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Part-Load Performance Characterization and Energy Savings Potential of the RTU Challenge Unit: Daikin Rebel

W Wang
S Katipamula

September 2013



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Richland, Washington 99352

ABSTRACT

In 2011, the U.S. Department of Energy's Building Technology Office (DOE's BTO), with help from the Better Buildings Alliance (BBA) members, developed a specification (RTU Challenge) for high performance rooftop air-conditioning units with capacity ranges between 10 and 20 tons (DOE 2013). Daikin's Rebel was the first rooftop unit system recognized by DOE in May 2012 as meeting the RTU Challenge specifications. This report documents the development of part-load performance curves and their use with the EnergyPlus simulation tool to estimate the potential savings from the use of Rebel units compared to other standard options.

The conventional direct-expansion cooling model in EnergyPlus and its performance curve equations are not appropriate to use to capture the Rebel's performance; therefore, performance curves suitable for the Rebel unit were developed. The data for the performance curves were collected through a series of laboratory tests.

A detailed EnergyPlus model was developed for a prototypical big-box retail store. The model used the new performance curves along with detailed energy management control code to estimate the energy consumption of the prototypical big-box retail store in three locations (Houston, Los Angeles and Chicago). The energy consumption by the big-box store was then compared to a store that used three different reference units. The first reference unit (Reference 1) represents existing rooftop units (RTUs) in the field, so it can be considered as the baseline to estimate potential energy savings from other RTU replacement options. The second reference unit (Reference 2) represents RTUs in the market that just meet the current (2013) Federal regulations for commercial equipment standards. So, it can be used as the baseline to estimate the potential for energy savings from Rebel units in comparison with new RTUs that meet the minimum efficiency requirements. The third reference unit (Reference 3) represents the latest ASHRAE 90.1-2010 requirements. For RTUs with cooling capacity greater than 11,000 Btu/h, ASHRAE 90.1-2010 (ASHRAE 2010) requires two-speed fan control or variable-speed fan control.

The following conclusion can be drawn about the comparison of energy cost for Rebel unit with the three reference units:

- Using Reference 1 as the baseline, Rebel units lead to about 45% lower heating, ventilation and air conditioning (HVAC) energy cost in Houston and Los Angeles, and 33% lower cost in Chicago. The percentage savings of electricity cost averages around 50% for all three locations.
- Using Reference 2 as the baseline, Rebel units lead to about 37% lower HVAC energy cost in Houston, 40% lower cost in Los Angeles, and 29% lower cost in Chicago. The percentage savings of electricity cost is 40%, 42%, and 50%, respectively in the above three locations.
- Using Reference 3 as the baseline, Rebel units lead to about 27% lower HVAC energy cost in Houston, 18% lower cost in Los Angeles, and 15% lower cost in Chicago. The percentage savings of electricity cost is 30%, 19%, and 28%, respectively in the above three locations.

Based on the simulation results, the Rebel RTU Challenge unit, if widely adopted, could lead to significant energy, cost and emission reductions. Because the cost of these units was not available and because the costs would be specific to a given installation, no attempt was made to estimate the potential payback periods associated with any of the three reference scenarios. However, if the incremental cost for any of the three reference cases is known, one can easily estimate a simple payback period.

ACKNOWLEDGEMENT

The authors would like to acknowledge the Buildings Technologies Office of the U.S. Department of Energy (DOE) Office of Energy Efficiency and Renewable Energy for supporting this research, development and demonstration effort. The authors would also like to thank Jason Koman and Alan Schroeder, Technology Development Managers, and Linda Sandahl, Program Manager at PNNL for their valuable guidance; Nick Fernandez for providing the technical review of the document; and Sue Arey for editorial support. Yulong Xie and Christopher Pratt helped on the curve development. The authors would also like to thank Eric Schlichting, Gregg Laing and Don Winter from Daikin for valuable comments and help during testing and development of the EnergyPlus model.

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1 INTRODUCTION

In 2011, the U.S. Department of Energy’s Building Technology Office (DOE’s BTO), with help from the Better Buildings Alliance (BBA) members, developed a specification for high performance rooftop air-conditioning units (also known as the RTU Challenge) with capacity ranges between 10 and 20 tons (DOE 2013). The goal of the RTU Challenge was to spur the market introduction of cost-effective, high-performance commercial rooftop unit air conditioners. Five manufacturers—Daikin, Carrier, Lennox, 7AC Technologies, and Rheem—originally showed interest in developing products that meet the RTU Challenge requirements. Daikin’s Rebel was the first rooftop unit system recognized by DOE in May 2012 as meeting the RTU Challenge specifications.

Funded by the DOE BTO, Pacific Northwest National Laboratory (PNNL) coordinated the laboratory test of a 10-ton Rebel air-conditioning unit with gas heat. The testing was done at Intertek labs (Cortland, NY) in July, 2012. Based on the lab test data, PNNL developed performance curves to characterize the 10-ton Rebel unit’s cooling performance. Then, these curves were used in EnergyPlus simulation programs to model the Rebel unit’s performance. This document reports 1) development of performance curves for the Rebel unit; and 2) the simulated energy and cost savings for a prototypical “big-box” building at three locations (Houston, Los Angeles, and Chicago) using the Rebel unit compared to the same building that uses conventional RTUs.

The development of the Rebel performance curves are discussed in the Section 2, followed by the description of the prototypical building characteristics in Section 3. The energy management system used to model the complex controls of the Rebel unit in EnergyPlus are discussed in Section 4. The simulation results are provided in Section 5. The report concludes with a discussion in Section 6 and list of references in Section 7.

2 PERFORMANCE CURVE DEVELOPMENT FOR DAIKIN REBEL

The Daikin Rebel packaged rooftop system was the first unit to meet DOE's RTU Challenge specification. The 10-ton air-conditioner with gas furnace has two compressors with one variable-speed inverter compressor, composite condenser fan(s), variable-speed ECM (electronic commuted motor) fan motors, modulating hot gas reheat, MicroTech® III controls that can be integrated with optional BACnet or LonMark building automation systems, electronic expansion valves (Figure 1). Unlike the conventional units, Rebel modulates the supply fan in response to the zone conditions and modulates the compressor speed to maintain discharge air temperatures.

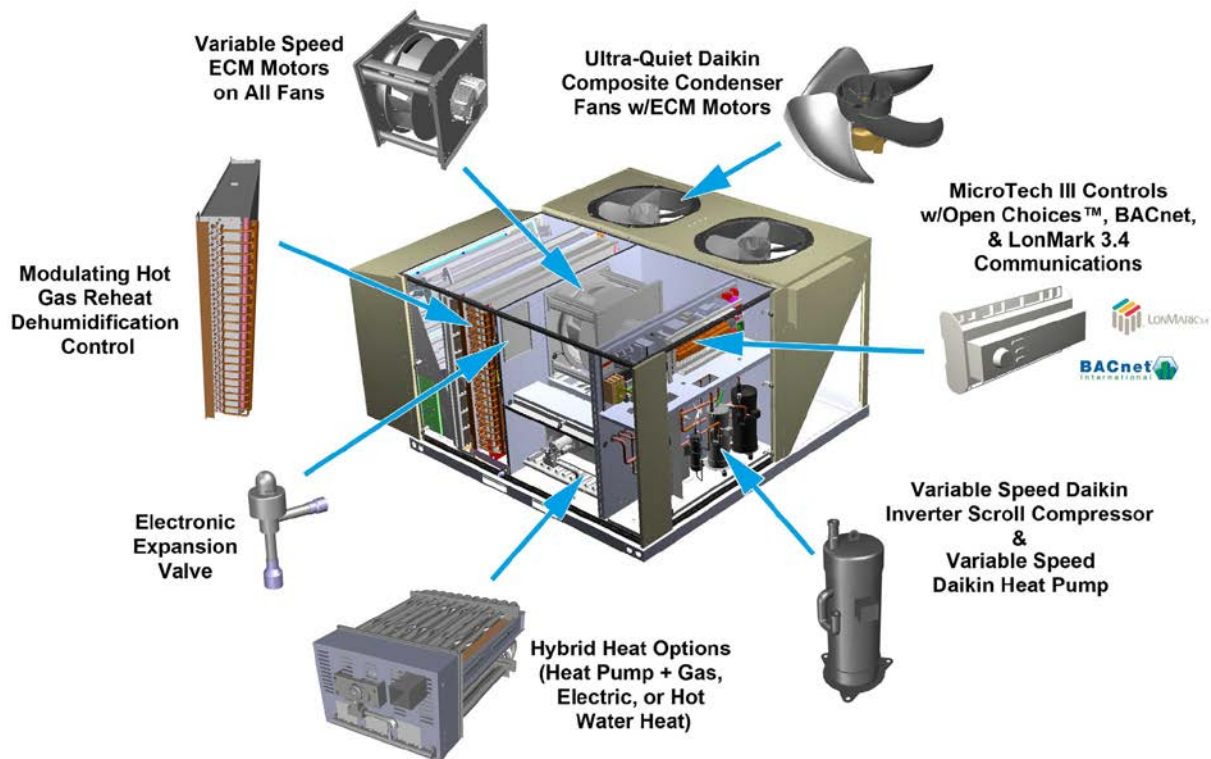


Figure 1: Daikin Rebel Unit (Source: Daikin). Reprinted with permission from Daikin

A 10-ton Rebel packaged RTU (DPS-010) from Daikin was tested at Intertek labs. More detailed technical specifications about the Rebel unit can be found from the manufacturer's website¹ (Daikin 2013). Table A - 1, Table A - 2 and Table A - 3 in the Appendix provides information on all tests conducted in the lab.

- The tests in Table A - 1 are the standard rating tests for cooling according to the ANSI/AHRI Standard 340/360 (AHRI 2007). The integrated energy efficiency ratio (IEER) calculated from tests A1-A4 can be used to verify the unit's compliance with RTU Challenge requirements (IEER \geq 18.0). Test A-1 establishes a reference for the

¹ <http://www.daikinapplied.com/rooftop-rebel.php>

full-load cooling capacity and the full-load energy efficiency to characterize the unit's part-load performance.

- The tests reported in Table A - 2 are used to correlate the unit's delivered cooling and energy efficiency as a function of temperature. This group of tests cover four outdoor dry-bulb temperatures (65°F, 80°F, 90°F, and 110°F); two indoor dry-bulb temperatures (70°F and 80°F); three indoor wet-bulb temperatures (61°F, 63°F, and 67°F); and two discharge-air temperatures (50°F and 65°F). The indoor dry-bulb and wet-bulb temperatures refer to the air entering the cooling coil. All tests listed in Table A-2 are conducted at the rated supply air flow rate.
- The tests described in Table A - 3 are used to correlate the unit's delivered cooling and energy efficiency as a function of supply air flow rate. Three flow fractions, namely, 25%, 50% and 75% of rated flow, are used in the test (100% rated air flow rate is already covered in the tests in Table A - 2). These three flow fractions are combined with three outdoor dry-bulb temperatures (65°F, 90°F, and 110°F) and two indoor wet-bulb temperatures (61°F and 67°F), leading to a total of 18 steady-state tests. For these 18 tests, the discharge-air temperature and the indoor dry-bulb temperature are fixed at 55°F and 80°F, respectively.

The following key variables were measured at a 1-minute frequency for tests: coil inlet air dry-bulb and wet-bulb temperatures, coil outlet air dry-bulb and wet-bulb temperatures, outdoor air dry-bulb temperature, supply air flow rate, supply air static pressure, compressor suction and discharge pressures, compressor suction and discharge temperatures, refrigerant flow rate, indoor fan power, compressor power, control power and RTU total power. The data also includes computed variables: total cooling capacity and the sensible cooling capacity for each minute. The measured values of each variable are averaged across each test period. The average values are then used in development of the performance curves.

Conventional packaged RTUs serving single zones normally 1) have a constant supply air flow; 2) cycle the compressor ON and OFF to meet the space thermal loads; and 3) do not maintain the discharge air set point temperature. However, the Rebel unit modulates the fan speed in response to the space thermal loads. In addition, the compressor modulates its speed to control the discharge air temperature. Thus, the conventional direct-expansion cooling model in EnergyPlus and its performance curve equations are not appropriate for capturing the Rebel's performance for the following reasons²:

- The compressor modulates its speed to control the discharge-air temperature; the compressor may not run at its total maximum capacity in most operation times. Therefore, we need to develop the performance curves to correlate the equipment delivered cooling capacity, instead of the total maximum cooling capacity, to temperatures and flow fractions.

² A variable-speed cooling coil model was added to EnergyPlus version 7.2 released in October, 2012. This model uses up to 10 discrete speed levels to emulate variable-speed compressors. Each speed needs a set of performance curves. Because Rebel's lab test was performed in May 2012, the test design, curve development and simulation of Rebel units were not targeted to use the variable-speed cooling coil model available after version 7.2.

- The discharge-air temperature is a controllable parameter and it affects the unit's delivered cooling and possibly efficiency. It is necessary to include the discharge-air temperature in the performance curves.

Several combinations of regression forms were used to identify the best form of the equation based on the coefficient of determination (R^2), the mean squared error (MSE), and the significance of each independent variable from the t-test results. The final performance curves resulting from the regression efforts are described below. For each curve, the equation form is given first. Then, the regression coefficients for that form are provided in a table. A scatter chart comparing the model-predicted value to the actual is shown.

2.1 DELIVERED COOLING MODIFIER CURVE AS A FUNCTION OF TEMPERATURE

As Equation 1 shows, this curve has three independent variables: wet-bulb temperature of the air entering the cooling coil ($T_{wb,i}$), dry-bulb temperature of the air entering the cooling coil ($T_{db,i}$), and discharge-air dry-bulb temperature ($T_{db,o}$). The output of this curve indicates the ratio of the total delivered cooling at the specific operating conditions ($T_{wb,i}$, $T_{db,i}$, $T_{db,o}$, and $ff = 1$ full flow) and the total maximum cooling capacity at the rated conditions ($T_{wb,i} = 67^\circ F$, $T_{db,i} = 80^\circ F$, $T_{c,i} = 95^\circ F$, and $ff = 1$).

$$\text{TotCapTempModFac} = C_0 + C_1(T_{wb,i})^2 + C_2(T_{db,i})^2 + C_3(T_{db,o})^2 + C_4(T_{wb,i})(T_{db,i}) + C_5(T_{wb,i})(T_{db,o}) + C_6(T_{db,i})(T_{db,o}) \quad (1)$$

Table 1 shows the regression coefficients of Equation 1, based on the experimental data from tests B1-B45 in Table A - 2. The regression equation fits the experimental data very well, with R^2 close to 1 (Figure 2).

Table 1: Regression coefficients for the delivered cooling modifier curve as a function of temperature

Coefficient	Value
C0	0.31711905
C1	0.02042291
C2	0.00190283
C3	0.00328537
C4	-0.0107672
C5	-0.0237556
C6	0.00845473

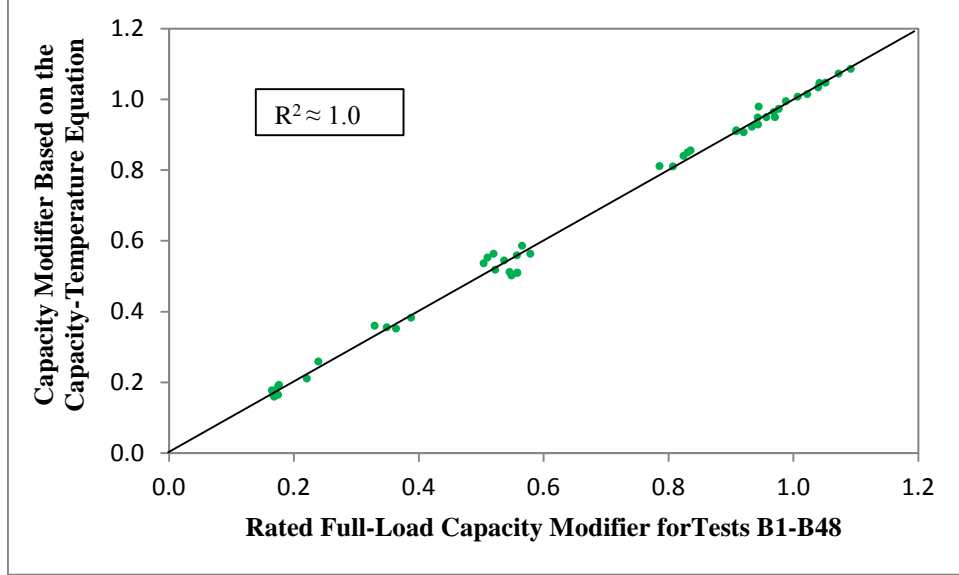


Figure 2: Comparison of predicted and measured capacity modifiers as a function of temperatures

2.2 ENERGY INPUT RATIO (EIR) MODIFIER CURVE AS A FUNCTION OF TEMPERATURE

EIR is the inverse of the coefficient of performance (COP). Equation 2 shows the EIR modifier curve as a function of temperature. This curve has four independent variables: wet-bulb temperature of the air entering the cooling coil ($T_{wb,i}$), dry-bulb temperature of the air entering the cooling coil ($T_{db,i}$), dry-bulb temperature of the air entering the air-cooled condenser coil ($T_{c,i}$), and discharge air dry-bulb temperature ($T_{db,o}$). The output of this curve indicates the ratio of the EIR at specific operating conditions ($T_{wb,i}$, $T_{db,i}$, $T_{db,o}$, $T_{c,i}$, and $ff = 1$ full flow) and the EIR at rated conditions ($T_{wb,i} = 67^{\circ}F$, $T_{db,i} = 80^{\circ}F$, $T_{c,i} = 95^{\circ}F$, and $ff = 1$).

$$\text{EIRTempModFac} = C_0 + C_1(T_{db,o}) + C_2(T_{wb,i})^2 + C_3(T_{db,i})^2 + C_4(T_{db,o})^2 + C_5(T_{c,i})^2 + C_6(T_{wb,i})(T_{db,i}) + C_7(T_{wb,i})(T_{c,i}) + C_8(T_{db,i})(T_{db,o}) + C_9(T_{db,i})(T_{c,i}) + C_{10}(T_{db,o})(T_{c,i}) \quad (2)$$

Table 2 shows the regression coefficients of Equation 2, based on the experimental data from tests B1-B45 in Table A - 2. The regression equation fits the experimental data very well, with $R^2 = 0.96$ (Figure 3).

Table 2: Regression coefficients for the EIR modifier curves a function of temperatures

Coefficient	Value
C0	-0.5966663
C1	0.24754897
C2	-0.0088454
C3	-0.0036095
C4	-0.0085282
C5	0.00072631
C6	0.01271645
C7	-0.0018991
C8	-0.0024883
C9	-0.0008663
C10	0.00250916

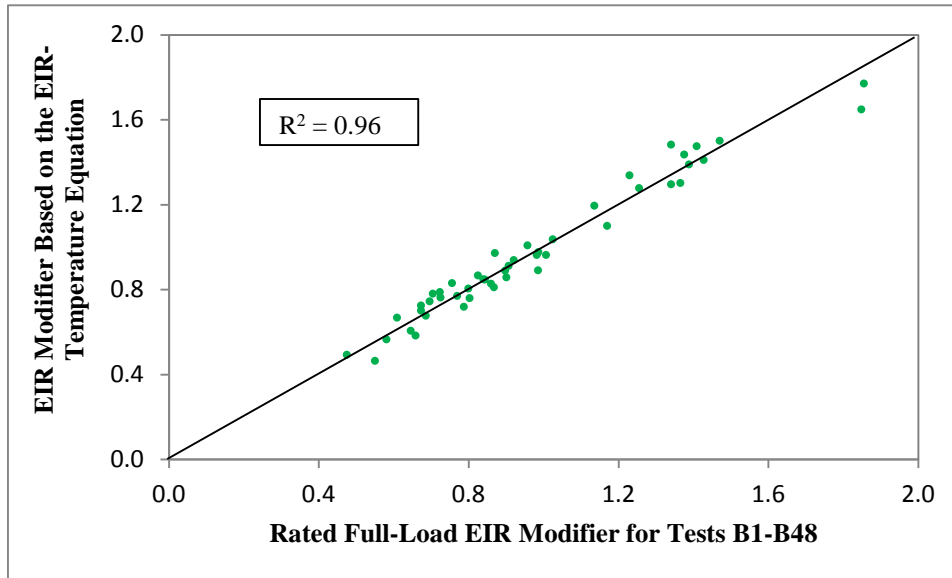


Figure 3: Comparison of predicted and measured EIR modifiers as a function of temperatures

2.3 DELIVERED COOLING MODIFIER CURVE AS A FUNCTION OF FLOW FRACTION

As Equation 3 shows, a cubic curve is used to indicate how the delivered cooling varies with the air flow fraction (ff). The output of this curve indicates the ratio of the total delivered cooling at the specific air flow fraction and the total cooling capacity at the rated air flow rate ($ff = 1$).

$$\text{TotCapFlowModFac} = C_0 + C_1(ff) + C_2(ff)^2 + C_3(ff)^3 \quad (3)$$

Table 3 shows the regression coefficients of Equation 3, based on the experimental data from tests C1-C18 in Table A-3. The regression equation fits the experimental data very well, with R^2 close to 1 (Figure 4).

Table 3: Regression coefficient for the delivered cooling modifier curve as a function of flow fraction

Coefficient	Value
C0	-0.036240442
C1	1.275963118
C2	-0.288191485
C3	0.048967033

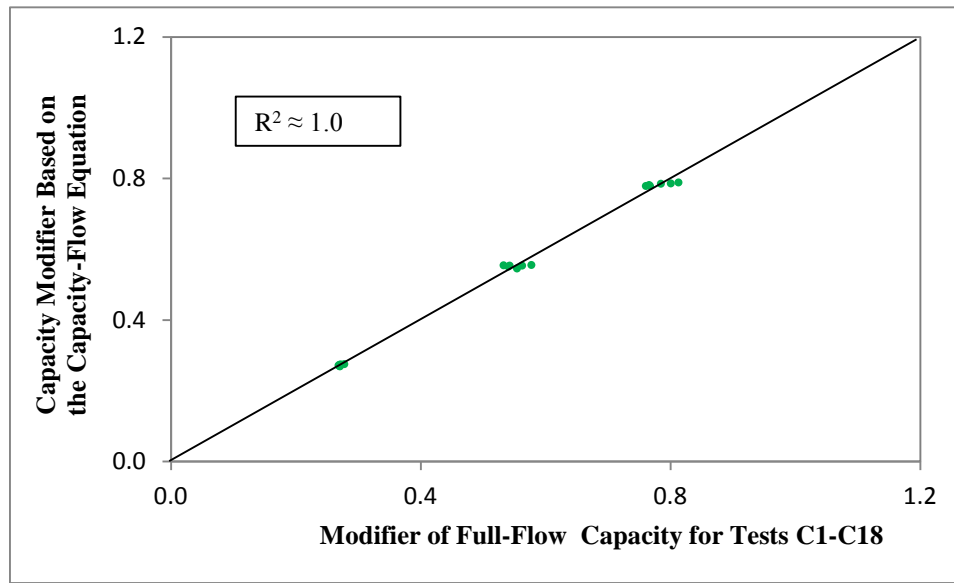


Figure 4: Comparison of predicted and measured capacity modifiers as a function of flow fraction

2.4 ENERGY INPUT RATIO (EIR) MODIFIER CURVE AS A FUNCTION OF FLOW FRACTION

A cubic curve is also used to indicate how the EIR varies with the air flow fraction (ff). The output of this curve indicates the ratio of the EIR at a specific air flow fraction and the EIR at the rated air flow rate ($ff = 1$).

$$\text{EIRFlowModFac} = C_0 + C_1(ff) + C_2(ff)^2 + C_3(ff)^3 \quad (4)$$

Initially, all data from tests C1-C18 are used to generate the regression coefficients. However, it was found that several tests (C1, C10, C11 and C16) caused large deviations between the model predictions and the experimental data. It is not clear what factors triggered the large deviations for those tests. As a result of the variation, those four points were treated as outliers and removed from the regression analysis. Table 4 shows the regression coefficients of Equation 4, after removing the above four outliers from tests C1-C18 in Table A-3. The regression equation fits the experimental data with $R^2 = 0.78$ (Figure 5).

Table 4: Regression coefficient for the EIR modifier curve as a function of flow fraction

Coefficient	Value
C0	-0.120651751
C1	9.026346984
C2	-15.86922715
C3	7.966206349

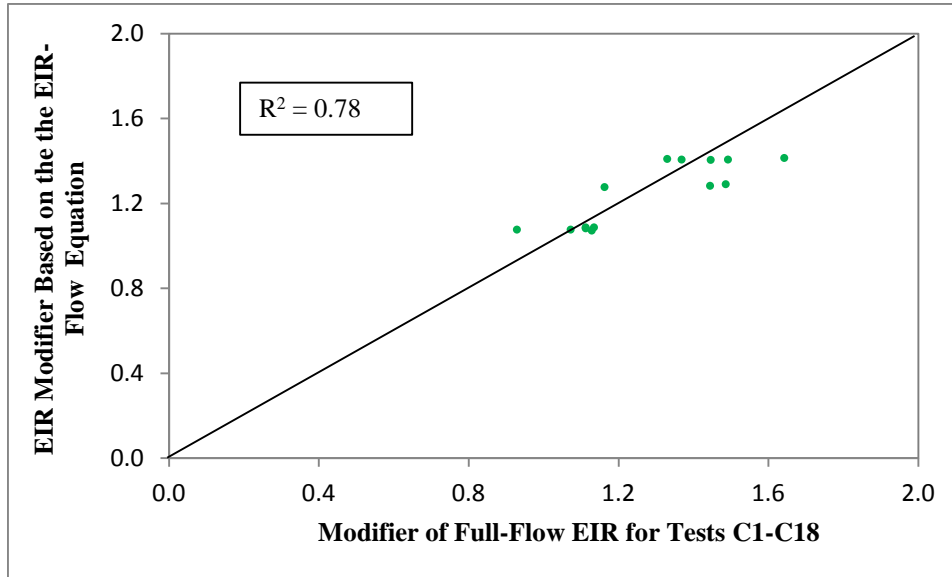


Figure 5: Comparison of predicted and measured EIR modifiers as a function of flow fraction

The following items need to be noted when using the regressed curves:

- The temperature unit is in **Celsius** instead of Fahrenheit. The Celsius unit is selected because a major motivation of this work is to generate performance curves for EnergyPlus, which uses SI units as default.
- The curves are generated from the test data with limited coverage of the independent variables. Therefore, the following boundary values are suggested: $12.8^{\circ}\text{C} \leq T_{wb,i} \leq 23.9^{\circ}\text{C}$, $18.3^{\circ}\text{C} \leq T_{db,i} \leq 29.4^{\circ}\text{C}$, $10^{\circ}\text{C} \leq T_{db,o} \leq 18.3^{\circ}\text{C}$, $21.1^{\circ}\text{C} \leq T_{c,i} \leq 46.1^{\circ}\text{C}$, and $0.3 \leq ff \leq 1$.
- The curves are generated from the test data based on a 10-ton Rebel packaged RTU (DPS-010). Although the curves were generated using data from one model, they can be reasonably expected to work for other Rebel models with two compressors as long as the control logic is the same.
- The part-load fraction correlation as a function of part-load ratio available in EnergyPlus models for packaged RTUs is not used in this work. This is simply because the use of variable-speed compressors reduces the impact of compressor cycles on efficiency losses.

3 BUILDING SIMULATION MODELS

Figure 6 shows the axonometric view of the big-box retail store modeled in this work. This building model is based on the stand-alone retail building prototype used to support ASHRAE Standard 90.1 development (Thornton et al. 2011). However, the footprint area was tripled to better represent big-box stores, which are deemed as a major market for RTU Challenge units. Thus, the modeled big-box store has a total floor area of 75,000 ft². Based on the space usage, the store is divided into five areas: front entry (0.2%), storage space (9.6%), core retail (82.4%), front retail (3.9%), and cashier area (3.9%), where the number in parenthesis indicates the percentage of that space area.

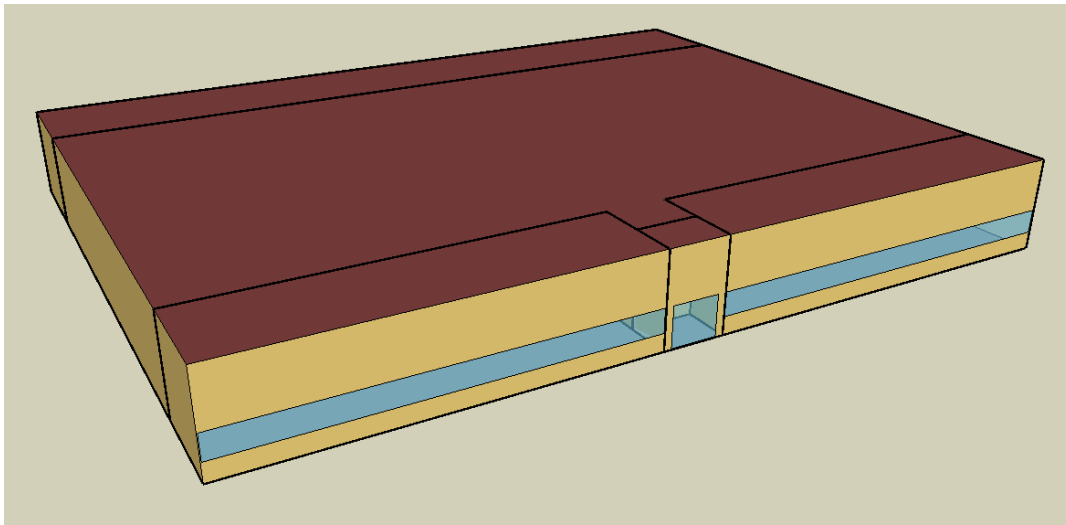


Figure 6: Axonometric view of the big-box retail store

The building model has the following opaque envelope construction elements: concrete block exterior walls, a flat roof with insulation entirely above a metal deck, and a slab-on-grade concrete floor with carpets. All exterior opaque assembly construction is configured according to Appendix A in ASHRAE Standard 90.1-2004 (ASHRAE 2004). The window construction is modeled in EnergyPlus based on the specified overall heat transfer coefficient and solar heat gain coefficient. ASHRAE Standard 90.1-2004 is followed to establish the requirements on building envelope thermal performance. The report (Thornton et al. 2011) can be referred to for more details on envelope construction and internal load profiles.

In the EnergyPlus building model, each of the five spaces is regarded as one thermal zone. Except for the front entry served by a unit heater, the other thermal zones are each equipped with a packaged unit with mechanical cooling and gas furnace heating. The packaged equipment efficiency and control strategies are discussed next.

3.1 SELECTION OF THE PACKAGED EQUIPMENT

In general, RTU Challenge-compliant units such as Rebel are considered by building owners/designers as an option for either new construction or existing building retrofits. Other options include RTUs that just meet the Federal minimum standard or other local code

requirements, such as ASHRAE Standard 90.1, or high-end RTUs that are usually available from most manufacturers' product lines. Three "references" are defined here to make the performance comparison more useful.

3.2 REFERENCE 1

Reference 1 represents existing RTUs in the field, so it can be used as the baseline to estimate the potential of energy savings from a full upgrade to the Rebel from existing equipment.

The AHRI-rated (American Heating and Refrigeration Institute) efficiency of the baseline or Reference 1 equipment performance is based on ASHRAE Standard 90.1-2004 requirement. ASHRAE 90.1-2004 specifies that for a 10-ton packaged air conditioner with gas heat, the rated energy efficiency ratio (EER) should be at least 9.9. This EER includes the supply-fan power. However, EnergyPlus models the supply fan and packaged cooling separately. Therefore, it is necessary to obtain the cooling performance excluding the supply-fan power. Deru et al. (2011) used the following equation to derive packaged equipment's coefficient of performance (COP) suitable for EnergyPlus modeling from the AHRI-rated EER:

$$COP = \left(\frac{EER}{3.413} + R \right) / (1 - R) \quad (5)$$

where R is the ratio of supply-fan power to total equipment power at the rated condition. Deru et al. (2011) used an R value of 0.12, which is a reasonable value that represents a broad class of products. Based on the laboratory test data from several products, PNNL found that using the above equation with R equal to 0.12 overestimates the COP by between 5% and 10%. Therefore, for the current work, the COP estimated from Equation 1 was adjusted.

Furthermore, the existing unit's actual performance in the field degrades with time. Therefore, it is important to apply a degradation factor to consider the actual field performance for existing RTUs. The degree of performance degradation varies with many factors such as the number of years operated in the field, number of hours of operations, weather conditions, and the level of maintenance. An arbitrary, 10% performance degradation factor was used for this work. Thus, in the EnergyPlus model, the unit's COP (excluding supply-fan energy consumption) is calculated as:

$$COP = \left(\frac{\frac{9.9}{3.413} + 0.12}{1 - 0.12} \right) * 0.90 = 2.94 \quad (6)$$

In the above equation, the number 1.05 represents the adjustment factor to correct the COP overestimation; the number 0.90 represents the adjustment factor to account for the performance degradation in the field.

To estimate the annual energy consumption of the prototypical building, the RTU's part-load performance is also needed. However, because of the lack of availability of part-load performance data of existing RTUs in the field, a commercial RTU's performance data at different operating conditions were used to develop the performance curves. Because the performance curves are normalized, using the manufacturer's data to develop the curves is

reasonable. The data used to develop the curves for Reference 1 equipment were provided by a manufacturer, and they represent an equipment from the low-end of their product line. These curves take the standard equation forms as used in EnergyPlus. The temperature curves are presented below in Equations 7 and 8, while the flow fraction curves take the same form as those in Equations 3 and 4. Table 5 lists the curve coefficients used by the Reference 1 model.

Total cooling capacity modifier curve as a function of temperature:

$$\text{TotCapTempModFac} = C_0 + C_1 (T_{wb,i}) + C_2 (T_{wb,i})^2 + C_3 (T_{c,i}) + C_4 (T_{c,i})^2 + C_5 (T_{wb,i})(T_{c,i}) \quad (7)$$

Energy input ratio (EIR) modifier curve as a function of temperatures:

$$\text{EIRTempModFac} = C_0 + C_1 (T_{wb,i}) + C_2 (T_{wb,i})^2 + C_3 (T_{c,i}) + C_4 (T_{c,i})^2 + C_5 (T_{wb,i})(T_{c,i}) \quad (8)$$

Table 5: Curve coefficients for the single-stage direct expansion cooling coil in Reference 1 equipment

Coefficient	TotCapTempModFac	ETRTempModFac	TotCapFlowModFac	ETRFlowModFac
C0	0.7503	0.4152	1.0	1.0
C1	0.0161	0.0093	0.0	0.0
C2	0.0008	0.0002	0.0	0.0
C3	-0.0036	0.0150	0.0	0.0
C4	-0.0002	0.0008	—	—
C5	0.0000	-0.0018	—	—

In addition to the rated performance, RTUs compatible with Reference 1 have the following features:

- Constant-speed supply fan. When the RTU operates, its supply fan always runs at full speed (100%) regardless of the operational modes (e.g., heating, cooling and ventilation).
- Single-stage cooling. The direct expansion (DX) cooling has one constant-speed compressor cycling on and off to meet the space cooling loads.
- Gas furnace. The gas furnace cycles on and off to meet the space heating loads.
- Integrated air-side economizer. The economizer control is modeled as differential dry-bulb temperature even though other control types such as fixed dry-bulb or enthalpy may be more common in the field or more suitable for some locations. Because economizer control is not Rebel’s unique feature distinguishable from conventional RTUs, the same economizer control type is used for all three references and the Rebel unit.
- No demand-controlled ventilation (DCV). DCV is not Rebel’s unique feature distinguishable from conventional RTUs. Therefore, DCV is not modeled for all three references and the Rebel unit to make reasonable comparisons.

3.3 REFERENCE 2

Reference 2 represents RTUs in the market that just meet the current (2013) Federal regulations for commercial equipment standards. So, it can be used as the baseline to estimate the potential of energy savings from Rebel units in comparison with new RTUs that meet the minimum efficiency requirements. Except for the rated full-load efficiency, RTUs compatible with Reference 2 are the same as Reference 1 and they have the following features:

- Current Federal minimum standard requires that a 10-ton packaged air conditioner with gas heat have a minimum EER of 11 at rated conditions. Based on Equation 1 and using the overestimation factor of 1.05, the unit's COP excluding the supply-fan energy consumption is calculated at 3.62.
- The same performance curves developed for Reference 1 are used to model the unit's part-load performance.
- The unit has a constant-speed supply fan, as defined for Reference 1.
- The unit has single-stage DX cooling and cycles on and off to meet space cooling loads, as defined for Reference 1.
- The gas furnace cycles on and off to meet space heating loads, as defined for Reference 1.
- Integrated air-side economizer is used, as defined for Reference 1.
- No DCV is used, as explained for Reference 1.

3.4 REFERENCE 3

Reference 3 represents the latest ASHRAE 90.1-2010 requirements. For RTUs with cooling capacity greater than 11,000 Btu/h, ASHRAE 90.1-2010 (ASHRAE 2010) requires two-speed fan control or variable-speed fan control. RTUs compatible with Reference 3 have the following features:

- ASHRAE requires that a 10-ton packaged air conditioner with or without gas heat has a minimum EER of 11 at rated conditions, the same as the Federal minimum standard. Based on Equation 1 and using an overestimation factor of 1.05, the unit's COP excluding the supply-fan consumption is 3.62. This COP is the efficiency at the rated conditions. A unit with fan-speed control normally has cooling capacity control to avoid coil freezing. Reference 3 models two-stage cooling, so separate sets of performance data for the two stages are needed. Because ASHRAE 90.1-2010 does not specify the 1st stage cooling efficiency, the rated COP is derived by referring to the efficiency ratio between two stages for a commercial RTU. Based on the same product used for Reference 1, the 2nd (or full) stage cooling is about 24% higher efficiency than the 1st stage cooling. Therefore, in this work, the modeled COP for 1st stage cooling is $3.62/1.24 = 2.92$.
- The curves developed for Reference 1 are used to model the unit's 2nd stage part-load performance. A new set of curves are developed for the 1st stage cooling based on the same product as used for Reference 1, but only the 1st stage cooling performance data are used. Table 6 lists the curve coefficients for the 1st stage cooling.

- Two-speed fan control is used. The supply fan has two speeds with the low speed at two thirds of the full speed. The low fan speed is used for ventilation and 1st stage DX cooling.
- Two-stage DX-cooling with equally sized constant speed compressors is used. The switch from low stage to high stage can be based either the space temperature deviation from the cooling set point or time lag for not reaching the cooling set point. Time lag is modeled in this work as explained later with the control sequence.
- Gas furnace cycles on and off, as defined for Reference 1.
- Integrated air-side economizer is used, as defined for Reference 1.
- No DCV is used, as explained for Reference 1.

Table 6: Curve coefficients for the 1st stage DX cooling coil in Reference 3 model

Coefficient	TotCapTempMod Fac	ETRTempMod Fac	TotCapFlowMod Fac	ETRFlowMod Fac
C0	0.8499	0.6649	1.0	1.0
C1	0.0176	-0.0263	0.0	0.0
C2	0.0010	0.0019	0.0	0.0
C3	-0.0088	0.0204	0.0	0.0
C4	-0.0002	0.0011	—	—
C5	0.0000	-0.0028	—	—

3.5 RTU CHALLENGE UNIT: DAIKIN REBEL

In comparison with the above three reference units, a Rebel unit has the following features:

- A 10-ton Rebel unit has the rated cooling capacity of 123,000 Btu/h and the rated EER of 12.6. Based on the laboratory test data, the supply-fan power is approximately 1,000 W at rated conditions. Based on the above information, the rated COP excluding the supply-fan power is calculated to be 4.11.
- Customized performance curves developed from our previous work are used (described previously in the report - Performance Curve Development).
- Fan speed varies based on the space temperature using a proportional-integral (PI) control loop (Daikin 2013). Because of the use of ECM and airfoil fans, the simulation model for Rebel units uses a higher fan total efficiency (63% vs. 57%) than the conventional units in the three reference models.
- When DX cooling is needed, the variable-speed compressor modulates to maintain the discharge-air temperature set point (at a value chosen by the owner). For this analysis, a 55°F discharge-air set point was used.
- When heating is needed, the gas furnace modulates its burner capacity to maintain the discharge-air temperature set point at 85°F. For the Rebel unit with a gas furnace, the discharge-air temperature can be reset as high as 140°F. However, for this analysis the temperature was held constant at 85°F.

- Integrated air-side economizer is used, as defined for Reference 1.
- No DCV is used, as explained for Reference 1.

4 ENERGY MANAGEMENT SYSTEM (EMS) IMPLEMENTATION

The traditional controls built into EnergyPlus are not capable of modeling many control options required to simulate the Rebel unit, for example, control of supply-fan speed based on the deviation of the space temperature compared to the space set point. To model the sequence of operation of the Rebel unit requires specialized control. The EMS feature in EnergyPlus is used to customize the sequence of operations. The EMS provides a variety of sensors and actuators much like an actual building automation system. The sequence of operations embedded in the EnergyPlus input files is used to override the traditional EnergyPlus control and to add the desired control functionality into the simulation.

4.1 DESCRIPTION OF RTU'S OPERATIONAL MODE

Depending on the space temperature T , the RTU has four basic operation modes: idle, ventilation, heating, and cooling.

Idle mode. The RTU is in the idle mode if 1) the space temperature lies between the heating and cooling set points, and 2) the space is unoccupied. In the idle mode, the fan, the heating and the cooling are all off. Note that the heating and cooling set points during occupied and unoccupied modes may be different.

Ventilation mode. The RTU operates in the ventilation mode if 1) the space temperature lies between the heating and cooling set points, and 2) the space is occupied.

Heating mode. The RTU operates in the heating mode if the space temperature is less than the heating set point. Once heating is initiated, it continues until the space temperature rises above the heating set point plus a differential (e.g., 1.8°F).

Cooling mode. The RTU operates in the cooling mode if the space temperature is greater than the cooling set point. Depending on the economizer and mechanical cooling stage control status, the RTU may work in two different sub-modes: economizing or mechanical cooling.

- Economizing (outdoor air cooling) mode. Right after the cooling mode is initiated, the system controller determines whether the outdoor air (OA) is suitable for free cooling. If the outdoor air is favorable for cooling, the economizing mode is used. This mode continues as long as the space temperature lies between the cooling set point and the cooling set point minus the differential. Otherwise, if the space temperature decreases below the cooling set point minus the differential, the economizer mode ends and the outdoor-air damper returns to the ventilation-only position. The controller initiates mechanical cooling if the following conditions are met: 1) the space temperature is above the cooling set point; 2) the time in economizing mode exceeds the inter-stage delay time limit (e.g., 5 minutes); and 3) the discharge-air temperature is greater than the threshold (e.g., 58°F).
- Mechanical cooling mode. For single-stage cooling, the rooftop unit runs at its full cooling capacity until the space temperature is cooled to the cooling set point minus the differential. For two-stage cooling, the unit runs stage 1 cooling first. Stage 1 cooling continues as long as the space temperature lies between the cooling set point and the

cooling set point minus the differential. Otherwise, the mechanical cooling ends when the space temperature decreases below the cooling set point minus the differential. The 2nd stage mechanical cooling is initiated if the following conditions are met: 1) the space temperature is above the cooling set point; 2) the time in stage 1 cooling mode exceeds the inter-stage delay time limit (e.g., 5 minutes); and 3) the discharge-air temperature is greater than the threshold (e.g., 58°F). Once initiated, the 2nd stage cooling continues until the space temperature decreases below the cooling set point minus the differential.

4.2 DEFINITION OF RTU'S OPERATION IN DIFFERENT MODES

In the ventilation mode, both heating and cooling are off. The OA damper is at its minimum position to provide the design ventilation. The supply fan runs at 100% for the Reference 1 unit, at 67% for the Reference 2 unit, and 40% for the Rebel model.

In the heating mode, heating is on and cooling is off. The OA damper is at its minimum position to provide the design ventilation. The gas furnace runs at its maximum capacity for both reference models while it modulates the burner output to maintain the discharge-air temperature at 85°F. The supply fan runs at 100% for all three reference units, while it modulates based on the space temperature using a proportional-integral (PI) control loop for the Rebel model. The control parameters include Gain, Period (ST), and Project Ahead Time (PAT). In the example file, these control parameters are set as follows:

- Gain = 1.5,
- ST = 60 seconds, and
- PAT = 120 seconds.

The PI loop is implemented in the following manner:

- The space temperature is trended. The space temperature at the previous time step and at two time steps prior is denoted as T₁ and T₂, respectively. The projected space temperature at the current time step is calculated as:

$$T_{prj} = T_1 + \frac{T_1 - T_2}{ST} * (ST + PAT) \quad (9)$$

- Then, the change of fan speed (i.e., the supply-air flow rate in the simulation model) at the current time step is given as:

$$\Delta\text{Speed} = \text{Gain} * [(T_{HSP} + \delta) - T_{prj}] \quad (10)$$

where T_{HSP} is the heating set point and δ is the temperature differential.

- The calculated fan speed change needs to be checked to make sure it is no more than the maximum allowed change. In addition, the final adjusted fan speed needs to be checked to make sure it lies between the minimum and maximum fan speed for heating.

In the economizing (OA cooling) mode, both heating and mechanical cooling are off. The OA damper is modulated to maintain the discharge-air temperature at the economizer discharge-air set point (e.g., 58°F). The supply fan runs at 100% for both reference models. For the Rebel model, the fan speed modulates based on the space temperature using a similar PI loop as used for heating.

In the mechanical cooling mode, the compressor is on for cooling and off during heating. The OA damper is fully open if the OA condition is favorable for cooling; otherwise, the OA damper is at the minimum position required to maintain ventilation. For the Reference 1 unit, the supply fan runs at 100%. For the Reference 2 unit, the supply fan runs at 67% and 100%, respectively for the 1st and 2nd stage cooling. For the Rebel unit, the fan speed modulates based on the space temperature using a similar PI loop as used for heating.

Appendix B of this report has an example file illustrating the use of EMS for Rebel control. The procedure to apply the EMS code developed for this work to a specific model is also provided in Appendix B.

5 ENERGYPLUS SIMULATION RESULTS

The EnergyPlus simulation models were run for three locations: Houston, Los Angeles, and Chicago. Both HVAC energy savings and cost savings are presented in this section.

5.1 ENERGY SAVINGS

Table 7 shows the energy end uses including cooling energy (kWh), fan energy (kWh), heating energy (Therm), total RTU electricity (kWh), and total RTU energy (MMBtu). Figure 7 shows the RTU electricity savings is determined for the three locations from using the Rebel units in comparison with the three reference units for the whole building. Figure 8 shows the total RTU energy savings (including both electricity savings and natural gas penalties). Both figures have two parts: the top part shows the percentage of savings and the bottom part shows the absolute savings.

Table 7: Simulated annual HVAC energy end uses for the modeled retail building

Location	RTU Type	Cooling (kWh)	Fan (kWh)	Heating (Therm)	Total RTU Electricity (kWh)	Total RTU Energy (MMBtu)
Houston	Reference 1	317,747	152,733	3,197	470,481	1,925
	Reference 2	258,378	152,733	3,197	411,111	1,722
	Reference 3	265,494	84,325	3,741	349,819	1,568
	Rebel	202,764	43,136	3,733	245,900	1,212
Los Angeles	Reference 1	99,511	131,683	1,342	231,194	923
	Reference 2	80,917	131,683	1,342	212,600	860
	Reference 3	79,242	72,933	1,718	152,175	691
	Rebel	74,006	48,586	1,709	122,592	589
Chicago	Reference 1	116,156	147,956	11,763	264,111	2,077
	Reference 2	94,453	147,956	11,763	242,408	2,003
	Reference 3	94,128	75,297	13,263	169,425	1,904
	Rebel	79,244	42,778	13,405	122,022	1,757

Table 7, Figure 7 and Figure 8 lead to the following conclusions:

- Using Reference 1 as the baseline, Rebel units consume 37% less HVAC energy (including electricity for fan and cooling and natural gas for heating) in Houston, 36% in Los Angeles, and 15% in Chicago. Rebel units save about 50% RTU electricity consumption in all three locations. The electricity savings come from the higher cooling efficiency of Rebel units and the use of an energy-efficient supply fan with variable-speed control. Table 7 shows that Rebel’s variable-speed supply fan contributes to a significant amount of HVAC energy savings. The fan energy savings are only slightly less than cooling energy savings in Houston, but are much greater than cooling energy savings in the other two locations. Because the supply-fan energy was simulated as heat

gain to the supply-air stream, reducing fan energy also leads to an increase in heating energy consumption.

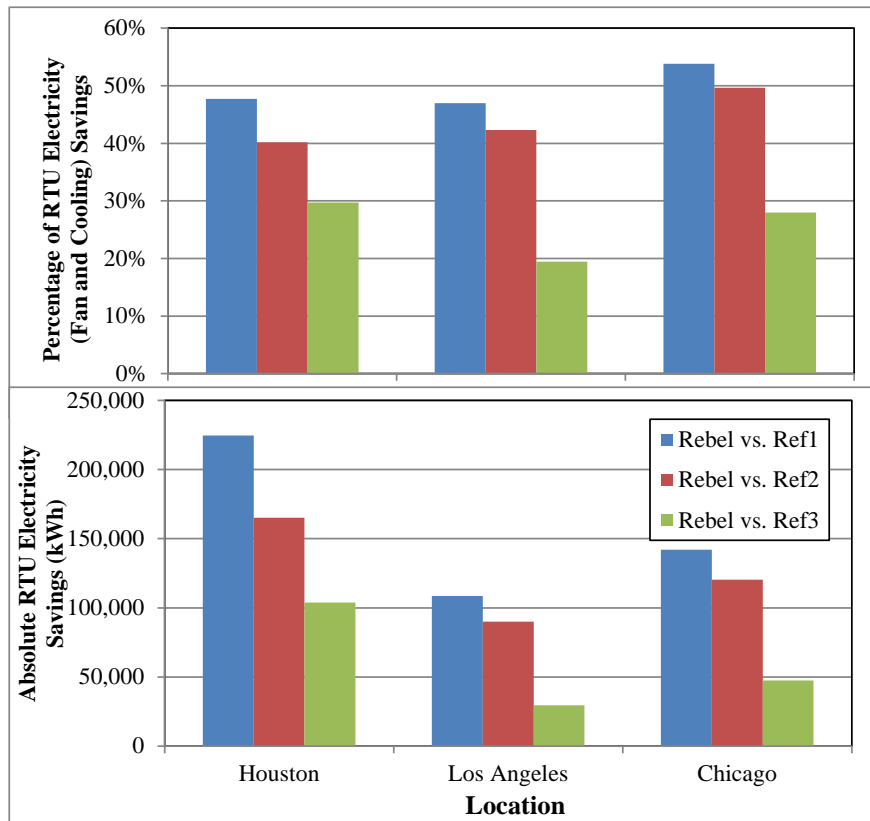


Figure 7: Annual RTU electricity savings from the use of Rebel units compared to the three reference units for the modeled retail building

- Using Reference 2 as the baseline, Rebel unit consumes 30% less HVAC energy in Houston, 32% less in Los Angeles, and 12% less in Chicago. Rebel units save about 40% RTU electricity consumption in Houston and Los Angeles, and about 50% in Chicago. Because both Reference 2 and Reference 1 have constant-speed supply fans, the fan savings remain the same after the baseline changes from Reference 1 to Reference 2.
- Using Reference 3 as the baseline, Rebel unit consumes 23% less HVAC energy in Houston, 15% less in Los Angeles, and 8% less in Chicago. Rebel units save about 30% RTU electricity consumption in Houston, 19% in Los Angeles, and 28% in Chicago.

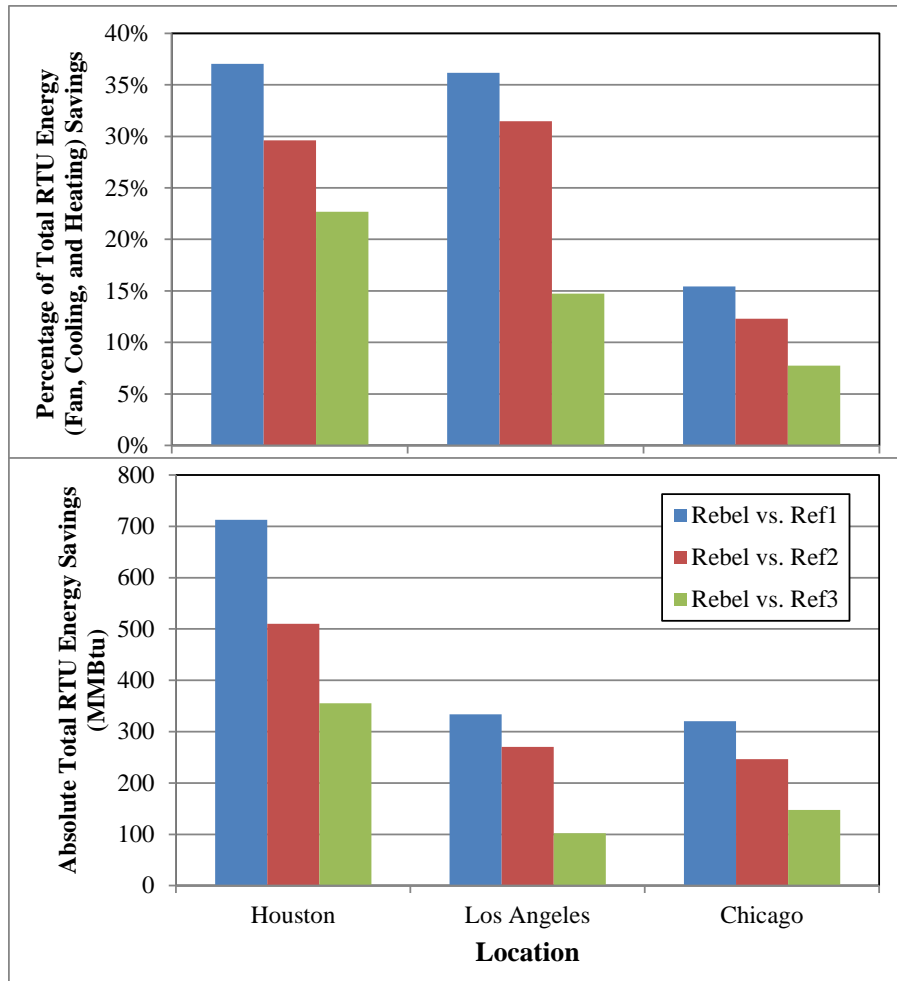


Figure 8: Annual RTU total energy savings from the use of Rebel units compared to the three reference units for the modeled retail building

5.2 COST SAVINGS

Average blended gas and electricity prices from EIA (2013) are used for the analysis. Table 8 provides the 2012 electricity and natural gas prices for the three locations. Based on these prices and the energy simulation results from Table 7, energy costs are calculated as shown in Table 9. Figure 9 shows the RTU electricity cost savings in the three locations from using the Rebel units in comparison with the three reference units for the whole building. Figure 10 shows the total RTU energy cost savings (including both electricity cost savings and natural gas cost penalties).

Table 8: Electricity and natural gas prices by locations

Location	Electricity (\$/kWh)	Natural Gas (\$/Therm)
Houston	0.081	0.707
Los Angeles	0.126	0.713
Chicago	0.079	0.779

Table 9: Annual HVAC energy costs of the modeled retail building

Location	RTU Type	Cooling Energy Cost (\$)	Fan Energy Cost (\$)	Heating Energy Cost (\$)	Total RTU Electricity Cost (\$)	Total RTU Energy Cost (\$)
Houston	Reference 1	25,738	12,371	2,260	38,109	40,369
	Reference 2	20,929	12,371	2,260	33,300	35,560
	Reference 3	21,505	6,830	2,645	28,335	30,981
	Rebel	16,424	3,494	2,639	19,918	22,557
Los Angeles	Reference 1	12,538	16,592	957	29,131	30,087
	Reference 2	10,196	16,592	957	26,788	27,744
	Reference 3	9,984	9,190	1,225	19,174	20,399
	Rebel	9,325	6,122	1,218	15,447	16,665
Chicago	Reference 1	9,176	11,688	9,163	20,865	30,028
	Reference 2	7,462	11,688	9,163	19,150	28,313
	Reference 3	7,436	5,948	10,332	13,385	23,717
	Rebel	6,260	3,379	10,442	9,640	20,082

Table 9, Figure 9 and Figure 10 lead to the following conclusions:

- Using Reference 1 as the baseline, Rebel units lead to about 45% lower HVAC energy cost in Houston and Los Angeles, and 33% lower cost in Chicago. The percentage savings of electricity cost is around 50% for all three locations.
- Using Reference 2 as the baseline, Rebel units lead to about 37% lower HVAC energy cost in Houston, 40% lower cost in Los Angeles, and 29% lower cost in Chicago. The percentage savings of electricity cost is 40%, 42%, and 50%, respectively in the above three locations.
- Using Reference 3 as the baseline, Rebel units lead to about 27% lower HVAC energy cost in Houston, 18% lower cost in Los Angeles, and 15% lower cost in Chicago. The percentage savings of electricity cost is 30%, 19%, and 28%, respectively in the above three locations.

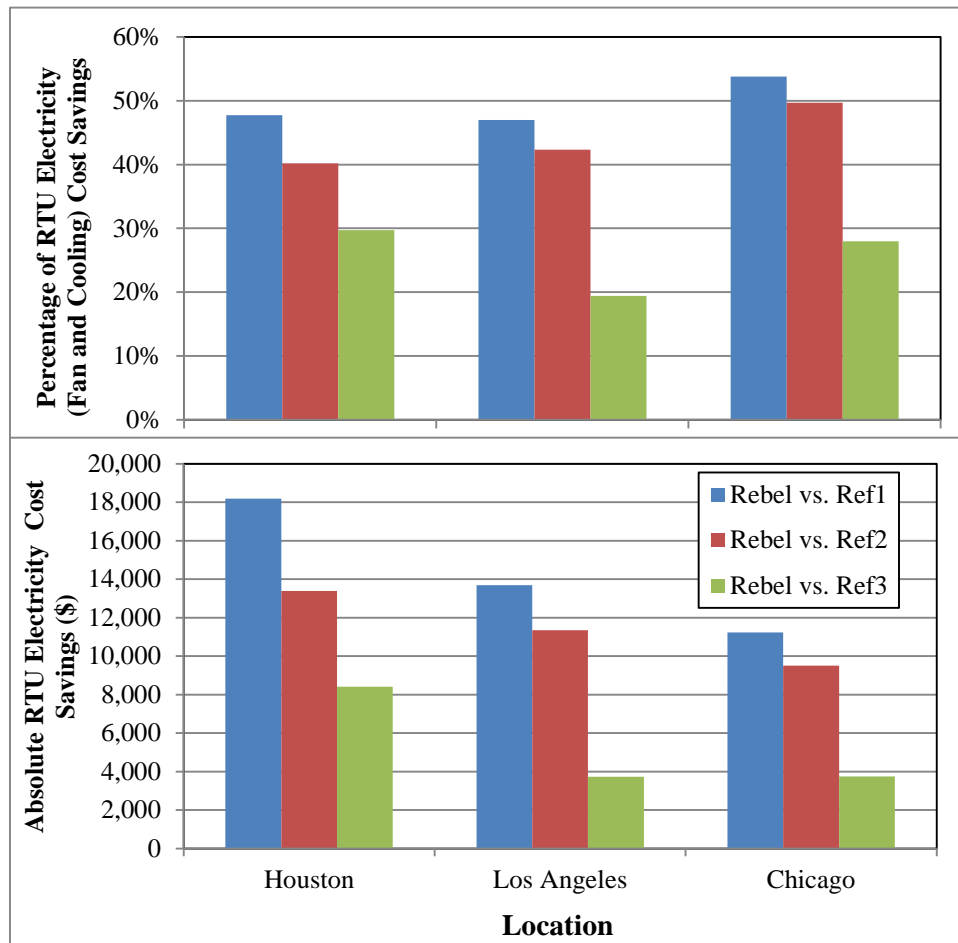


Figure 9: Annual RTU electricity cost savings from the use of Rebel units compared to the three reference units for the modeled retail building

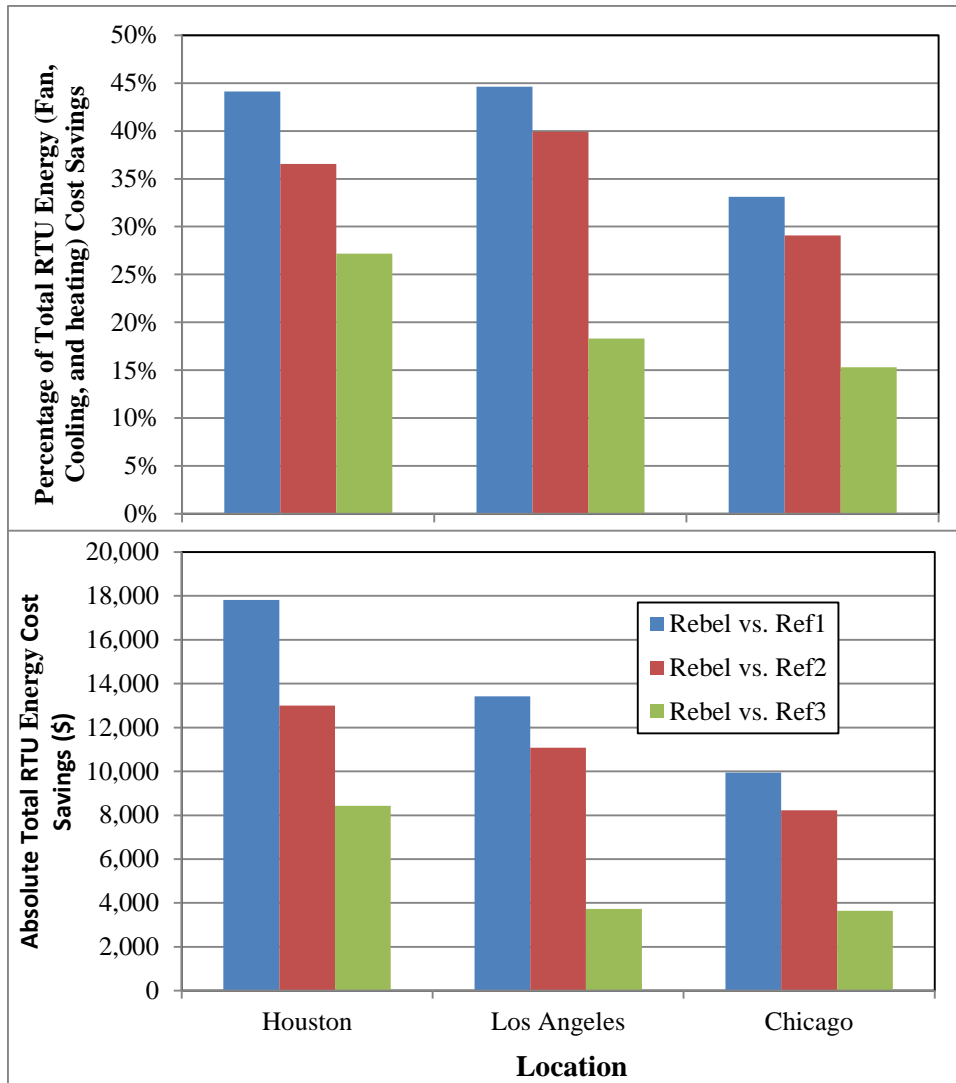


Figure 10: Annual RTU total energy cost savings from the use of Rebel units compared to the three reference units for the modeled retail building

In summary, Table 10 provides a summary of the energy and cost savings in both relative and absolute terms. For natural gas savings and electricity savings, there is only one fuel type involved in the calculation of percentage savings. Therefore, the percentage savings are the same for energy and cost. However, because there are mixed fuel types in total savings, the percentage savings needs to be distinguished between energy and cost, as shown in the table.

Table 10: Summary of annual energy and cost savings for the modeled retail building

Location	Natural Gas Savings			Electricity Savings			Total Savings			
	%	Therm	\$	%	kWh	\$	% Energy	% Cost	MMBtu	\$
Rebel (EER= 12.6, IEER* = 18.1) vs. Reference 1 (EER= 8.9, IEER = 12.8)										
Houston	-17	-536	-379	48	224,581	18,191	37	44	713	17,812
Los Angeles	-27	-367	-261	47	108,603	13,684	36	45	334	13,423
Chicago	-14	-1,642	-1,279	54	142,089	11,225	15	33	321	9,946
Rebel (EER= 12.6, IEER = 18.1) vs. Reference 2 (EER= 11.0, IEER = 12.8)										
Houston	-17	-536	-379	40	165,211	13,382	30	37	510	13,003
Los Angeles	-27	-367	-261	42	90,008	11,341	32	40	270	11,080
Chicago	-14	-1,642	-1,279	50	120,386	9,511	12	29	247	8,231
Rebel (EER= 12.6, IEER = 18.1) vs. Reference 3 (EER= 11.0, IEER = 12.8)										
Houston	0	9	6	30	103,919	8,417	23	27	355	8,424
Los Angeles	1	9	7	19	29,583	3,728	15	18	102	3,734
Chicago	-1	-141	-110	28	47,403	3,745	8	15	148	3,635

*IEER = Integrated Energy Efficiency Ratio

6 DISCUSSION

The building model as presented in Section 3 (Building Simulation Models) has four thermal zones, each served by a packaged RTU. Although autosized RTU capacities may vary with different locations and thermal zones, the rated full-load efficiency and normalized part-load performance curves representing a 10-ton cooling capacity are used for all models. This simplification is expected to have negligible impact on performance comparisons.

In this performance assessment of Rebel units, efforts were focused on those features that lead to cooling and fan energy savings. Gas furnace performance including both rated full-load and part-load efficiency was kept the same across all compared RTUs. As a result, the savings will be underestimated if Rebel improves heating efficiency relative to conventional air conditioners.

The Reference 1 model uses the performance curves developed for new equipment from a manufacturer's low-end product line. Existing RTUs in the field may have degraded part-load performance as well. Therefore, the energy and cost savings are likely underestimated when Reference 1 is used to approximate existing RTUs. Similarly, the product, upon which the performance curves were developed, has an IEER of 11.8, which is higher than the ASHRAE 90.1-2010 minimum IEER requirement of 11.0. Higher savings are expected if performance curves in References 2 and 3 closely match the minimum codes and standards requirements.

The simulation models did not capture the energy efficiency loss caused by compressor cycling. In comparison with conventional units, Rebel is capable of modulating cooling capacity using variable-speed compressors. More cooling savings could be obtained if the efficiency loss from compressor cycling is considered in the reference models.

Based on the simulation results, the RTU Challenge unit, if widely adopted, could lead to significant energy, cost and emission reductions. Because the cost of these units was not available and because the costs would be specific to a given installation, no attempt was made to estimate the potential payback periods associated with any of the three reference scenarios. However, if the incremental cost for any of the three reference cases is known, one can easily estimate a simple payback period.

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APPENDIX A: LABORATORY TEST MATRICES

Table A - 1: Test matrix used to characterize full-load performance

Test No	Test Conditions ¹	OUTDOOR SECTION: Air Entering (Condenser Coil)		Return Air		OA Ratio	INDOOR SECTION : Air Entering (Evaporator Coil)	
		Dry-Bulb	Wet-Bulb	Dry-Bulb	Wet-Bulb		Dry-Bulb	Wet-Bulb
A1	IEER 100% ^{2,3}	95°F	75°F	80°F	67°F	0%	80°F	67°F
A2	IEER 75% ^{2,4}	81.5°F	66.3°F					
A3	IEER 50% ^{2,4}	68.0°F	57.5°F					
A4	IEER 25% ^{2,4}	65°F	52.8°F					
A5	Low Temp Cooling Condition ^{2,4}	67°F	57°F	NA	NA	100%	67°F	57°F
A6	Max Operating Condition ^{2,4}	115°F	75°F	80°F	67°F	0%	80°F	67°F
A7	Condensate Disposal ^{2,4}	80°F	75°F	NA	NA	100%	80°F	75°F
A8	Hot-Dry Annual Average Condition ^{3,4,5}	90°F	64°F	78°F	64°F	30%	82°F	64°F
A9	Hot-Humid Annual Average Condition ^{3,4,5}	80°F	77°F	80°F	67°F	30%	80°F	70°F

Notes

1. All tests shall operate the unit against 0.7 H₂O of external static
2. Based on ANSI/AHRI Standard 340/360 2007
3. Indoor-coil air flow rate not to exceed 37.5 SCFM per 1000 Btu/h of rated capacity
4. Air flow rate should be adjusted to maintain or try to maintain the "IEER 100%" test condition measured leaving air dry-bulb temperature
5. Operate all refrigeration circuits at maximum capacity

OA – Outdoor Air

IEER – Integrated Energy Efficiency Ratio

Table A - 2: Test matrix used to develop part-load performance curves as a function of temperature

Test	Outdoor Dry-Bulb Temperature (°F)	Indoor Dry-Bulb Temperature (°F)	Indoor Wet-Bulb Temperature (°F)	Discharge Air Temperature (°F)
B1	65	70	61	50
B2	80	70	61	50
B3	90	70	61	50
B4	110	70	61	50
B5	65	70	63	50
B6	80	70	63	50
B7	90	70	63	50
B8	110	70	63	50
B9	65	70	67	50
B10	80	70	67	50
B11	90	70	67	50
B12	110	70	67	50
B13	65	80	61	50
B14	80	80	61	50
B15	90	80	61	50
B16	110	80	61	50
B17	65	80	63	50
B18	80	80	63	50
B19	90	80	63	50
B20	110	80	63	50
B21	65	80	67	50
B22	80	80	67	50
B23	90	80	67	50
B24	110	80	67	50
B25	65	70	61	65
B26	80	70	61	65
B27	90	70	61	65
B28	110	70	61	65
B29	65	70	63	65
B30	80	70	63	65
B31	90	70	63	65
B32	110	70	63	65
B33	65	70	67	65
B34	80	70	67	65
B35	90	70	67	65
B36	110	70	67	65
B37	65	80	61	65
B38	80	80	61	65

Test	Outdoor Dry-Bulb Temperature (°F)	Indoor Dry-Bulb Temperature (°F)	Indoor Wet-Bulb Temperature (°F)	Discharge Air Temperature (°F)
B39	90	80	61	65
B40	110	80	61	65
B41	65	80	63	65
B42	80	80	63	65
B43	90	80	63	65
B44	110	80	63	65
B45	65	80	67	65
B46	80	80	67	65
B47	90	80	67	65
B48	110	80	67	65

Table A - 3: Test matrix used to develop part-load performance curves as a function of flow fraction

Test	Discharge Air Temperature (°F)	Outdoor Dry-Bulb Temperature (°F)	Indoor Dry-Bulb Temperature (°F)	Indoor Wet-Bulb Temperature (°F)	Flow Fraction (% of Rated Flow)
C1	55	65	80	61	25
C2	55	65	80	61	50
C3	55	65	80	61	75
C4	55	90	80	61	25
C5	55	90	80	61	50
C6	55	90	80	61	75
C7	55	110	80	61	25
C8	55	110	80	61	50
C9	55	110	80	61	75
C10	55	65	80	67	25
C11	55	65	80	67	50
C12	55	65	80	67	75
C13	55	90	80	67	25
C14	55	90	80	67	50
C15	55	90	80	67	75
C16	55	110	80	67	25
C17	55	110	80	67	50
C18	55	110	80	67	75

APPENDIX B: EXAMPLE ENERGYPLUS FILE FOR MODELING REBEL UNITS



Example File

Major steps to deploy the EMS code for a specific model:

1. Make sure each packaged system in the model has its own operation schedule defined. This is required even if all systems follow the same schedule. For example, in the attached example file, the four packaged systems have different operation schedule objects defined as PSZ1_HVACOperationSchd, PSZ1_HVACOperationSchd, PSZ1_HVACOperationSchd, and PSZ1_HVACOperationSchd.
2. Make sure each DX cooling object in the model has its own set of performance curves. This is required even if all DX cooling objects have the same part-load performance. For example, in the attached example file, the four DX cooling coil objects (CoilCooling:DX:SingleSpeed) have different sets of curves defined. The coefficients used in these curve objects do not matter because the EMS code will override them.
3. Copy the EMS code in the block of “Objects related to all zones” from the example file to the model.
4. For each of the packaged systems in the model, copy the EMS code in the block of “Objects related to PSZ1” from the example file to the model. Rename the EMS variables to avoid using the same name for different systems. This can be simply done by replacing “PSZ1” with some meaningful name.
5. For the internal variables in the EMS code, change the “Internal Data Index Key Name” as necessary to match the object names used in the model. For the sensors and actuators defined in the EMS code, change the “key name” as necessary to match the object names used in the model.
6. Customize the control as necessary. For example, the example file uses integrated differential dry-bulb economizer. Changes are needed if the model has other economizer controls.



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