, a week or a day).

Economizer model

To cool the outside air in the variable case, an economizer is inserted in the AHU. In section (iii), a comparison between the economizer functions of the different programs where made. To analyse the differences between the models, an outside air estimation was made (Page 55, section (iii)). The results from IDA-ICE overestimate the other results. The reason for that is the inserted return air temperature which is not considering the return fan dT. According to this, the dT is inserted into the formula:

Approximation of the outside air temperature:

$$t_{outside}^{approx} = \frac{m_{Supply} t_{incoil}^{approx} - (m_{Supply} - m_{outside})(t_{return}^{approx} + dt_{fan})}{m_{outside}}$$

If the fan dT is inserted, the IDE results correspond to the measure.

	without dT	with fan dT	Measure
Min	1.54	**-1.52**	**-0.6**
Max	15.60	**15.60**	**15.6**
Mean	6.75	**5.31**	**5.7**

Conclusion

IDA-ICE has a detailed zone model witch needs many input parameters.

This is due to the detailed interior solar distribution calculation. All the additional values are adapted either from the "ASHRAE Handbook" or from the "Recknagel, Sprenger, Schramek; Taschenbuch für Heizung- + Klimatechnik".

The system level from IDA-ICE works partly with simplified models e.g. the fan model. It has the advantage that the user quickly can describe a simple HVAC system. In spite of the simplifications, the results do not deviate from the results of programs with more detailed HVAC models.

IDA-ICE proved to be an adequate simulation tool for these test cases.

6. General Conclusions and Recommendations for Future Works

The overall results from this work show that the computer models were successful in calculating the measured values of several parameters of interest in building energy simulation. The calculated values generally fall within the range of experimental uncertainty of the measured data. As to be expected, some differences occur as a result of "real system" behavior as compared to "ideal system" behavior. System dynamics and controls play a major role in the manner in which the building operates. A case in point is the discharge air temperature from the air-handling units. Even though the temperature was specified to be constant, measured values show the temperature deviates about the set point as a result of system dynamics such as the chiller cycling on and off.

The cases studied in this work focused on relatively simple HVAC system configurations and control strategies. Future work should expand on what has been done here to include more realistic HVAC system operations that include energy conservation measures. In addition, it would be useful to validate other models often incorporated in building simulation. These include daylighting and ventilation models.

Future Work: Recommended Additional Cases

The participating experts identified the following new empirical validation test cases as high priority should further empirical validation exercises be undertaken as part of Task 22:

- Daylighting – HVAC Interaction
- Economizer Control
- Heat Recovery

Appendix A Specifications for Energy Modeling of the Energy Resource Station

This appendix provides the basic information used by IEA modelers to create building load input files for energy simulation of the Iowa Energy Center's Energy Resource Station (ERS). This appendix also includes a description of the computer model output used to compare models to each other and to compare model results to measured data.

Information in this appendix comes from the architectural drawings and construction documents available from the ERS Manager. The ERS drawings are provided on the CD-ROM that accompanies this report. All drawing files are in the folder named *Drawings*. Information presented in this appendix is organized in a manner similar to a DOE2 loads input file. Only the building envelope is described in this appendix. Details of the system input are provided in Appendix B. Appendix B also describes the test conditions that were used in the validation study.

Part 1: INPUT FOR LOAD CALCULATION

1. RUN-PERIOD

This is used to specify the initial and final dates of the desired simulation period. These dates are specified for each set of tests conducted. Testing dates are provided in Appendix B.

2. WEATHER-DATA

Weather data during each test period were collected and converted into TMY format. The name of the weather file is Ankeny.IA1. This TMY file is available on the CD-ROM in the folder named *Weather*.

3. BUILDING-LOCATION

This specifies the location of the building and information about time. The ERS is located in Ankeny, Iowa USA

- Latitude: 41.71 degree
- Longitude: 93.61 degree
- Altitude: 938.0 feet
- Time zone: 6, central time zone in US
- Daylight saving time: Yes or No depending on the test date.

4. BUILDING-SHADE

The site location for ERS provides sufficient separation between adjacent building and obstacles (such as trees) that except for sun angles within five degrees of sunset, there is no external shading from nearby objects. The surrounding ground cover is nearly all grass with a limited amount of concrete walkways approaching the doors.

5. FLOOR-PLAN

Figure A.1 is a simplified floor plan, and Figure A.2 is a 3-D view of the building. This floor plan is used to identify each space for the building model. Details of the floor plan are available in the architectural drawings on the CD-ROM.

6. CONSTRUCTION LAYER DESCRIPTION

This specifies the material layers of each construction element in the model. These include the cross section of an exterior wall, interior wall, ceiling, door, underground floor and roof. Details of the building construction are shown in the building drawings.

6.1 LAYER TYPE IDENTIFICATION

For the exterior walls of the test rooms, two different construction layers are defined because of the different wall constructions that are present. These are referred to as the “bottom” and “top” wall layers. The bottom layer is from the floor to 8’-6” above the floor, while the top layer begins at 8’- 6” and continues to the roof. For example, the top wall is an exterior wall in the plenum of a room, and the bottom wall is an exterior wall in the conditioned space of a room.

Table A.1 classifies construction layers used in the building. LAY-R* under the layer type column is used for the horizontal roof of the building. LAY-W* is used for the vertical exterior wall of the room. LAY-P* is used for the vertical interior partition wall of the room. LAY-C1 is for the ceiling of the room, and LAY-F1 is for the floor of the room.

6.2 LAYER DESCRIPTION

Layers for exterior walls and roofs are described from inside to outside. Layer type is referred to the layer identification in Section 6.1. Table A.2 describes thickness and thermal properties of materials used for the construction layers. English units are used.

T: thickness, in inches
K: conductivity, in Btu/(hr-ft-^{o}F)
D: density, in lbm/ft^2
C_p: specific heat, in Btu/(lbm-^{o}F)
R: resistance, in (hr-ft^2-^{o}F/Btu).

An inside film resistance for the inside wall surface, is 0.68 (hr-ft^2-^{o}F/Btu). The outside film resistance value was unspecified so that a simulation program could calculate it based on actual wind speed data. Solar absorptances for the exterior walls and roofs are 0.6 and 0.29, respectively.

Table A.1. Identification of construction layers used in the ERS building

Layer type	Description
LAY-R1	Layers for the roof of all spaces except for the classroom.
LAY-R2	Layers for the roof of the classroom.
LAY-W1	Layers for the exterior wall in bottom of test rooms
LAY-W2	Layers for the exterior wall in top of test rooms
LAY-W3	Layers for the spandrel wall in bottom of computer room and office
LAY-W4	Layers for the exterior wall in top of computer room and office
LAY-W5	Layers for the exterior wall of the classroom
LAY-W6	Layers for the exterior wall in bottom of the other spaces
LAY-W7	Layers for the exterior wall in top of the other spaces
LAY-P1	Layers for the 6-inch interior partition wall of all spaces
LAY-P2	Layers for the 4-inch interior partition wall of all spaces
LAY-P3	Layers for the 1/8-inch interior glass partition wall of test rooms
LAY-P4	Layers for the door of all spaces
LAY-C1	Layers for the ceiling of all spaces
LAY-F1	Layers for the ground floor of all spaces

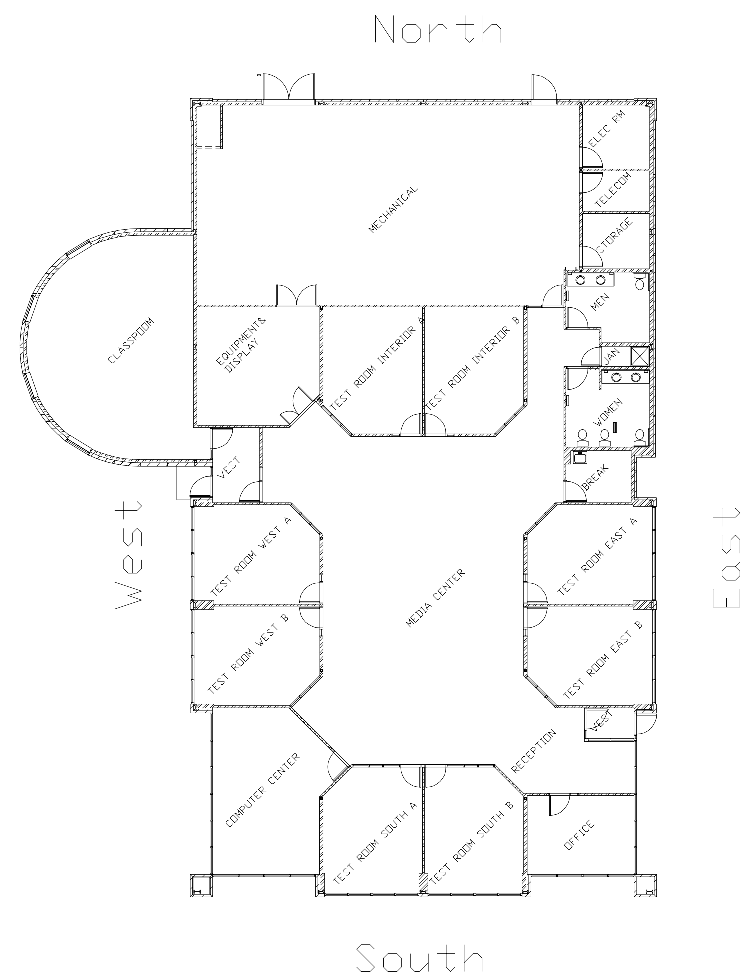

Figure A.1. A simplified floor plan of the Energy Resource station

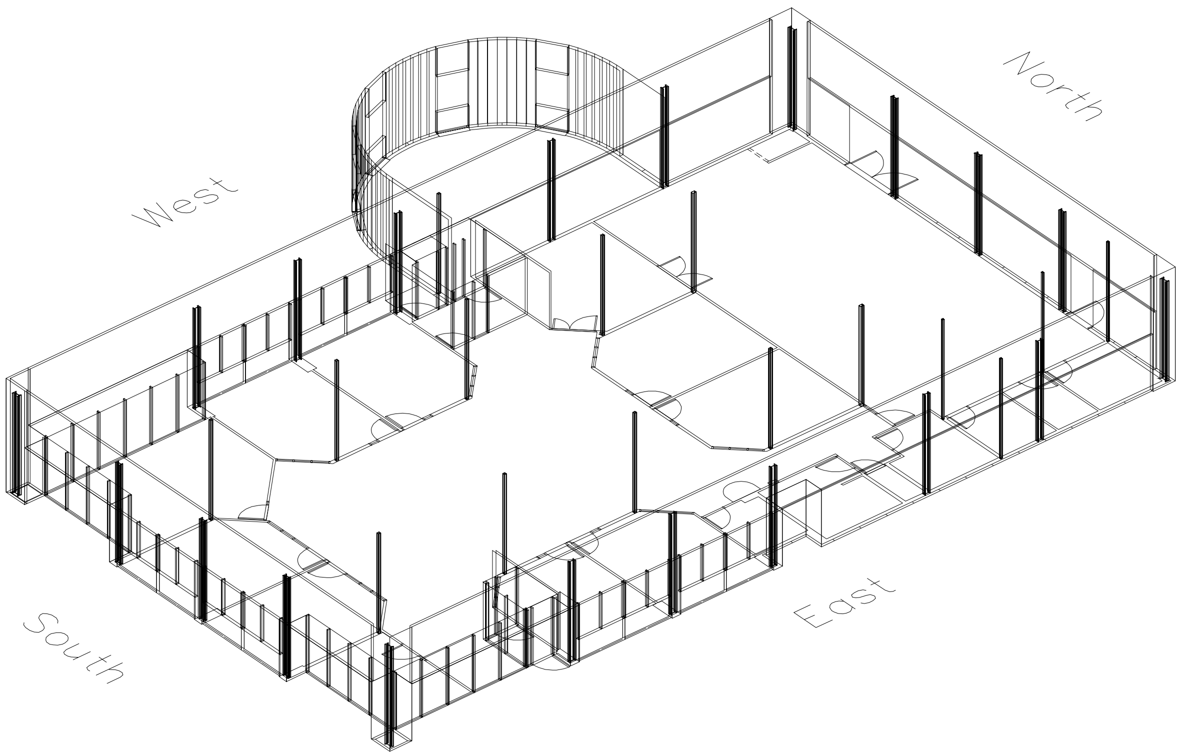

Figure A.2. A three dimensional view of the Energy Resource Station

Table A.2. Thickness and thermal properties used for construction layers

Layer type	Description	T	K	D	Cp	R
LAY-R1	Inside surface					
	2 in heavy weight concrete	2.00	0.7576	140	0.2	0.22
	4 in horizontal air space	4.00	-	-	-	0.87
	2 in heavy weight concrete	2.00	0.7576	140	0.2	0.22
	Vapor barrier	-	-	-	-	0.06
	4 in insulation	4.00	0.0133	1.5	0.38	25.06
	Single-ply membrane	-	-	70	0.35	0.44
	Washed river rock	1.00	0.8340	55	0.4	0.10
	Outside surface					
LAY-R2	Inside surface					
	22 gage steel deck	0.034	26.0	480	0.1	
	4 in insulation	4.00	0.0133	1.5	0.38	25.06
	Single-ply membrane	-	-	70	0.35	0.44
	Washed river rock	1.00	0.8340	55	0.4	0.10
	Outside surface					
LAY-W1	Inside surface					
	5/8 in gypsum board	0.63	0.0926	50	0.2	0.56
	Vapor barrier	-	-	-	-	0.06
	3/8 in vertical air space	0.38	-	-	-	0.90
	1.5 in rigid insulation with foil face	1.50	0.0133	1.5	0.38	9.39
	4 in pre-cast conc.	4.00	0.7576	140	0.2	0.44
	Outside surface					
LAY-W2	Inside surface					
	5/8 in gypsum board	0.63	0.0926	50	0.2	0.56
	3/8 in vertical air space	0.38	-	-	-	0.90
	1 in rigid insulation with foil face	1.00	0.0133	1.5	0.38	6.26
	6 in pre-cast conc.	6.00	0.7576	140	0.2	0.66
	Outside surface					
LAY-W3	Inside surface					
	5/8 in gypsum board	0.63	0.0926	50	0.2	0.56
	Vapor barrier	-	-	-	-	0.06
	Metal stud framing with R13 batt insulation with foil face	3.50	0.0250	0.6	0.2	12.96
	1 in rigid insulation	1.00	0.0133	1.5	0.38	6.26
	4.75 in vertical air space	4.75				0.92
	1 in spandrel glass	1.00	-	-	-	2.08
	Outside surface					

Table A.2 (continued)

Layer type	Description	T	K	D	Cp	R
LAY-W4	Inside surface					
	Metal stud framing with R13 batt insulation with foil face	3.50	0.0250	0.6	0.2	12.96
	3/4 in vertical air space	0.75	-	-	-	0.90
	1 in rigid insulation	1.00	0.0133	1.5	0.38	6.26
	6 in pre-cast conc.	6.00	0.7576	140	0.2	0.66
	Outside surface					
LAY-W5	Inside surface					
	3/4 in gypsum board	0.75	0.0926	50	0.2	0.67
	Vapor barrier	-	-	-	-	0.06
	Metal stud framing with R13 batt insulation with foil face	3.50	0.0250	0.6	0.2	12.96
	1 3/8 in vertical air space	1.38	-	-	-	0.89
	1 in rigid insulation	1.00	0.0133	1.5	0.38	6.26
	6 in pre-cast conc.	6.00	0.7576	140	0.2	0.66
	Outside surface					
LAY-W6	Inside surface					
	5/8 in gypsum board	0.63	0.0926	50	0.2	0.56
	Vapor barrier	-	-	-	-	0.06
	Metal stud framing with R13 batt insulation with foil face	3.50	0.0250	0.6	0.2	12.96
	3/4 in vertical air space	0.75	-	-	-	0.90
	1 in rigid insulation	1.00	0.0133	1.5	0.38	6.26
	4 in pre-cast conc.	4.00	0.7576	140	0.2	0.44
	Outside surface					
LAY-W7	Inside surface					
	5/8 in gypsum board	0.63	0.0926	50	0.2	0.56
	Metal stud framing with R13 batt insulation with foil face	3.50	0.0250	0.6	0.2	12.96
	3/4 in vertical air space	0.75	-	-	-	0.90
	1 in rigid insulation	1.00	0.0133	1.5	0.38	6.26
	6 in pre-cast conc.	6.00	0.7576	140	0.2	0.66
	Outside surface					
LAY-P1	5/8 in gypsum board	0.63	0.0926	50	0.2	0.56
	Metal stud framing with fiberglass fill, insulation	3.50	0.0225	3.0	0.33	12.96
	5/8 in gypsum board	0.63	0.0926	50	0.2	0.56
LAY-P2	5/8 in gypsum board	0.63	0.0926	50	0.2	0.56
	Metal stud framing with fiberglass fill, insulation	2.37	0.0225	3.0	0.33	8.78
	5/8 in gypsum board	0.63	0.0926	50	0.2	0.56
LAY-P3	1/8 in glass with steel frame	1/8	0.797	138	0.18	0.013
LAY-P4	Door	1.75	-	-	-	4.16
LAY-C1	Ceiling	0.75	0.033	18	0.32	1.89
LAY-F1	Carpet	-	-	-	0.34	1.23
	4 in heavy weight conc.	4.00	0.7576	140	0.20	0.44
	Perimeter insulation with a 2 inch wide	-	-	-	-	5.00

7. WINDOW TYPE AND DESCRIPTION

This section specifies the type of glass used in a window, the size and location of the window. This is used to describe the window in an exterior wall or skylight in a roof. Each type of window has information about the number of panes, shading coefficient, heat conductance of the total window (except for the outside film coefficient), width and height of the window. The windows in the exterior wall are located 3.5 feet above the floor level, and the skylight is on the roof of the media center. Units are English, and the glass conductance does not include the outside film coefficient but does include the frame. Table A.3 identifies window type and illustrates its position, size, shading coefficient and conductance.

W: width, in feet
H: height, in feet
P: number of panes
S: shading coefficient
C: heat conductance of the total window, in Btu/(hr-ft^2-^{o}F)

Table A.3. Window identification and its characteristics with size

Type	Location	W	H	P	S	C
WIN-TEST	Exterior wall in test rooms	14.0	5	2	0.85	0.55
WIN-TYP1	Exterior wall east in the office	11.8	5	2	0.31	0.30
WIN-TYP2	Exterior wall south in the office	15.3	5	2	0.31	0.30
WIN-TYP3	Exterior wall south in the computer room	15.3	5	2	0.31	0.30
WIN-TYP4	Exterior wall west in the computer room	24.0	5	2	0.31	0.30
WIN-TYP5	Exterior wall south in the classroom	3.5	5	2	0.31	0.30
WIN-TYP6	Exterior wall west in the classroom	7.0	5	2	0.31	0.30
WIN-TYP7	Exterior wall north in the classroom	3.5	5	2	0.31	0.30
WIN-TYP8	Exterior wall east in the reception room	7.9	5	2	0.31	0.30
WIN-TYP9	Door in vest east and west	3.0	7.0	2	0.31	0.30
WIN-SKY	Roof of the media center	10.0	10	1	0.35	0.24

8. SPACE DESCRIPTION

This section identifies each space. Once all spaces have been identified, then each surface of the space is described in terms of orientation, width, height and construction layer. Gross surface areas are presented in this section. Thus the areas include door and/or window areas. Window data were presented in Section 7. The size of a door is 3 feet wide and 7 feet tall.

8.1 SPACE IDENTIFICATION

Most of the rooms have a plenum space and a conditioned space. The mechanical room and storage room do not have plenum spaces. The ceiling height of most rooms is 8.5 feet, and the plenum height is 5.5 feet. Detailed information about the size is illustrated in Section 8.2. Since the test rooms are matched pairs, information provided on each orientation applies to either room. Table A.4 identifies a space as either plenum space or conditioned space. Plenum space is designated with the prefix “P.

Table A.4. Identification of plenum and conditioned space

Space-ID	Description
P-EAST	Plenum in the test room east
P-SOUTH	Plenum in the test room south
P-WEST	Plenum in the test room west
P-INTE	Plenum in the test room interior
P-BREAK	Plenum in the break room, restrooms of women and men
P-RECEPT	Plenum in the reception room
P-OFFICE	Plenum in the office
P-COMPUTE	Plenum in the computer center
P-CLASS	Plenum in the classroom
P-DISPLAY	Plenum in the display room
P-MEDIA	Plenum in the media center
EASTROOM	Conditioned space in the test room east
SOUTHROOM	Conditioned space in the test room south
WESTROOM	Conditioned space in the test room west
INTEROOM	Conditioned space in the test room interior
BREAKROOM	Conditioned space in the break room, restrooms of women and men
RECEPTION-RM	Conditioned space in the reception room
OFFICE	Conditioned space in the office
COMPUTE-RM	Conditioned space in the computer center
CLASSROOM	Conditioned space in the classroom
DISPLAY-RM	Conditioned space in the display room
STORAGE-RM	Conditioned space in the storage room, elec./comm. room
MEDIA-CENTER	Conditioned space in the media center
MECH-ROOM	Conditioned space in the mechanical room

8.2 SPACE DESCRIPTION

Each space has at least six surfaces associated with it, but for simplification, it is assumed that all spaces have six surfaces. For a better understanding of the surface geometry, a capital letter representing the position of the surface is used.

C: a horizontal surface used for the ***ceiling***
E: a vertical surface used for the ***wall east***
F: a horizontal surface used for the ***floor***
N: a vertical surface used for the ***wall north***
R: a horizontal surface used for the ***roof***
S: a vertical surface used for the ***wall south***
W: a vertical surface used for the ***wall west***

Table A.5 describes the spaces identified in Section 8.1 with detailed information about the surfaces. As mentioned in Section 6.1, all walls used in the building are vertical, and the ceiling, roof and floor are horizontal.

There are several things the modeler should remember when using this information. Be familiar with the locations and the names of the rooms in the ERS. The modeler will be comfortable if he has a simple floor plan of the building. This building is oriented for a true north/south. In the same way, the space identified in Table A.5 was described by surface orientation such as north, east, south and west. For example, let us assume that we model the space called "P-EAST" that is a plenum space in the test room east. P-EAST is located east side of the building at FL+8.5. It is surrounded by six surfaces: one east-facing exterior wall, one interior wall north, one interior wall south, one interior wall west, one ceiling and one roof. Once the surface orientations are specified, detailed information about the six surfaces that make up "P-EAST" must be provided. This includes the size of the surface, the construction layer of the surface, and any windows or doors, if present.

For a better understanding, let's take another example for a conditioned space called "SOUTHROOM" that is located on the south side of the building at FL+0.0. This space also is surrounded by six surfaces: one south-facing exterior wall that has a window, one north-facing interior wall that has a door, one east-facing interior wall, one west-facing interior wall that is adjacent to the computer room, one ceiling that is adjacent to the plenum space called "P-SOUTH", one floor.

Table A.5. Description of the space and details of its six surfaces

Space	Orientation	Width (ft)	Height (ft)	Layer	Window	Door
P-EAST	R	17.74	15.50	LAY-R1	-	-
	C	17.74	15.50	LAY-C1	-	-
	N	17.74	5.50	LAY-P2	-	-
	E	15.50	5.50	LAY-W2	-	-
	S	17.74	5.50	LAY-P2	-	-
	W	15.50	5.50	LAY-P1	-	-
P-SOUTH	R	15.50	17.74	LAY-R1	-	-
	C	15.50	17.74	LAY-C1	-	-
	N	15.50	5.50	LAY-P2	-	-
	E	17.74	5.50	LAY-P2	-	-
	S	15.50	5.50	LAY-W2	-	-
	W	17.74	5.50	LAY-P1	-	-
P-WEST	R	17.74	15.50	LAY-R1	-	-
	C	17.74	15.50	LAY-C1	-	-
	N	17.74	5.50	LAY-P2	-	-
	E	15.50	5.50	LAY-P1	-	-
	S	17.74	5.50	LAY-P2	-	-
	W	15.50	5.50	LAY-W2	-	-
P-INTE	R	15.50	17.74	LAY-R1	-	-
	C	15.50	17.74	LAY-C1	-	-
	N	15.50	5.50	LAY-P2	-	-
	E	17.74	5.50	LAY-P2	-	-
	S	15.50	5.50	LAY-P2	-	-
	W	17.74	5.50	LAY-P1	-	-
P-BREAK	R	10.66	36.60	LAY-R1	-	-
	C	10.66	36.60	LAY-C1	-	-
	N	10.66	6.00	LAY-P2	-	-
	E	36.60	6.00	LAY-W7	-	-
	S	10.66	6.00	LAY-P2	-	-
	W	36.60	6.00	LAY-P2	-	-

Table A.5. (continued)

Space	Orientation	Width (ft)	Height (ft)	Layer	Window	Door
P-RECEPT	R	17.74	13.00	LAY-R1	-	-
	C	17.74	13.00	LAY-C1	-	-
	N	17.74	5.50	LAY-P2	-	--
	E	13.00	5.50	LAY-W4	-	-
	S	-	-	-	-	-
	W	-	-	-	-	-
P-OFFICE	R	16.40	12.10	LAY-R1	-	-
	C	16.40	12.10	LAY-C1	-	-
	N	-	-	-	-	-
	E	12.10	5.50	LAY-W4	-	-
	S	16.40	5.50	LAY-W4	-	-
	W	12.10	5.50	LAY-P1	-	-
P-COMPUTE	R	16.30	25.10	LAY-R1	-	-
	C	16.30	25.10	LAY-C1	-	-
	N	16.30	5.50	LAY-P2	-	-
	E	25.10	5.50	LAY-P1	-	-
	S	16.30	5.50	LAY-W4	-	-
	W	25.10	5.50	LAY-W4	-	-
P-CLASS	R	22.20	34.67	LAY-R2	-	-
	C	22.20	34.67	LAY-C1	-	-
	N	22.20	1.00	LAY-W5	-	-
	E	-	-	-	-	-
	S	22.20	1.00	LAY-W5	-	-
	W	34.67	1.00	LAY-W5	-	-
P-DISPLAY	R	17.83	17.74	LAY-R1	-	-
	C	17.83	17.74	LAY-C1	-	-
	N	17.83	5.50	LAY-P2	-	-
	E	17.74	5.50	LAY-P1	-	-
	S	-	-	-	-	-
	W	-	-	-	-	-
P-MEDIA	R	30.00	60.80	LAY-R1	-	-
	C	30.00	57.20	LAY-C1	-	-
	N	-	-	-	-	-
	E	-	-	-	-	-
	S	-	-	-	-	-
	W	6.00	5.50	LAY-W7	-	-
EASTROOM	C	17.74	15.50	LAY-C1	-	-
	F	17.74	15.50	LAY-F1	-	-
	N	17.74	8.50	LAY-P2	-	-
	E	15.50	8.50	LAY-W1	WIN-TEST	
	S	17.74	8.50	LAY-P2	-	-
	W	15.50	8.50	LAY-P3	-	LAY-P4
SOUTHROOM	C	15.50	17.74	LAY-C1	-	-
	F	15.50	17.74	LAY-F1	-	-
	N	15.50	8.50	LAY-P3	-	LAY-P4
	E	17.74	8.50	LAY-P2		
	S	15.50	8.50	LAY-W1	WIN-TEST	-
	W	17.74	8.50	LAY-P1	-	-
WESTROOM	C	17.74	15.50	LAY-C1	-	-
	F	17.74	15.50	LAY-F1	-	-
	N	17.74	8.50	LAY-P2	-	-
	E	15.50	8.50	LAY-P3	-	LAY-P4
	S	17.74	8.50	LAY-P2	-	-
	W	15.50	8.50	LAY-W1	WIN-TEST	-
INTEROOM	C	15.50	17.74	LAY-C1	-	-
	F	15.50	17.74	LAY-F1	-	-
	N	15.50	8.50	LAY-P2	-	-
	E	17.74	8.50	LAY-P2	-	-
	S	15.50	8.50	LAY-P3	-	LAY-P4
	W	17.74	8.50	LAY-P1	-	-
BREAKROOM	C	10.66	36.60	LAY-C1	-	-
	F	10.66	36.60	LAY-F1	-	-
	N	10.66	8.00	LAY-P2	-	-
	E	36.60	8.00	LAY-W6	-	-
	S	10.66	8.00	LAY-P2	-	-
	W	36.60	8.00	LAY-P2	-	LAY-P4

Table A. 5 (continued)

Space	Orientation	Width (ft)	Height (ft)	Layer	Window	Door
RECEPTION-RM	C	17.74	13.00	LAY-C1	-	-
	F	17.74	13.00	LAY-F1	-	-
	N	17.74	8.500	LAY-P2	-	-
	E	13.00	8.50	LAY-W4	WIN-TYP8	-
	S	17.74	8.50	LAY-P2	-	-
	W	-	-	-	-	-
OFFICE	C	16.40	12.10	LAY-C1	-	-
	F	16.40	12.10	LAY-F1	-	-
	N	16.40	8.50	LAY-P2	-	LAY-P4
	E	12.10	8.50	LAY-W3	WIN-TYP1	-
	S	16.40	8.50	LAY-W3	WIN-TYP2	-
	W	12.10	8.50	LAY-P1	-	-
COMPUTER-RM	C	16.30	25.10	LAY-C1	-	-
	F	16.30	25.10	LAY-F1	-	-
	N	16.30	8.50	LAY-P2	-	-
	E	25.10	8.50	LAY-P1	-	LAY-P4
	S	16.30	8.50	LAY-W3	WIN-TYP3	-
	W	25.10	8.50	LAY-W3	WIN-TYP4	-
CLASSROOM	C	22.20	34.67	LAY-C1	-	-
	F	22.20	34.67	LAY-F1	-	-
	N	22.20	9.00	LAY-W5	WIN-TYP7	-
	E	34.16	9.00	LAY-P1	-	LAY-P4
	S	22.20	9.00	LAY-W5	WIN-TYP5	-
	W	34.67	9.00	LAY-W5	WIN-TYP6	-
DISPLAY-RM	C	17.83	17.74	LAY-C1	-	-
	F	17.83	17.74	LAY-F1	-	-
	N	17.83	8.50	LAY-P2	-	-
	E	17.74	8.50	LAY-P1	-	-
	S	17.83	8.50	LAY-P2	-	LAY-P4
	W	17.74	8.50	LAY-P2	-	-
STORAGE-RM	C	10.55	25.30	LAY-C1	-	-
	F	10.55	25.30	LAY-F1	-	-
	N	10.55	14.00	LAY-W6	-	-
	E	25.30	14.00	LAY-W6	-	-
	S	10.55	14.00	LAY-P2	-	-
	W	15.30	14.00	LAY-P2	-	LAY-P4
MEDIA-CENTER	R	10.50	10.50	LAY-R1	WIN-SKY	-
	C	30.00	57.20	LAY-C1	-	-
	F	30.00	60.80	LAY-F1	-	-
	N	-	-	-	-	-
	E	-	-	-	-	-
	S	-	-	-	-	-
	W	6.00	8.50	LAY-W6	WIN-TYP9	
MECH-ROOM	R	66.30	30.60	LAY-R1	-	-
	F	66.30	30.60	LAY-F1	-	-
	N	57.80	14.00	LAY-W7	-	-
	E	25.30	14.00	LAY-P2	-	-
	S	57.80	14.00	LAY-P2	-	LAY-P4
	W	25.30	14.00	LAY-W7	-	-

9. TEST ROOMS OPERATION

The following conditions apply to all of the test rooms. These conditions do not apply to the rest of the building where occupants may be present and lighting and window shading devices are used.

- No shading device on windows.
- No infiltration.
- Other details about the operation of the test rooms are test specific and are provided in Appendix B.

Part 2: OUTPUT REPORTS

This section describes the output desired from each model. This output parameters were used to compare the results from the various models to each other as well as to compare the model results to the actual building data collected at the ERS during the test periods. ERS test data are only available for the systems and spaces associated with the test rooms.

1. INPUT VERIFICATION REPORT

This report will be used to verify building information modelers used for the ERS building. The report should include the following information.

1.1 General
- Latitude
- Longitude
- Altitude
- Time zone

1.2 Summary of spaces occurring in the model
- Number of spaces
- Number of exterior walls
- Space information: name, height, area

1.3 Details of exterior surfaces occurring in the model
- Number of exterior surfaces
- Surface information: name, height, width, azimuth angle, tilt angle and U-value

1.4 Details of interior surfaces occurring in the model
- Number of interior surfaces
- Surface information: name, area and U-value

1.5 Details of windows occurring in the model
- Number of windows
- Window information: name, height, width, shading coefficient and U-value

2. SUMMARY REPORT

This report will be used to compare simulation results from the models. Results should be reported on an hourly basis where hour 1 represents the end of the time interval midnight to 1 AM local standard time. The report should include the following information.

2.1 Weather report

- Month, day and hour
- Outside air-dry bulb temperature and wet-bulb temperature
- Solar irradiation (direct normal and total horizontal)

Table 6 is an example of the weather report for July 26, 1998. Values in the table should correspond to data from the Ankeny.IA1 TMY weather file with appropriate conversion of units.

Table A.6. A sample global report

Month	Day	Hour[a]	Db-temp[b]	Wb-temp[c]	Dir-solar[d]	Hor-solar[e]
7	26	1	69	66	0	0
7	26	2	67	64	0	0
7	26	3	66	63	0	0
7	26	4	64	62	0	0
7	26	5	65	63	0	0
7	26	6	67	64	31	16
7	26	7	69	65	73	49
7	26	8	72	66	110	93
7	26	9	75	67	168	156
7	26	10	76	68	195	215
7	26	11	78	70	140	226
7	26	12	78	68	136	243
7	26	13	79	68	169	263
7	26	14	81	69	211	268
7	26	15	81	69	246	255
7	26	16	80	68	167	171
7	26	17	79	68	76	98
7	26	18	78	68	46	69
7	26	19	76	69	36	28
7	26	20	74	68	4	3
7	26	21	71	67	0	0
7	26	22	69	66	0	0
7	26	23	69	66	0	0
7	26	24	69	66	0	0

Note (a) Hour is local standard time.
(b) Db-temp is the dry-bulb temperature, ^oF
(c) Wb-temp is the wet-bulb temperature, ^oF
(d) Dir-solar is the direct normal solar radiation, $Btu/hr.ft^2$
(e) Hor-solar is the total horizontal radiation, $Btu/hr.ft^2$

2.2 Zone report

- Month, day and hour
- Load without ventilation load
- Zone temperature*
- Supply air flow rate*
- Reheat energy*

Table A.7 is an example of a zone report for East Test Room A on July 26, 1998. The hour in the table represents for local standard time.

Table A.7. A sample zone report for East Test Room A

Month	Day	Hour[a]	Load[b]	Zn-temp[c]	Zn-cfm[d]	Htg-engy[e]
7	26	1	-293	72.1	600	-10327
7	26	2	-354	72.1	600	-10374
7	26	3	-391	72.1	600	-10457
7	26	4	-446	72.1	600	-10531
7	26	5	-444	72.1	600	-10555
7	26	6	889	72.1	600	-9239
7	26	7	2933	72.2	600	-7242
7	26	8	4698	72.3	600	-5498
7	26	9	6132	72.3	600	-4082
7	26	10	6273	72.4	600	-3925
7	26	11	4745	72.3	600	-5386
7	26	12	3136	72.2	600	-6885
7	26	13	1970	72.2	600	-8056
7	26	14	1348	72.1	600	-8645
7	26	15	989	72.1	600	-8965
7	26	16	869	72.1	600	-9080
7	26	17	813	72.1	600	-9156
7	26	18	685	72.1	600	-9243
7	26	19	364	72.1	600	-9586
7	26	20	44	72.1	600	-9877
7	26	21	-149	72.1	600	-10098
7	26	22	-250	72.1	600	-10217
7	26	23	-287	72.1	600	-10279
7	26	24	-308	72.1	600	-10299

Note (a) Hour is local standard time.
(b) Load is the zone total load in Btu/hr which includes conduction load through walls and roofs, solar load through windows, internal loads by people, lights and equipment, and infiltration load. This value does not include the ventilation load
(c) Zn-temp is the dry-bulb temperature for the zone, ^{o}F
(d) Zn-cfm is the discharge airflow rate from the VAV box, ft^3/min
(e) Htg-engy is the zone heating coil energy, Btu/hr.
* These parameters are either directly measured at the ERS or they can be calculated from measured parameters.

2.3 System report

- Month, day and hour
- Supply air flow rate*
- Outside air flow rate*
- Temperature of air entering cooling coil*
- Temperature of air leaving cooling coil*
- Temperature of return air*
- Cooling coil energy input*

Table A.8 is an example of a system report for system-A on July 16, 1998.

Table A.8 A sample system report for system-A

Month	Day	Hour[a]	Sa-cfm[b]	Oa-cfm[c]	Clg-eat[d]	Clg-lat[e]	Ra-temp[f]	Clg-engy[g]
7	26	1	2060	400	74.4	53.1	75.6	56407
7	26	2	2060	400	74.0	53.1	75.5	53465
7	26	3	2060	400	73.7	53.0	75.5	52195
7	26	4	2060	400	73.3	53.0	75.5	50990
7	26	5	2060	400	73.5	53.0	75.5	52115
7	26	6	2060	400	73.9	53.0	72.7	53399
7	26	7	2060	400	74.3	53.1	72.7	54550
7	26	8	2060	400	74.8	53.1	72.7	55767
7	26	9	2060	400	75.4	53.1	72.7	57024
7	26	10	2060	400	75.6	53.1	75.5	58399
7	26	11	2060	400	76.0	53.1	75.5	61236
7	26	12	2060	400	76.0	53.1	75.5	58355
7	26	13	2060	400	76.2	53.1	75.5	58424
7	26	14	2060	400	76.7	53.1	75.5	59835
7	26	15	2060	400	76.7	53.1	75.6	59860
7	26	16	2060	400	76.5	53.1	75.6	58583
7	26	17	2060	400	76.3	53.1	75.6	58880
7	26	18	2060	400	76.2	53.1	75.6	58571
7	26	19	2060	400	75.8	53.1	75.6	60002
7	26	20	2060	400	75.4	53.0	75.6	58475
7	26	21	2060	400	74.8	53.1	75.6	57284
7	26	22	2060	400	74.4	53.1	75.6	55851
7	26	23	2060	400	74.4	53.1	75.6	55843
7	26	24	2060	400	74.4	53.1	75.5	55797

Note (a) Hour is local standard time.
(b) Sa-cfm is the total system supply airflow rate, CFM
(c) Oa-cfm is the outside airflow rate, CFM
(d) Clg-eat is the temperature of air entering the cooling coil, ^{o}F
(e) Clg-lat is the temperature of air leaving the cooling coil, ^{o}F
(f) Ra-temp is the return air temperature on the down stream side of the return fan, ^{o}F
(g) Clg-engy is the total central cooling coil energy, Btu/hr
* These parameters are either directly measured at the ERS or they can be calculated from measured parameters.

Appendix B Documentation on the Energy Resource Station Exercises

The purpose of this appendix is to describe the documents and data found on the CD ROM that pertain to the specific tests conducted at the Energy Resource Station for the three empirical validation exercises described in this report. This information is provided so other researchers and model developers will have access to the information and results used by the Task 22 participants in the ERS exercises.

Separate folders are used for each exercise. The folder names are the beginning dates (in year-month-day format) for each exercise. The first exercise is folder 990618, the second exercise is folder 990205, and the third exercise is folder 990612. The three exercises were not done in chronological order, but are ordered according to the type of test. The first exercise is a constant-air-volume system with moderate internal loads. The second exercise is a variable-air-volume system with scheduled internal loads. The third exercise is a variable-air-volume system with variable internal loads along with an operating schedule for the VAV system.

The structure for each test folder is the same. Each folder contains six files and one sub-folder. The first exercise folder (990618) will be used as an example. Table B.1 summarizes the file names, file types, and file contents.

Table B.1. File names in the 990618 folder

File name	File type	Contents
Data point names and locations	PDF	Drawings illustrating the relative locations of sensors and their point names for the test rooms and the air-handling units.
990618 test description	PDF	Description of the test conditions (including operating conditions, schedules, set points, etc.)
990618 hourly results	Excel	Hourly averaged values of data collected during the test. These data were compared to model output for validation purposes. The values are in SI.
Nomenclature for hourly results	Word	Description of each variable name that appears in the hourly data spreadsheet file.
Nomenclature for raw data	Word	Description of each variable name that appears in the minute-by-minute data. These data are in the sub-folder RAWdata.
Ankeny.ia1	Text	TMY weather file modified to include actual weather data collected during the test period.
RAW data	folder	12 additional data files containing one-minute interval data collected during the test period.

The sub-folder RAWdata has twelve text files containing the minute-by-minute data collected during the test period. These data are in English units. These data were used to produce the hourly-averaged values used for model validation.

The first two columns of each file contain the date (in year-month-day format) and time for each data value recorded during the test. The time is the local time for Ankeny, Iowa, which is in the central time zone in the United States. Depending on the time of year, daylight saving time may be in affect. The remaining columns contain data from specific points in the data acquisition system. Refer to the "Nomenclature for raw data" file for a description of the data point names, and refer to the "Data point names and locations" file for diagrams illustrating the locations of the points. Table B.2 summarizes the file names and file contents for the RAW data sub-folder.

Table B.2. File names in the sub-folder RAWdata under the folder 990618

File name	Contents
990618adjT	Temperatures in spaces adjacent to the test rooms.
990618ahuA	Temperatures, flow rates, fan power, etc. for air-handling unit A.
990618ahuB	Temperatures, flow rates, fan power, etc. for air-handling unit B.
990618eastA	Temperatures, reheat coil power, airflow rates, etc. for the East A test room.
990618eastB	Temperatures, reheat coil power, airflow rates, etc. for the East B test room.
990618intA	Temperatures, reheat coil power, airflow rates, etc. for the Interior A test room.
990618intB	Temperatures, reheat coil power, air flow rates, etc. for the Interior B test room.
990618southA	Temperatures, reheat coil power, airflow rates, etc. for the South A test room.
990618southB	Temperatures, reheat coil power, airflow rates, etc. for the South B test room.
990618westA	Temperatures, reheat coil power, airflow rates, etc. for the West A test room.
990618westB	Temperatures, reheat coil power, airflow rates, etc. for the West B test room.
990618weather	Solar instruments and weather station data used to create the TMY weather file Ankeny.ia1.

Appendix C Modeling Baseboard Heaters at the Energy Resource Station

When internal thermal loads are modeled, it is important to know what fraction of the heat energy is convective and what fraction is radiative. Electric baseboard heaters are used at the ERS to generate sensible internal heat loads in the test rooms. This appendix presents a brief description of the baseboard heaters and some simple experiments performed to determine the radiation/convection heat transfer from the electric baseboard heaters.

Each test room at the ERS has two electric baseboard heaters. The manufacturer's power rating for each heater is 1,000 W; however, measured power levels produce values that range from 860 W to 900 W. For the West-A test room, the heater ratings are 860 W each. The heaters operate with on/off control with no provision for part-load heating. Therefore, the internal load from the baseboard heaters in the West-A test room is 860 W if one heater is on and 1,720 W if both heaters are on.

Figure C.1 is a photograph of one of the baseboard heaters in the West-A test room. Ambient room air enters the heater through an opening along the front and bottom section of the heater. The air is warmed as it passes along the aluminum fins that are attached to the electrical heating element. The air leaves through the opening visible in the picture.

Figure C.1: Baseboard heater in ERS West-A test room.

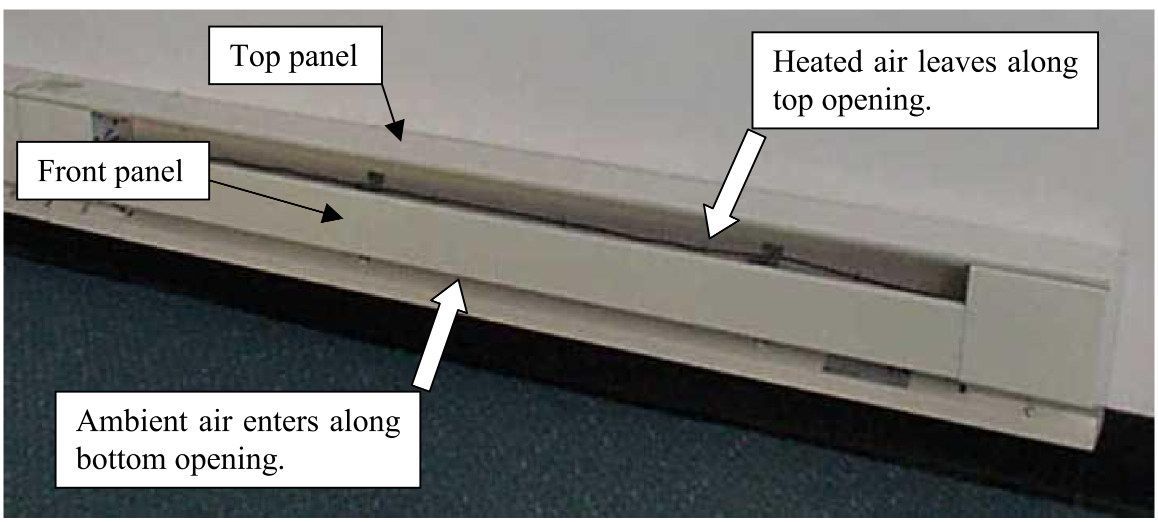

Figure C.2 shows a baseboard heat with the front panel removed. In this figure the finned heating element is visible. Figure C.3 provides a sketch showing some of the components of the heater along with their dimensions (inches).

In order to quantify the various heat transfer modes associated with the baseboard heaters, measurements were made while the unit was operating at steady-state conditions. The measurements included entering and leaving air temperatures, entering air velocity, the surface temperature of the front panel, the surface temperature of the top panel, and two heat flux measurements between the heater and the wall. The sketch also shows the location of some of the instruments that were used to measure the air temperature entering and leaving the heater as well as the air velocity entering the heater. Surface temperature measurements of the front panel and the top panel were also made.

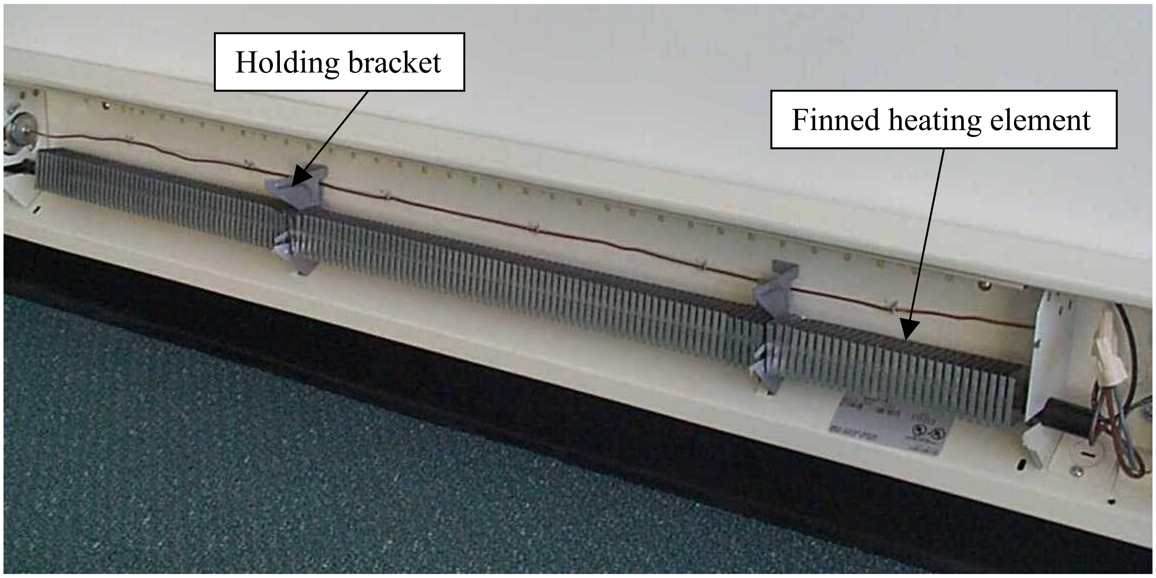

Figure C.2: Baseboard heater with front panel removed.

From the measurements, I calculated the heat transfer rates due to convection, radiation and conduction from the heater. Radiation heat transfer from the bottom of the heater and natural convection from the top and front panels were neglected. The results are summarized in Table C.1.

Table C.1 Summary of heat transfer rates for the baseboard heater.

Mode of heat transfer	Heat transfer rate	Percentage
Convection of heat to the air	893 W	95.6
Radiation from the top panel	19 W	2.0
Radiation from the front panel	7 W	0.8
Conduction to the wall	15 W	1.6
Total	**934 W**	**100**

The calculated total heat transfer rate of 934 W exceeds the power input of 860 W by 8.6% as a result of the experimental uncertainty. The important finding lies in the magnitude of the various heat transfer modes. Convection heat transfer accounts for more than 95% of the heat transfer to the room while radiation accounts for about 3%.

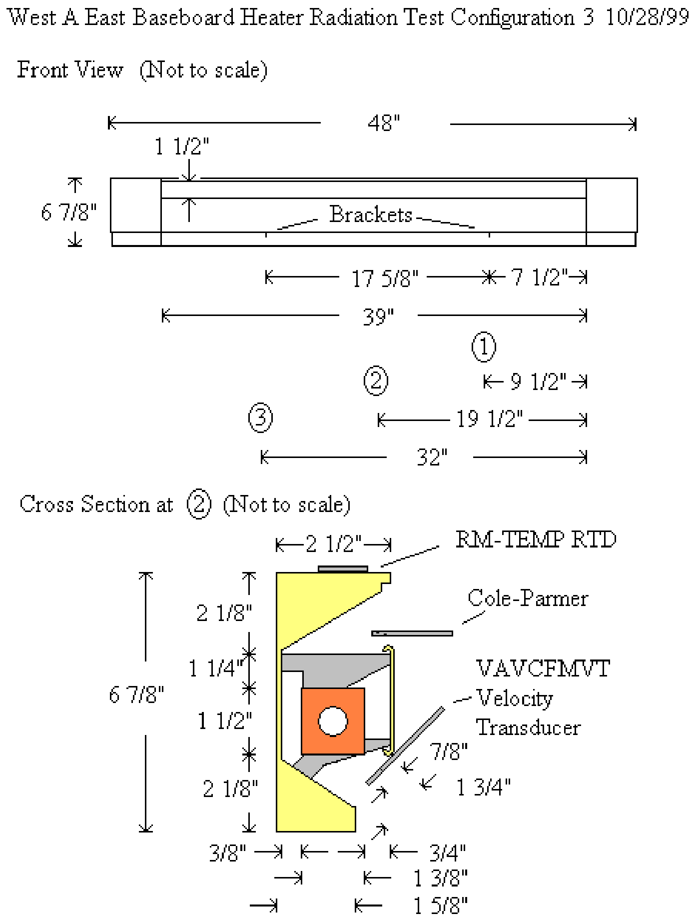

Figure C.3: Dimensions of the baseboard heater(inches).

Appendix D Experimental Uncertainty

Most errors can be categorized into two general classes: *bias error* and *precision error*. Bias error is the difference between the average value and the true value. Calibration errors, scale-reading errors, data-acquisition errors, installation errors, and systemic errors cause bias errors. Bias error can be estimated by comparison of the instrument output to a more accurate standard. Because of their nature, they tend to remain consistent from measurement to measurement within a system.

Precision error is the difference between an individual measurement and the average measured value. Precision errors are caused by random fluctuations that exist in all measurement systems. Sources of these fluctuations include electrical noise. Generally precision errors are estimated by calculating the average and standard deviation from large numbers of samples taken while holding the system constant.

An estimate of the overall accuracy of the measured data taken at the ERS is presented in Table D.1. These accuracy values are obtained either from the manufacturers' data or from calibrations performed by the ERS staff. The table summarizes the point names along with the unit of each point and the accuracy of each instrument.

Table D.1. Accuracy of ERS instrumentation

Point Name	Units	Accuracy
CHW-FLOW	gallon/min	± 0.09 gpm (0-18 gpm) ± 0.5% reading (18-80 gpm) plus ± 0.03 gpm
CLG-DAT	°F	± 3.0 °F
CLG-EWT	°F	± 0.25 °F
CLG-LWT	°F	± 0.25 °F
CLG-MWT	°F	± 0.25 °F
DA-HUMID	% RH	± 2 %RH (0-90 %RH) ± 3 %RH (90-100 %RH)
DUCT-STC	inches of WG	± 0.025 inches. WG
HTG-DAT	°F	± 3.0 °F
MA-TEMP	°F	± 0.25 °F
OA-DUCT	°F	± 0.18 °F
OA-CFM	ft^3/min	± 2 % of reading (> 500 ft^3/min) ± 10 ft^3/min (< 500 ft^3/min)
OA-TEMP	°F	N/A
RA-CFM	ft^3/min	± 2 % of reading (> 500 ft^3/min) ± 10 ft^3/min (< 500 ft^3/min)
RA-HUMID	% RH	± 2 %RH (0-90 %RH) ± 3 %RH (90-100 %RH)
RA-TEMP	°F	± 0.18 °F
RF%-SF	% speed	N/A
RF-WATTS	Watts	± 2 % of reading
SA-CFM	ft^3/min	± 2 % of reading (> 500 ft^3/min) ± 10 ft^3/min (< 500 ft^3/min)
SA-TEMP	°F	± 0.18 °F
SF-WATTS	Watts	± 2 % of reading

Table D.1. Accuracy of ERS instrumentation (Continued)

Point Name	Units	Accuracy
PLN-TEMP	°F	N/A
RM-HUMID	% RH	± 2 %RH (0-90 %RH) ± 3 %RH (90-100 %RH)
RM-TEMP	°F	± 0.25 °F
VAV-DAT	°F	± 0.25 °F
VAV-EAT	°F	± 0.25 °F
VAV-CFM	ft^3/min	N/A
VAVECWAT	Watts	N/A
Adjacent room temperature	°F	± 0.25 °F
BAR-PRES	millibars	± 0.75 millibars
OD-HUM	°F	± 0.18 °F
OD-TEMP	% RH	± 2 % RH
PYRANOM	Btu/h•ft^2	± 0.5 % of reading
PYRHELI	Btu/h•ft^2	± 0.5 % of reading
WIND-DIR	degrees	± 1 °
WIND-SPD	mile/hour	± 1 mile/hour

Propogation of Uncertainty

In some cases, measured data are used in computations to determine desired quantities. Two examples include energy balance equations used to compute the reheat energy supplied to a room and the cooling coil load. Because measurements include some *uncertainty* or *error*, these individual uncertainties will propagate into the calculated results. This is called *propagation of uncertainty*. This appendix also will address some of these propagation of uncertainty issues. The calculations will be performed for the reheat energy at the TAB and the cooling load of the cooling coil in the AHU.

Reheat Energy Error

Reheat energy rate at the TAB is calculated based on several independent measured quantities such as the air, entering and leaving air temperature as well as reheat coil power consumption.

The reheat energy rate on the cooling coil on the air side is calculated using the following formula :

$$\dot{q}_{reheat} = 60\rho_{air}C_{p,air}\dot{Q}(VAV\_DAT - VAV\_EAT) \qquad \text{(Eq. 1)}$$

$\dot{q}_{reheat}$	= VAV box reheat energy rate	[Btu/hr]
ρ_{air}	= density of air	[lbm/ft^3]
$C_{p,air}$	= specific heat of air	[Btu/lbm•R]
$\dot{Q}$	= VAV box air flow rate	[ft^3/min]

VAV_DAT	= VAV reheat coil discharged air temperature	[°F]
VAV_EAT	= VAV reheat coil inlet air temperature	[°F]

Using the properties of air at 56 °F, $C_{p,air}$ is equal to 0.2404 Btu/lbm•R and ρ_{air} is equal to 0.077 lbm/ft^3.

The uncertainty for $\dot{q}_{reheat}$ is calculated using Eq. 2.

$$\sigma^2_{\dot{q}_{reheat}} = \left(\frac{\partial \dot{q}_{reheat}}{\partial \rho_{air}}\sigma_{\rho_{air}}\right)^2 + \left(\frac{\partial \dot{q}_{reheat}}{\partial C_{p,air}}\sigma_{C_{p,air}}\right)^2 + \left(\frac{\partial \dot{q}_{reheat}}{\partial \dot{Q}}\sigma_{\dot{Q}}\right)^2 + \left(\frac{\partial \dot{q}_{reheat}}{\partial VAV\_DAT}\sigma_{VAV\_DAT}\right)^2 + \left(\frac{\partial \dot{q}_{reheat}}{\partial VAV\_EAT}\sigma_{VAV\_EAT}\right)^2 \quad \text{(Eq. 2)}$$

Using the experimental data for VAVRH test performed on June 12-16, 1999, the results are tabulated in Table D.2. The data used is only when the fan is in occupied mode.

Table D.2. VAV reheat coil energy rate uncertainty summary (air side)

$\frac{\partial \dot{q}_{reheat}}{\partial \rho_{air}}$	7,472	$\sigma_{\rho_{air}}$	0.0005
$\frac{\partial \dot{q}_{reheat}}{\partial C_{p,air}}$	2,392	$\sigma_{C_{p,air}}$	0.00005
$\frac{\partial \dot{q}_{reheat}}{\partial \dot{Q}}$	1.18	$\sigma_{\dot{Q}}$	10
$\frac{\partial \dot{q}_{reheat}}{\partial VAV\_DAT}$	542	σ_{VAV_DAT}	0.25
$\frac{\partial \dot{q}_{reheat}}{\partial VAV\_EAT}$	-542	σ_{VAV_EAT}	0.25

The average value of reheat energy rate from the experimental data is 547 Btu/hr and from the Eq. 2, the propagation error for reheat energy rate is +/- 192 Btu/hr or 35 %.

Another method of measuring the reheat energy rate is from a direct reading by a power transducer at the electric reheat coil. The only error associates with this measurement is bias error or the accuracy of the instrument. However this value is not available at the time experiment performed.

Cooling Load Error

Cooling load on cooling coil is calculated based on several independent measured quantities such as the air and water flow rate, entering and leaving air temperature as well as entering and mixing water temperature. The *propagation of uncertainty* for cooling load error is calculated in the same manner as the VAV reheat energy rate calculation.

The cooling load on the cooling coil on the air side is calculated using the following formula :

$$\dot{q}_{cooling} = 60\rho_{air} C_{p,air} \dot{Q}(HTG\_DAT - CLG\_DAT) \qquad \text{(Eq. 3)}$$

$\dot{q}_{cooling}$	= cooling coil load	[Btu/hr]
ρ_{air}	= density of air	[lbm/ft^3]
$C_{p,air}$	= specific heat of air	[Btu/lbm•R]
$\dot{Q}$	= air flow rate	[ft^3/min]
HTG_DAT	= heating coil discharged air temperature	[°F]
CLG_DAT	= cooling coil discharged air temperature	[°F]

Using the properties of air at 56 °F, $C_{p,air}$ is equal to 0.2404 Btu/lbm•R and ρ_{air} is equal to 0.077 lbm/ft^3.

The uncertainty for $\dot{q}_{cooling}$ is calculated using Eq. 4.

$$\sigma^2_{\dot{q}_{cooling}} = \left(\frac{\partial \dot{q}_{cooling}}{\partial \rho_{air}}\sigma_{\rho_{air}}\right)^2 + \left(\frac{\partial \dot{q}_{cooling}}{\partial C_{p,air}}\sigma_{C_{p,air}}\right)^2 + \left(\frac{\partial \dot{q}_{cooling}}{\partial \dot{Q}}\sigma_{\dot{Q}}\right)^2 + \left(\frac{\partial \dot{q}_{cooling}}{\partial HTG\_DAT}\sigma_{HTG\_DAT}\right)^2 + \left(\frac{\partial \dot{q}_{cooling}}{\partial CLG\_DAT}\sigma_{CLG\_DAT}\right)^2 \qquad \text{(Eq. 4)}$$

Using the experimental data for VAVRH test performed on June 12-16, 1999, the results are tabulated in Table D.3. The data used is only when the fan is in occupied mode.

Table D.3. Cooling coil load uncertainty summary (air side)

$\dfrac{\partial \dot{q}_{cooling}}{\partial \rho_{air}}$	490,747	$\sigma_{\rho_{air}}$	0.0005
$\dfrac{\partial \dot{q}_{cooling}}{\partial C_{p,air}}$	157,089	$\sigma_{C_{p,air}}$	0.00005
$\dfrac{\partial \dot{q}_{cooling}}{\partial \dot{Q}}$	21.99	$\sigma_{\dot{Q}}$	10
$\dfrac{\partial \dot{q}_{cooling}}{\partial HTG\_DAT}$	1,906	σ_{HTG_DAT}	3
$\dfrac{\partial \dot{q}_{cooling}}{\partial CLG\_DAT}$	-1,906	σ_{CLG_DAT}	3

The average value of cooling coil load from the experimental data is 37,772 Btu/hr and from the Eq. 4, the propagation error for cooling load is +/- 8,094 Btu/hr or 21 %.

The cooling load on the cooling coil on the water side is calculated using the following formula :

$$\dot{q}_{cooling} = \frac{60}{7.48055} \rho_{water} C_{p,water} \dot{Q}(CLG\_MWT - CLG\_EWT) \qquad \text{(Eq. 5)}$$

$\dot{q}_{cooling}$	= cooling coil load	[Btu/hr]
ρ_{water}	= density of water	[lbm/ft^3]
$C_{p,water}$	= specific heat of water	[Btu/lbm•R]
$\dot{Q}$	= water flow rate	[gallon/min]
CLG_MWT	= mixed chilled water temperature	[°F]
CLG_EWT	= entering chilled water temperature	[°F]

Using the properties of air at 44 °F, $C_{p,air}$ is equal to 1.004 Btu/lbm•R and ρ_{air} is equal to 62.418 lbm/ft^3.

The uncertainty for $\dot{q}_{cooling}$ is calculated using Eq. 2 as follows:

$$\sigma^2_{\dot{q}_{cooling}} = \left(\frac{\partial \dot{q}_{cooling}}{\partial \rho_{water}} \sigma_{\rho_{water}}\right)^2 + \left(\frac{\partial \dot{q}_{cooling}}{\partial C_{p,water}} \sigma_{Cp,water}\right)^2 + \left(\frac{\partial \dot{q}_{cooling}}{\partial \dot{Q}} \sigma_{\dot{Q}}\right)^2 + \left(\frac{\partial \dot{q}_{cooling}}{\partial CLG\_MWT} \sigma_{CLG\_MWT}\right)^2 + \left(\frac{\partial \dot{q}_{cooling}}{\partial CLG\_EAT} \sigma_{CLG\_EAT}\right)^2 \quad \text{(Eq. 6)}$$

Using the experimental data for VAVRH test performed on June 12-16, 1999, the results are tabulated in Table D.4. The data used is only when the fan is in occupied mode.

Table D.4. Cooling coil load uncertainty summary (water side)

$\frac{\partial \dot{q}_{cooling}}{\partial \rho_{water}}$	501	$\sigma_{\rho_{water}}$	0.0000005
$\frac{\partial \dot{q}_{cooling}}{\partial C_{p,water}}$	31,117	$\sigma_{C_{p,water}}$	0.0005
$\frac{\partial \dot{q}_{cooling}}{\partial \dot{Q}}$	2,206	$\sigma_{\dot{Q}}$	0.09
$\frac{\partial \dot{q}_{cooling}}{\partial CLG\_MWT}$	7,117	σ_{CLG_MWT}	0.25
$\frac{\partial \dot{q}_{cooling}}{\partial CLG\_EAT}$	-7,117	σ_{CLG_EAT}	0.25

The average value of cooling coil load from the experimental data is 31,247 Btu/hr and from the Eq.6, the propagation error for cooling load is +/- 2,524 Btu/hr or 8 %.