MODELING HVAC OPTIMIZATION CONTROL STRATEGIES FOR HIGH PERFORMANCE BUILDINGS

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ABSTRACT

High efficient equipment selections and advance system designs do not necessarily lead to high performance buildings. In order to truly be a high performance building, the building must operate effectively to maximize the potential energy cost savings for the building owner. Producing these high performance building results requires the implementation and proper use of HVAC optimization control strategies. This paper will prove that equipment should be specified through the use of energy simulation optimization to insure that the greatest potential energy savings can be achieved.

INTRODUCTION

Most designers specify HVAC equipment based on design full load efficiency and or non-standard part load value (NPLV). Once they have completed a building load calculation in simulation software, such as Trace 700 or Hourly Analysis Program (HAP), they turn to equipment representatives to provide selections based on the capacity required at design conditions. This potentially leads to highly efficient equipment selections and the building performance may even meet LEED requirements, but does not maximize the potential energy cost savings for the building owner responsible for the utility bills.

Building optimization controls take into account current HVAC design, load profile, ambient conditions, utility rate structures and programs such as auto demand response (ADR), which allow the building HVAC system to operate at the most ideal conditions for energy savings and utility costs. Advanced control strategies have changed the way individual pieces of equipment interact with the system. Specifying equipment based on a complete optimized building energy model rather than full load equipment efficiency and or NPLV will increase the building potential energy savings and can even reduce initial equipment costs.

An energy simulation of HVAC optimization controls will show that the energy savings from an optimized building greatly outweigh those achieved by an efficient chiller selection alone; regardless of the criteria used in the selection process. Furthermore, an energy simulation model will enforce why the equipment selections should be made based on the optimized energy simulation model to achieve the greatest energy cost savings for the building.

BACKGROUND

Poor controls can waste more energy than great equipment can save. A car driven with a lead foot may achieve 10.0 mpg but if the driver changes his or hers aggressive driving style, the mpg rating can be dramatically increased. This is the same principal used in HVAC optimization. Even the most efficient chiller on the market cannot create a high performance building if not controlled properly. All systems have their “sweet spots” and when they run outside these desired conditions, efficiency can be dramatically reduced.

RELATED WORK

Current building mechanical HVAC design is complimented with equipment selections based on individual component efficiency. The selection is usually specified based on full load efficiency or IPLV/NPLV at design conditions. This potentially leads to highly efficient equipment selections for perhaps higher performance buildings, but does not maximize the potential energy cost savings for the building owner responsible for the utility bills.

The part load equipment ratings for chillers, integrated part load value (IPLV) and non-standard part load value (NPLV) are easy ways to compare the performance of two machines beyond the full load design-efficiency. Many discussions have taken place on whether to selection a chiller based on full load efficiency or NPLV. The HVAC&R Engineering Update by Johnson Controls titled “Use Only NPLV to

Specify Chiller Efficiency" outlines the benefits of the NPLV rating and why they believe NPLV is the correct approach for equipment selection. The 2009 ASHRAE journal published an article "A Closer Look at Chiller Ratings" which compared the IPLV/NPLV rating assumptions to the actual operation of equipment and found that the NPLV and full load efficiency are highly inaccurate at predicting actual performance and only help compare the unloading characteristics of similar chillers and do not infer economic savings. The author's further suggested that equipment selections be based on a full year 8,760 hour energy simulation model rather than IPLV or NPLV criteria, especially in multi-chiller applications.

The energy simulation model approach uses 8760 hour yearly weather data, and the correct building load profile to allow for a true analysis of energy use and chiller performance. This idea can be expanded to look at the HVAC system as a whole, analyzing building energy consumption, demand and utility costs at the meter level.

HVAC OPTIMIZATION CONTROL STRATEGIES

There are numerous ways a building can be controlled when it comes to the building automation system. Ultimately, it is the sustained strategy that saves the most energy and money for the owner that is the winner. The following HVAC optimization control strategies were implemented in the energy simulation model to prove why the use of energy simulation optimization to aid equipment selections will maximize a building's overall energy performance.

Cooling Tower Optimization

Condenser water reset measures adjust the condenser water temperature set point based on ambient conditions. Raising the condenser water set point will typical lower the cooling tower fan and pump energy, but will also increase lift on the chiller compressor resulting in an increase of chiller energy usage. Conversely, lowering the set point will typically have the opposite effect; raising energy use at the tower and pumps but lowering it at the chiller.

Cooling tower optimization will automatically reset the condenser water temperature set point to provide for optimum system operation, minimizing the total energy consumption of the cooling tower fan and chiller combination. This strategy takes into account the performance curves of the centrifugal chillers, outdoor wet bulb temperatures and the cooling tower fan energy efficiency. During part-load and favorable outside air conditions (cool/dry), the cooling towers can be used to reduce chiller energy demand by reducing the condenser water temperature. As load increases and/or the outside air conditions deteriorate the condenser water temperature is reset to a higher temperature to remain maximum system efficiency. Variable speed centrifugal chillers can be extremely efficient when the chilled water supply temperature is high, or the condenser water supply temperature is low, or a combination of both of these conditions.

Most buildings expend cooling tower energy in a wasteful manner in an effort to drop chiller kW. In many cases, reset strategies try to drive the condenser water temperature as low as possible all of the time. It has been seen cooling tower fan systems drawing more energy than the chillers when they were operated in this manner. (Duncan, 2001)

When modeling any condenser water reset the limitations of the equipment must be considered. Most notably the condenser water temperature must not be driven so low as to decrease the temperature difference between the condenser and evaporator of the chiller beyond the rated minimum.

Pump Differential Pressure Optimization

The chiller plant automation system resets the system chilled water pressure set point to the minimum value necessary to maintain control at the loads, rather than maintaining a fixed pressure set point.

Pump pressure reset saves energy without compromising space comfort. Chilled water pressure is reset based on critical valve position. As cooling demand is reduced at the cooling coils, the valves start to close; the most open of these is the critical valve. The critical valve can be reset fully open, other valves are reset proportionally, and the chilled water pumps are then slowed down to maintain the same amount of flow through the system. This ensures that the chilled water pumps maintain only the pressure required by the system to meet the cooling load resulting in significant energy savings.

Critical Zone Static Pressure Reset

This control strategy automatically maintains the duct static pressure set point of a variable air volume (VAV) system at the lowest possible level without sacrificing occupant comfort. The fan's static pressure setpoints are reset based on the VAV zone damper position. As cooling demand is reduced in a space, the VAV zone damper closes. For this optimization strategy, the most open of these dampers is the critical zone. The critical zone damper is reset to fully open, other dampers are reset proportionally. This creates a reduced required static pressure situation which allows the supply air fans' speeds to reduce and still maintain the same air flow rate through the zone dampers. This produces significant savings in large ducted HVAC system designs.

Building System Optimum Start & Optimum Stop

This control strategy saves energy by calculating the optimal time before occupancy the fan systems must turn on to meet temperature set point during occupancy. At the end of the day, this control strategy saves energy by calculating the optimal time before the occupants leave for the day the fans systems can turn off to meet the drift temperature set point right at closing time. Building heat transfer rate is continually measured to account for outside temperate variations. Savings from this strategy are realized from reduced run time from the fans and cooling system.

The above mentioned HVAC optimization control strategies work together lowering energy demand across the entire cooling system. When modeled correctly, the energy model will show greatly improved building energy performance. This is illustrated by comparing two identical energy simulation building models, with identical cooling systems and efficiencies, and implementing optimizations utilized on only one.

ENERGY MODEL SIMULATION

An 8,760 hour energy simulation model was created and analyzed in TRACE 700. The baseline energy model was created in accordance to ASHRAE 90.1 - 2007. A typical office building located in Los Angeles California was used in this analysis using TMY3 weather data from LAX airport. Southern California Edison energy rates were used in the analysis based on a time of use structure.

Table 1 Building energy simulation description

BUILDING DESCRIPTION	
Building Floor Area	450,000 Sqft.
Number of Floors	20
Building Use	Weekdays 9am-5pm Weekends 9am-1pm
Construction	ASHRAE 90.1 – 2007 Appendix G
Lighting Load	ASHRAE 90.1 – 2007 Appendix G
Occupancy Density	ASHRAE 62.1 – 2007
Miscellaneous Load	1.0 W / Sqft.
Ventilation	15 cfm/ per person

Airside System

The building was modeled with a typical overhead VAV system with built up air units. The air units had dry bulb temperature controlled airside economizers with a high level cut off of 75˚F. The supply and return fans had variable frequency drives (VFDs) installed in both the baseline and proposed models.

Central Cooling Plant

The central plant consists of two 500 ton centrifugal chillers (COP 6.1). The chiller curves and full load efficiencies were based on ASHRAE 90.1 – 2007, as were all set points and ranges. The baseline energy model's chilled water loop is configured in a primary secondary arrangement. Pumping power for both the chilled water and condenser water side were set to the 90.1 baseline standards.

Central Heating Plant

The heating plant was modeled to conform to ASHRAE 90.1 – 2007. No alterations were made to the central heating plants as it is outside of the scope of this analysis.

Controls

The baseline model has all control strategies/resets as required by ASHRAE 90.1 – 2007 Appendix G. For simplicity, demand control ventilation was omitted from this analysis.

Proposed Simulation Energy Model

As previously mentioned in the HVAC optimization control strategies section, the following control strategies were implemented into the proposed energy simulation model.

- Cooling tower optimization
- Pump differential pressure optimization
- Critical zone static pressure reset
- Building system optimum start and optimum stop

Cooling Tower Optimization

The baseline model chillers as they follow ASHRAE 90.1 Appendix G, has a minimum entering condenser temperature of 70˚F. As previously mentioned chiller entering condenser water can get a lot colder than 70˚F and still function properly. The proposed model's minimum condenser water temperature was reduced to 55˚F and the cooling tower optimization strategies were implemented to balance the condenser water temperature between 55˚F and design conditions to maximize efficiency.

Pump Differential Pressure Optimization

The pump differential pressure was reset based on critical valve position. This reduces the unnecessary head pressure during low loads.

Critical Zone Static Pressure Reset

The static pressure was reset based on the critical zone. This reduces unnecessary fan pressure during low loads.

Building System Optimum Start and Optimum Stop

The baseline Trace 700 model building operation was schedule to only operate during occupied hours. It must be noted, applying optimum start can increase energy consumption in cases where there are significant amount of unmet cooling hours. Although, optimum start allows the system to start at an optimal time reducing demand charges and unnecessary overcooling of the building. This strategy plays a major role when it comes to energy cost savings. Optimum stop will promote energy consumption savings by allowing the central plant to turn off at the optimum time at the end of the day before the occupants leave the building.

Key Points

No HVAC equipment changes were made between alternatives. Only control optimizations were implemented. Therefore, the same performance curves were used in the baseline and proposed energy model.

DISCUSSION & RESULTS ANALYSIS

Table 2 shows the ton-hrs for the cooling plant load used for the baseline energy simulation model in Trace 700.

Table 2 Monthly cooling plant ton-hrs summary

COOLING PLANT LOAD	
Month	ton-hrs
Jan	33,356
Feb	31,040
Mar	32,230
Apr	51,029
May	66,835
Jun	104,599
Jul	159,351
Aug	163,119
Sep	161,925
Oct	103,847
Nov	58,032
Dec	31,980
Total	997,342

Based on the total building ton-hrs and a total peak cooling load requirement of 889 tons, the equivalent full load hours (EFLH) of the central cooling plant totaled 1,122 hours. (Table 3)

Table 3 Engineering Checks

ENGINEERING CHECKS	
Max Cooling Load (tons)	889
EFLH	1,122

From Table 4, the baseline average annual efficiency of the central cooling plant was 0.83 kW/ton and the proposed optimization model was 0.72 kW/ton. With all the optimization strategies applied in the proposed model, the chillers are operating more frequently at a less efficient point on their performance curves. This drives the annual kW/ton upwards for the chiller. This further proves why NPLV and full load efficiency should not be used for equipment selections.

Table 4 Energy Summary

	CHILLER/ COMPRESSOR KW/TON	CENTRAL PLANT TOTAL KW/TON
Baseline	0.53	0.83
Proposed	0.58	0.72

From the building simulation results, the actual part load profile can be determined and a chiller can be selected based on the optimal efficiency for that load profile. By using this strategy, greater energy savings can be achieved over using NPLV and full load efficiency.

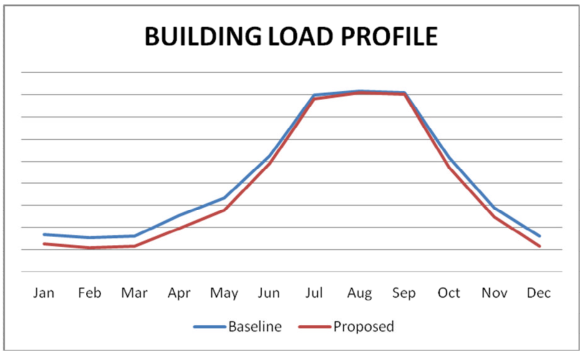

Figure 1 Building Load Profile

Figure 1 shows the building ton-hours per month. The HVAC optimizations allow the chillers to work less during months where part load conditions are more prevalent. (high chilled water supply temperature, low entering condenser water temperature, low outdoor wet bulb temperature)

Tables 5 and 6, along with Figures 2 and 3, illustrate the total annual energy consumption and demand for both the baseline and proposed energy models. There is an energy consumption reduction of 428,785 kWh and a 415 kW demand reduction from the optimization control strategies alone. This relates to a total energy cost savings of $148,871 a year. (16%) shown in Figure 4. The costs were based on Southern California Edison TOU-8-B rate structures.

Table 5 Energy Usage

	BASELINE USAGE (KWH/YR)	PROPOSED USAGE (KWH/YR)	ENERGY SAVINGS (KWH/YR)
Proposed Building - Optimized	4,722,407	4,293,622	428,785

Table 6 Energy Demand

	BASELINE ON-PEAK DEMAND (KW)	PROPOSED ON-PEAK DEMAND (KW)	ON-PEAK DR (KW)
Proposed Building - Optimized	2,068	1,653	415

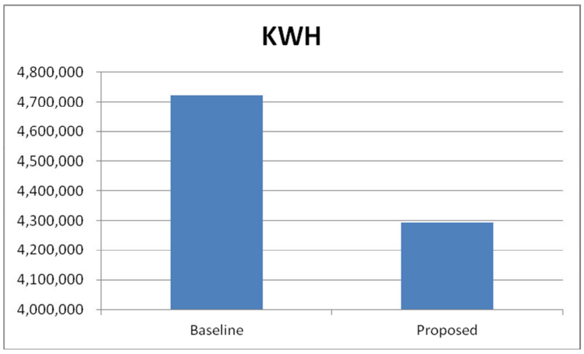

Figure 2 Building Energy Consumption Comparison

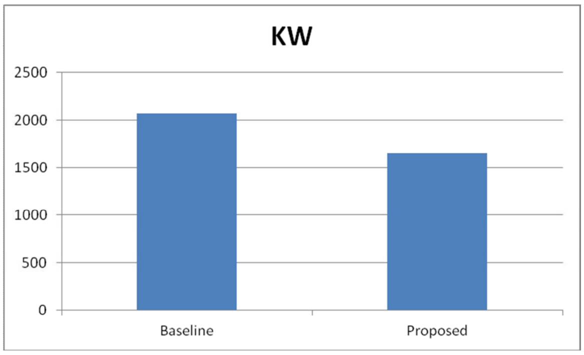

Figure 3 Building Energy Demand Comparison

Table 7 shows the energy utilization index and cost index for both the baseline and proposed model.

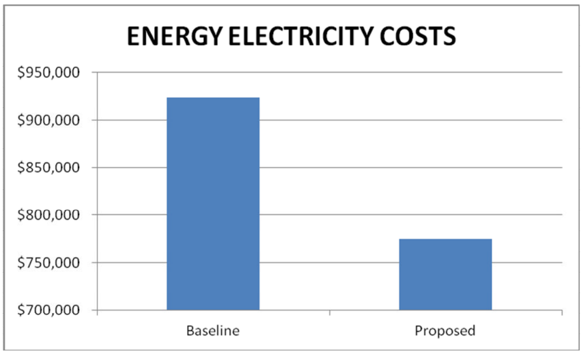

Figure 4 Building Energy Electricity Cost Comparison

Table 7 Energy Summary

ENERGY CONSUMPTION	EUI (KBTU/SF/YR)	CI ($/SF/YR)
Calibrated Model	36	2.05
Proposed Model	33	1.72

These figures clearly illustrate the energy and cost savings provided by HVAC controls optimization strategies. Cost savings of this magnitude will also provide for LEED EA Credit 1 points. These savings are across the entire building, not isolated to the chillers.

CONCLUSION & FUTURE WORK

HVAC controls optimization strategies provide for high performance buildings and the savings can be proven through energy simulation modeling. Chiller selections should be based on supporting total building performance, not equipment efficiency. An energy model of the building will show where the chillers will operate most frequently. Maximizing this efficiency will produce the most energy cost savings.

These strategies can be further leveraged for demand response. By using more aggressive optimizations during a demand response event, further cost savings can be realized. This will be shown in future publications.

The energy model can be further optimized with regards to the central cooling plant design. Designing a chiller for lower chilled/condenser water temperatures and variable flow rates allows for even less pump and fan energy on the water side and air side systems. Therefore pumps and fan sizes can be reduced resulting in lower first costs. Further savings could also result from optimized duct work design.

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