EMPIRICAL VALIDATION OF BUILDING ENERGY SIMULATION SOFTWARE: ENERGYPLUS

Som Shrestha¹, Gregory Maxwell²
¹Oak Ridge National Laboratory, Oak Ridge, Tennessee, USA
²Iowa State University, Ames, Iowa, USA

ABSTRACT
This paper compares the results from a study conducted at Iowa Energy Center’s Energy Resource Station with EnergyPlus simulation results. The building consists of controlled test rooms, dedicated air handling units and air-cooled chillers for the purpose of obtaining quality data suitable for empirical validation studies. Weather data were also collected at the facility and used for the simulation. Empirical validation can be performed on various levels of the program such as zone level, systems level, and plant level. This study is unique in the sense that it integrates the zones, system, and plant into one analysis. For this study, the difference between empirical and EnergyPlus predicted zone cooling loads varied from 1.7% to 10.2%, but the difference for the compressor power was as much as 22.4%. The paper also describes the potential reasons why simulation results might not match field data.

INTRODUCTION
Validation of building energy simulation programs is an important aspect of the software development process which provides assurance to the software developers and users that the predictions from the software are meaningful. It also offers opportunity to identify and fix anomalies in the software. There are three methods to validate a simulation program: analytical, comparative, and empirical. The underlying question for all building energy system simulation programs is how well they predict actual energy consumption in a building. Empirical validation is the only method that answers this question. Empirical validation can be performed on various levels of the program such as zone level, systems level, and plant level. This study integrates the zones, systems and plants into one analysis. This paper compares the results from a study conducted at Iowa Energy Center’s Energy Resource Station (ERS) with EnergyPlus 6.0 (E+) simulation results. ERS, located in Ankeny, Iowa, is a research and training facility with the ability to simultaneously test and demonstrate multiple, full-scale commercial building systems, which provides unique opportunities to conduct empirical validation studies. The systems and plants installed in the ERS are versatile. The facility has extensive instrumentation and electronic data acquisition systems. It also has a weather station for collecting local weather data including direct solar radiation, diffuse solar radiation, and infrared radiation. For empirical validation studies, the weather conditions at the time of a test must be provided to the simulation software if a meaningful comparison is to be made.

To perform an empirical validation study, it is essential to obtain reliable experimental data. Hence, accuracy of the sensors being used is critical to draw a valid conclusion from the study. Additionally, sensor calibration must be done periodically as specified by the manufacturer to minimize instrumentation errors. Errors in sensor readings may be detected by performing energy balances on various system components. Reliability of experimental data also depends on the stability of the control system. Measurement uncertainties for this study are described in Shrestha (2006).

The tests performed in this research were designed as part of a broader empirical validation study for the International Energy Agency (IEA) Solar Heating and Cooling Task 34/Annex 43.

FACILITY DESCRIPTION AND TEST SET UP
The ERS is comprised of eight test rooms, a computer room, offices, two classrooms and other rooms necessary for the support and operation of the facility. The mechanical room houses three air-handling units, two of which are identical and serve the test rooms. Three air cooled water chillers are located at the north side of the building. Four test rooms designated as “A” rooms are served by air handling unit “A” and the “A” chilled water loop including chiller “A”. Similarly, the other four test rooms are designated as “B” rooms are served by air handling unit “B” and the “B” chilled water loop including chiller “B”. The third air handling unit and chiller serve the rest of the facility.

The test rooms are grouped in pairs to provide simultaneous side-by-side testing. East, South and
West test rooms are located at the perimeter of the building while the interior rooms are located in the interior of the building. In this project, the “A” and “B” rooms, systems and plants were operated under the same set of test conditions; however, only data from the “A” components of the facility are used for the empirical validation.

Construction and configuration of the building and building materials are described in-depth by Price and Smith (2000). Previous research conducted at the ERS performed by Loutzenhiser (2003), Kuiken (2002) and Lee (1999) also described geometry and construction details of the building. Previous empirical validation work has focused at the zone level. In this study system and plant data are included in the empirical validation analysis

**Zone Configuration**

For this study, all exterior test rooms had identical clear glass windows with no internal or external shading devices. Each test room was equipped with a 135 W box fan located near the ceiling. The fan blows air towards the floor and promotes air mixing within the room to prevent thermal stratification. Electric baseboard heaters were used to create additional internal loads in the test rooms. The zone thermostat set-point temperature was fixed at 22.8°C for all test rooms.

**System Configuration**

The system operated as a cooling-only variable air volume system (VAV). The air handling unit (AHU) was operated with the outdoor air dampers closed; therefore, the system was 100% recirculation air. The control system maintained a discharge air temperature from the AHU (after the fan) by modulating the three-way chilled water control valve on the cooling coil. There were no provisions for controlling humidity. Both the supply fan and return fan were modulated by variable frequency drives; the supply fan speed was controlled to maintain a constant supply duct static pressure while the return fan speed was matched to the supply fan to assure a neutral air pressure in the test rooms.

**Plant Configuration**

The chiller used for the research is a McQuay air-cooled chiller model AGZ 010AS. The chiller was operated as per the manufacturer’s specification. The chilled-water temperature was set at 4.4°C, and the chilled-water pump power and flow rate were steady at 613 W ± 1% and 1.68 kg/s ± 2%, respectively. The chilled water system uses a 17.6% (by volume) propylene-glycol and water mixture.

**ENERGYPLUS MODELING**

The building parameters required to simulate the test rooms (zone models) were similar to those used by Loutzenhiser (2006). Therefore, only system modeling and plant modeling is described here.

**Supply and Return Air Fan**

A variable-volume fan model was used to characterize the supply-air fan. The fan power consumption is modeled as a fourth-degree polynomial (Equation 1). In this equation FF is the flow fraction and PLF is the fraction of full load power. The fan coefficients required for modeling the supply-air fan were calculated by curve fitting fan power versus airflow rate from the ERS test data.

\[ PLF = a + b(FF) + c(FF)^2 + d(FF)^3 + e(FF)^4 \] (1)

During the tests, the return fan power was virtually constant. Therefore, in the model, the heat dissipated from the return fan and motor was accounted for by adding fan power to the return air system.

**Cooling Coil**

In E+, cooling coils can be modeled in great detail, or the user can opt for a default coil model provided in the program. The built-in flat fin coil model is for continous plate fins. The detailed model requires a significant amount of input information for the coil. For a commercial cooling coil, much of this information is not available. Hence the built-in generic flat fin coil model available in E+ was used as a cooling coil model. Felsmann et al. (2009 part II) described modeling of this coil using experimental data.

**Air-cooled chiller**

The chiller contains two tandem hermetically-sealed scroll compressors and two condenser fans. The compressors operate in stages with a total nominal cooling capacity of 34.4 kW.

An electric Energy Input to cooling output Ratio (EIR) air-cooled water chiller model was used to characterize the chiller in E+. The model uses performance information at reference conditions along with three curve fits for cooling capacity and efficiency to determine chiller operation at off-reference conditions. To model an EIR chiller, E+ requires detailed specification of the plant. While most of these specifications are directly available in the manufacturer’s data sheet, some of the parameters require regression analysis of the values presented in the chiller’s performance tables. These parameters are discussed below.

**Cooling capacity as a function of temperature.** This curve is a biquadratic performance curve that parameterizes the variation of the chiller’s cooling capacity as a function of the chiller leaving water temperature (LWT) and the condenser entering air temperature (EAT). The curve is described by Equation 2.
ChillerCapFTemp = a + b(LWT) + c(LWT)^2 + d(EFT) + e(EFT)^2 + f(LWT)(EFT)  \hspace{1cm} (2)

Table 1 shows the chiller’s cooling capacity, compressor power and coefficient of performance (COP) for different evaporator LWTs and EFTs. The performance table is based on sea-level pressure and using water as an evaporator fluid. Corrections for the altitude and the propylene-glycol/water solution were made to match the test conditions. The COP in the table is for the entire chiller, including compressors, fan motors and control power. For E+, COP should not include fan motors and control power. Hence, COP for the model was calculated using Statistical Analysis Software (SAS). The values are:

\[ a = 0.9485 \quad b = 0.0407 \quad c = 0.00026 \]
\[ d = -0.0033 \quad e = -0.00005 \quad f = -0.00024 \]

Coefficient of determination (R^2) = 0.9999

**EIR as a function of temperature.** This curve is a biquadratic performance curve that parameterizes the variation of the EIR as a function of the LWT and the EFT as described by Equation 3.

ChillerEIRFTemp = a + b(LWT) + c(LWT)^2 + d(EFT) + e(EFT)^2 + f(LWT)(EFT)  \hspace{1cm} (3)

The EIR is the inverse of the COP. Numerical values of the coefficients of Equation 3 were calculated using SAS. The values are:

\[ a = 0.6468 \quad b = -0.0017 \quad c = 0.0004 \]
\[ d = -0.0022 \quad e = 0.0005 \quad f = -0.0009 \]

Coefficient of determination (R^2) = 0.9997

**EIR as a function of part load ratio (PLR).** This curve is a quadratic performance curve that parameterizes the variation of the EIR as a function of the chiller PLR as described by Equation 4. The PLR is the actual chiller cooling load divided by the design chiller cooling capacity. Manufacturer’s performance data gives expected EIR as a function of PLR in the range of 25% to 100% PLR. Figure 1 shows the variation of the EIR as a function PLR that was calculated from the manufacturer’s performance data.

![Figure 1 Variation of EIR as a function of PLR](image)

Modeling of the chiller and the simulation result relies on the manufacturer provided chiller performance data. However, manufacturer’s datasheet did not provide the uncertainty in the data in Table 1.

**Condenser Fan**

The condenser fans run based on the refrigerant pressure in the condenser. While one fan is constant speed, the other has a variable frequency drive to modulate the fan’s airflow in accordance to the condenser pressure. The fan sequencing was such that the variable-speed fan turns on first and turns off last while the fixed-speed fan turns on last and turns off first. The EIR air-cooled water-chiller model in E+ uses the “condenser fan power ratio” to model the condenser fan power associated with air-cooled condensers. The condenser fan power ratio is the ratio of the condenser fan power to the chiller cooling capacity at design conditions.

**Table 1 Manufacturer provided chiller performance data**

<table>
<thead>
<tr>
<th>LWT (°C)</th>
<th>25</th>
<th>30</th>
<th>35</th>
<th>40</th>
<th>45</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>kW</td>
<td>PWR Unit</td>
<td>kW</td>
<td>PWR Unit</td>
<td>kW</td>
</tr>
<tr>
<td>5</td>
<td>35.0</td>
<td>8.0</td>
<td>3.40</td>
<td>33.7</td>
<td>8.8</td>
</tr>
<tr>
<td>6</td>
<td>36.2</td>
<td>8.1</td>
<td>3.49</td>
<td>34.9</td>
<td>8.9</td>
</tr>
<tr>
<td>7</td>
<td>37.5</td>
<td>8.2</td>
<td>3.59</td>
<td>36.2</td>
<td>9.0</td>
</tr>
<tr>
<td>8</td>
<td>38.9</td>
<td>8.3</td>
<td>3.68</td>
<td>37.5</td>
<td>9.1</td>
</tr>
<tr>
<td>9</td>
<td>40.2</td>
<td>8.4</td>
<td>3.78</td>
<td>38.9</td>
<td>9.1</td>
</tr>
<tr>
<td>10</td>
<td>41.6</td>
<td>8.4</td>
<td>3.87</td>
<td>40.1</td>
<td>9.2</td>
</tr>
</tbody>
</table>
**Chilled Water Pump**

For all tests, the chilled water pump was operated at a constant speed and a constant flow rate. A 3-way control valve bypasses water around the cooling coil in order to modulate the cooling coil capacity. The pump power was stable and the average pump power variation among the tests was negligible; therefore, the pump was modeled as constant speed and constant-power pump.

**Outdoor Airflow Rate**

Although outdoor air (OA) damper was closed, measurements show that the OA damper has a small leakage rate of about 0.2% of supply airflow rate at the peak supply airflow rate. The leakage was seen to increase as the supply airflow rate increases. This is due to the negative static pressure in the air-handling unit (in the return air/outdoor air mixing section) at high supply airflow rate conditions. Therefore, the OA flow rate was calculated based on OA, return air (RA) and mixed air (MA) enthalpy and MA flow rates. The OA flow rate was entered into E+ by specifying an OA flow rate schedule.

**RESULTS**

Comparisons are made between the empirical data and the simulation results at zone, system, and plant level.

**Zone Level Comparisons**

The primary parameters of interest at the zone level are zone cooling load computed from the room temperature, the temperature of the air supplied to the room, and the cooling airflow rate supplied to the room. The room set-point temperature was fixed at 22.8°C. E+ tracked the room set-point temperature accurately. Although E+ allows for duct heat gain for the supply air system, no heat gain was modeled since the duct system is well insulated and the air temperature was only seen to increase by 0.4°C.

The sensible cooling load for each room was calculated using the supply air temperature, the room temperature, and the supply airflow rate. Neglecting changes in kinetic and potential energy, the energy balance yields:

\[
\dot{q}_s = \frac{\dot{Q}}{v} c_p (t_s - t_r)
\]  

(5)

where \(\dot{q}_s\), \(\dot{Q}\), \(v\), \(c_p\), \(t_s\), and \(t_r\) corresponds to room sensible load, supply airflow rate, specific volume of the air, specific heat of the air, the room temperature, and supply air temperature, respectively.

Figure 2 compares the sensible cooling loads for all four test rooms computed using Equation 5 and test data vs. E+ simulation output. Table 2 shows the average zone sensible cooling load and % difference between measured and predicted load. E+ predicted the cooling loads for the East and Interior zones fairly accurately but underestimated the cooling loads for the South and West zones, particularly during sunny hours.
Table 2 Comparison between measured and model predicted cooling loads

<table>
<thead>
<tr>
<th>Zone</th>
<th>Sensible cooling load, kW</th>
<th>% difference</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Measured</td>
<td>Simulation Result</td>
</tr>
<tr>
<td>East</td>
<td>2.477</td>
<td>2.384</td>
</tr>
<tr>
<td>South</td>
<td>2.654</td>
<td>2.384</td>
</tr>
<tr>
<td>West</td>
<td>2.758</td>
<td>2.619</td>
</tr>
<tr>
<td>Interior</td>
<td>2.303</td>
<td>2.259</td>
</tr>
<tr>
<td>Total</td>
<td>10.19</td>
<td>9.64</td>
</tr>
</tbody>
</table>

System Level Comparisons

System-level parameters center on the supply fan, outside airflow, and the cooling coil.

Supply Fan

For the supply fan, the parameters of interest include the airflow rate and the motor power. Figures 3 and 4 compare the measured and $E^+$ predicted supply airflow rate and supply fan motor power, respectively. The results show the measured supply airflow rate is about 10% greater than that predicted by $E^+$. This is consistent with the individual room airflow rate and sensible cooling load comparisons where the measured value of supply airflow rate to each room and cooling rate were slightly higher than that predicted by $E^+$.

Cooling Coil

For the cooling coil, the parameters of interest include the psychrometric state of the air entering and leaving the cooling coil, the supply air temperature, the temperature of the water entering and leaving the cooling coil, and the cooling coil water flow rate. Using these parameters, the heat loss by air and the heat gain by water in the cooling coil were calculated and the results compared to simulation results.

Comparison between measured values and simulation output values for the cooling coil entering air temperature (EAT) and the supply air temperature (SAT) showed good agreement. The maximum difference in EAT and SAT was about 1°C and 0.5°C, respectively.

Figure 5 compares cooling coil entering water temperature (EWT) and mixed-water temperature (MWT). The MWT is the temperature of water after the mixing control valve. Under part-load conditions, the valve reduces the chilled water flow through the coil by allowing chilled water to bypass the coil. As can be seen from the figure, the model predicts no variation in cooling coil EWT; however, a variation in the ERS chilled water temperature is seen. This is a result of the chiller’s water temperature controller which has a control dead band of approximately ± 2°C.

The chilled water flowrate through the cooling coil was not directly measured at the ERS, it was calculated based on an energy balance performed on the 3-way mixing valve using Equation 6.

Cooling coil water flow rate

\[ \text{Cooling coil water flow rate} = \frac{\text{Total chilled water flow rate}}{\text{MWT} - \text{EWT}} \]

Figure 5 Cooling coil entering water temperature and mixed water temperature comparison

The built-in generic flat cooling coil model available in $E^+$ was used as the cooling coil model for simulation. While the coil model predicted significantly less chilled water flow rate through the cooling coil, it predicted a higher leaving water...
temperature to match the net heat gain by water in the coil. Figure 6 compares calculated values of air-side and water-side cooling coil loads vs. $E^+$ predicted cooling coil load. The air-side energy balance differed from simulation result by 9.7% and the water-side energy balance differed by 17.7%. The discrepancy between water-side and the air-side energy balance is mainly due to measurement uncertainties.

![Figure 6 Comparison between measured air side and water side cooling load vs. simulation result](image)

**Plant Level Comparisons**

The chilled water pump and the air-cooled liquid chiller are the main components of the plant.

**Chilled water pump**

During the test, the chilled water pump was operated as a constant-speed pump. As described earlier, the water flowrate and the pump power were steady during the test. Likewise, the $E^+$ model predicted no variation in the chilled water flow rate and pump power. The average measured and $E^+$ predicted pump power was 613 W and 595 W, respectively.

**Air-cooled liquid chiller**

For the chiller, three parameters were considered for validation: (1) chiller cooling load, (2) compressor power and (3) condenser fan power. The chiller cooling load was computed from measured water flow rate along with entering and leaving chilled water temperatures. Compressor power and condenser fan power were measured directly. Figures 7 shows the comparison between empirical values and simulation output for the chiller cooling load. On average the $E^+$ predicted chiller cooling load was 15% lower than empirical value. This is consistent with the differences seen on the cooling coil load. Figure 8 shows the comparison between the empirical values and simulation output for compressor power. The $E^+$ predicted compressor power was 22.4% lower than the measured power.

$E^+$ calculates the condenser fan power as a fixed fraction of the chiller cooling capacity at reference condition which is a user specified input based on the rated values. Therefore, $E^+$ predicts no variation in condenser fan power. The measured condenser-fan power was 59% lower compared to $E^+$ predicted condenser-fan power.

**DISCUSSION**

Validation of building energy simulation software using the empirical method requires detailed information about the building, HVAC system and the equipment in order to produce as accurate of a model as possible. Moreover, the program user must have a complete understanding of the input parameters required to create the model. A small error in the input may lead to significant impact on the outputs. Therefore, it is essential to verify whether the computer model has the same physical and thermal property data and control algorithm as the building, system and plant being modeled.

While most of the parameters for modeling the building were readily available, some technical specifications needed to model system components (such as cooling coils) and plant components (such as the chiller) were not readily available from manufacturers’ data sheets. In particular, the chiller performance information provided by the manufacturer did not include parameters within the
operation range under which the tests were conducted. 

E+ underestimated the solar heat gain through the windows. This is the reason for lower zone sensible cooling loads estimated by the software compared to the actual room loads. This is particularly noticeable on sunny days. The windows were modeled based on manufacturer provided property data using Window5 data file. This study showed that E+ predicts zone cooling loads more accurately than system or plant cooling loads. The study also showed that E+ predicts zone cooling loads better in non-dynamic conditions compared to that in dynamic conditions. E+ predicted less chiller power than what was measured; however, this is consistent with the E+ predicted cooling coil loads being less than the experimentally measured cooling coil loads. Mostly the chiller was operating at lower atmospheric temperature and also producing chilled water at a lower temperature than the conditions covered by the manufacturer’s chiller performance data for off-design conditions. One reason for the discrepancy in compressor power is the discrepancy in evaporator cooling load. Another reason could be the operation of the chiller at the conditions not covered in the manufacturer’s specification sheet. The third reason is the discrepancy in the condenser entering air temperature and the air temperature used in the weather file.

The air temperature entering an air-cooled condenser coil affects the performance of the chiller. As the ambient air temperature increases, the chiller capacity decreases, along with the COP. This leads to an increase in compressor power required to meet the cooling load on the system. At the ERS the chillers are located in an outdoor area that has a concrete pad and is surrounded by concrete walls. The concrete walls provide an architectural barrier between the mechanical equipment and the surrounding landscape. On clear days, the solar irradiation on the walls causes the ambient air temperature in this area to rise above the atmospheric air temperature recorded by the weather station (located about 6 meters above the roof of the ERS). Also, heat dissipated by the chiller itself increases the air temperature surrounding the chiller. This increased air temperature affects the chillers performance. Figure 9 shows the air temperature measured at the weather station with the air temperature entering the condenser.

From the figure it can be seen that the temperature of the air entering the condenser is higher (on average 2.1°C) than the atmospheric temperature. E+ considers the air temperature entering the condenser to be the same as the dry-bulb temperature value in the weather file. Depending on the location of the air cooled equipment, this may not be a valid assumption.

![Figure 9 Outdoor air and condenser entering air temperature comparisons](image)

E+ evaluates the variation of the EIR (and hence the COP) as a function of the chiller LWT and the condenser EFT as described by Equation 3. An increase in condenser entering air temperature will decrease the COP of the chiller and increase the compressor power required to produce the same cooling capacity. Figure 10 shows the COP and % increase in compressor power from that at 20°C as a function of the condenser entering air temperature when the chiller leaving water temperature is constant at 4.4°C.

![Figure 10 Chiller COP and compressor power as a function of condenser entering air temperature at evaporator leaving water temperature = 4.4°C](image)

From the figure, an increase in ambient air temperature of 2°C would increase the compressor power by about 6%.

Most chillers control the chilled water temperature to a specific set point. Depending on the type of compressor controls, the chilled water temperature will fluctuate within a control dead band. For the chiller at the ERS, the chilled water temperature fluctuated +/- 2°C about the set point. The chiller model in E+ does not account for chiller control dynamics thus the chilled water temperature remains a constant.
CONCLUSIONS

This paper compares the results from a study conducted at Iowa Energy Center’s Energy Resource Station, a research and training facility, with EnergyPlus (E+) simulation results. Empirical validation can be performed on various levels of the program such as zone level, systems level, and plant level. This study is unique in the sense that it integrates the zones, system, and plant into one analysis.

The results show that good agreement between the experimental data and the E+ model can be achieved at the zone level. For this study, E+ predicted an average zone cooling load that was 5.4% lower than the average measured zone cooling load.

At the system level, greater differences were seen between predicted and measured cooling coil loads. Given the total zone cooling load predicted by E+ was 5.4 % lower than the empirical value, it is to be expected that the E+ predicted cooling coil loads would be lower than what was measured. The average error between the E+ cooling coil load and the empirical coil load was 13.7%. Thus approximately 8% difference could be attributed to the cooling coil model and the experimental uncertainties associated with the temperature and flow rate data.

At the plant level, E+ predicted a compressor power that is 22.4% lower than that measured. This difference can be attributed to the difference that exists between the cooling coil loads (13.7%) and the fact that the air temperature seen by the condenser at the ERS was warmer than the air temperature used by E+ which would result in a difference of approximately 6%.

In the “real world” the air temperature seen by the cooling equipment is often not the same as the air temperature recorded by weather station. For example, for a roof top unit on a hot roof during the summer, the air temperature entering the condenser is much greater than the air temperature far away from the roof. E+ does not allow the user to specify a different air temperature for the air-cooled equipment as compared to the air temperature used for other parts of the simulation. This study shows the need to add this feature to the model.

ACKNOWLEDGMENT

This research was funded by the Iowa Energy Center (Grant No. 96-08). Thanks to Mr. Curtis J. Klaassen for valuable advises and support and to Mr. Xiaohui Zhou and Mr. David Perry for their cooperation and assistance.

REFERENCES


