Advanced Cooling Technology with Thermally Activated Building Surfaces and Model Predictive Control

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Abstract

This research analyzes an advanced cooling system, termed a low-lift cooling system (LLCS), that comprises thermally activated building surfaces (TABS) and a parallel dedicated outdoor air system (DOAS) for dehumidification and ventilation. The system utilizes model predictive control (MPC) that, based on weather and load predictions, determines the cooling strategy over next 24 hours that minimizes energy consumption. Different objectives, such as minimizing the total cost of electricity, can be achieved by modifying the objective function. The LLCS performance was analyzed across 16 different U.S. climates relative to a variable refrigerant flow (VRF) for sensible cooling only, and to the VAV system for cooling, dehumidification and ventilation. Five dehumidification strategies that can be used in combination with the LLCS were also investigated. The results suggest that the electricity savings using the LLCS are up to 50% relative to the VAV system under conventional control and up to 23% relative to the VAV system under MPC. The savings were achieved through lower transport energy and better utilization of part-load efficiencies inherent in inverter-compressor equipment, a result of the TABS technology and the optimal control. The LLCS also had better performance than the conventionally controlled VRF system.

Keywords: advanced cooling technology, model predictive control, energy efficiency

1. Introduction

In most developed countries about 40% of the total energy and 70% of electricity is consumed by the building sector (7). Current projections suggest that growing trends in energy consumption in the building sector will continue, and at a somewhat faster rate for commercial than residential buildings. In an effort to reduce energy consumption, energy efficiency of buildings is slowly being promoted through different policies, such as the European Energy Performance of Buildings Directive (European Parliament, 2002). Numerous manuals and codes give valuable recommendations for an improved building

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envelope, building air tightness, equipment efficiency, and similar improvements for existing solutions. However, the building industry in general seems very conservative when it comes to the implementation of advanced technologies, necessary for the design of low energy buildings and their scaling to a larger market. Commercial buildings are in general dominated by internal loads rather than climate, due to a small surface-to-volume ratio, and high internal loads from people and equipment. Therefore, the building envelope improvements can help to a certain extent, but the majority of energy reduction needs to come through better lighting control and advances in cooling and ventilation technology. This papers analyzes the performance of the advanced cooling system referred as a low-lift cooling system (LLCS). The specific LLCS configuration comprises thermally activated building surfaces (TABS) for sensible cooling, and a parallel dedicated outdoor air system (DOAS) for dehumidification and ventilation. The TABS and DOAS are served by a water-to-air and air-to-air heat pump respectively, with variable speed drive for compressors, fans and pumps. The LLCS is operated under model predictive control (MPC) that optimizes its performance for the lowest energy consumption, although other objectives, such as price of electricity, are possible. The LLCS could also be used for heating, but this was not considered in this research.

The benefits of separate components of this system have been shown in numerous papers found in the literature. Decoupling the sensible (temperature) control from the latent (humidity) and ventilation control was suggested for the improved indoor air quality (IAQ) and energy savings (5, 28, 11). In a decoupled system, ventilation and humidity are controlled by the DOAS, which can also deliver a certain amount of sensible heating/cooling. The remaining sensible loads are met by a parallel system. The previous research showed that the combined DOAS and parallel cooling system can result in 14–60% annual energy savings and 17–50% peak power savings (35, 17, 27, 18, 21, 8, 26). The reported savings were demonstrated using simulations, as well as field projects, and were strongly dependent on climate, building type, system type and simulation assumptions. The most research was done for a typical office building, comparing the system with the radiant panels and DOAS against the VAV system. Although radiant systems have a good potential for a water-side economizer due to higher water supply temperatures, this was considered only in two analyses found in the literature (37, 31). Comparing the radiant system with parallel DOAS against the VAV system Tian and Love (37) reported the largest savings (up to 60%) for dry climates (hot and cold). Humid climates had lower savings due to the need for the continuous ventilation for dehumidification purposes. Stetiu (35) also reported lower savings in cold, moist climates with better potential for an VAV system air-side economizer.

The advantage of night precooling, with or without the use of an advanced control, was also thoroughly reported, mainly for VAV systems. The results showed 5–50% reduction in the operating cost and 10–50% peak load reduction (34, 4, 32, 22, 15, 24, 25, 40, 6, 3). For an optimally controlled building with a VAV system, factors identified as the driving factors for a cost saving potential were the utility rates, building mass, internal loads, equipment efficiency, and equipment part-load performance (16). The highest savings were achieved for a building with high utility incentives, low internal gains, and with the equipment characterized by good part-load performance. No real savings were achieved for a building with high internal loads, regardless of the thermal mass.

The novel concept of combining radiant panels, thermal energy storage, variable-drive and advanced control was proposed by Jiang et al. (19) and Armstrong et al.

(2). The reported annual cooling energy savings of up to 75% were found compared to a baseline ASHRAE 90.1-2004 VAV system (1). The subsequent analysts confirmed the saving potential across 16 U.S. climates (20) and showed that the LLCS can be a cost competitive technology when compared to a conventional system. An estimated component incremental cost for a large office building was approximately 7.5 $\$/\mathrm{m}^2$ (above the new construction cost of 82 $\$/\mathrm{m}^2$), while a medium office building even had a negative incremental cost of -6 $\$/\mathrm{m}^2$, mainly due to the large cost of a multi-zone rooftop system (used in a baseline configuration) relative to a comparable sized chiller. The experimental verification of the energy saving potential was provided by Gayeski et al. (14) for a typical summer week for Atlanta and Phoenix. The tests were performed in the experimental room at Massachusetts Institute of Technology, USA, equipped with the low-lift and standard variable refrigerant flow (VRF) configurations. Both the VRF system and low-lift configuration used the same compressor-condenser unit. The results for a typical summer week in Atlanta and Phoenix showed sensible cooling savings of 25% and 19% respectively, relative to the VRF system. The savings potential of the proposed system could be improved even further by advancements in the heat pump industry. A prototype of the chiller for a small temperature lift was recently developed by Wyssen et al. (39). The prototype included a specially sized expansion valve and the use of a reciprocating compressor to avoid high internal pressure ratios. It was suggested, based on the example of an office building, that for the same operating conditions the new prototype would result in an approximately 6°C smaller lift, and therefore the resulting COP would be 1.6 times higher than the existing chiller.

Although the previous study of the LLCS showed great energy savings potential, the analysis by Jiang et al. (19) was done using a relatively simple computational tool and some idealized assumptions, such as an ideal active thermal storage. Furthermore, the same study showed that the potential customers were somewhat discouraged by the use of active thermal storage, which in general takes useful space and is perceived to be challenging to control. In this paper the LLCS is compared to the VAV system using a more detailed simulation tool for buildings with MPC (42). It allows for the analysis of many factors that influence savings potential, such as temperature limits, pipe spacing, and transport power. It is also shown in this paper that the use of building mass can be a feasible and efficient method of avoiding active thermal storage. Furthermore, humidity control with the DOAS is especially an important issue for buildings with TABS due to possible condensations problems. Most of the work found in the literature is focused toward analyzing the possible benefits of a typical constant-air-volume DOAS with or without an enthalpy wheel (33, 30, 29). Although Gatley (12) proposed promising alternatives to the typical DOAS, the analysis of several DOAS configurations was performed here to determine their feasibility for different scenarios with the use of the LLCS. Finally, in addition to the comparison between the LLCS and VAV system, the LLCS is also compared to the variable refrigerant flow (VRF) system. VRF systems are recently becoming more popular, even for such large buildings as hotels, and are attractive since they can provide both heating and cooling, and can save transport energy compared to all-air systems.

2. Model description

The performance of the low-lift cooling system (LLCS) is compared to the VAV and VRF system performance using the modeling environment described in more detail in Zakula et al. (42). The LLCS performance analysis also considers several dehumidification configurations, assessing their energy use for different climates. The analyses are done for a typical summer week (two weekdays, weekend, three weekdays), and over the cooling season from May 1 until September 31. The typical weather conditions across 16 climates representative of the U.S. are simulated using TMY3 weather files.

Two control strategies used in the analyses are a conventional control and MPC. Under a conventional control, the system operates during the occupied hours, to maintain a given setpoint temperature. Under MPC, temperature limits are allowed to float between a lower and upper limit during the occupied hours, and the cooling rates are optimized for the lowest energy consumption, allowing for night precooling. The optimization variable for cooling with TABS is a chiller cooling rate, and the optimization variable for the VAV system and VRF system is a sensible cooling rate imposed on the room. The objective function is defined as a sum of the total daily electricity for cooling, electricity for transport and the temperature penalty, as shown in Zakula et al. (42). The temperature penalty ensures that the controlled variable, the operative temperature in this case, is inside the desired comfort range. Both the planning horizon (the time interval over which the objective function is evaluated) and the execution horizon (the time interval over which the control strategy is applied) are 24 hours since one has a perfect knowledge of weather conditions and loads in simulations. This results in 24-variable optimization, one cooling rate for each hour of a day. To calculate the energy required for conditioning of the air/water to the supply conditions, the optimization algorithm uses curve fits to the heat pump static optimization data, as explained in Zakula et al. (42).

2.1. Building model

The analysis is performed on the model of the MIT test room described in detail in Zakula et al. (42), representing a typical office space. The room was chosen because the experimental measurements for a typical summer week in Atlanta (13) were available to validate the model. The test room has floor pipes that can be used for hydronic sensible cooling or heating, and has an additional indoor unit (VRF system) for direct heating, cooling and dehumidification. Although the experimental measurements were performed with the pipe spacing of 0.3 m, the spacing for the analysis here was reduced to 0.15 m, as more appropriate for the cooling mode. The room is also equipped with lights and heat sources that can simulate internal convective and radiative heat gains for a typical office building, while the solar gains are neglected.

In the comparison of the LLCS and VAV system, the peak sensible internal load for the 19 m^2 room is 680 W (2 people each releasing 80 W, 220 W for lights and 300 W for the equipment), or approximately 36 W/m^2. The occupied hours are from 8–18 h, with 66% of the maximum internal loads from 8–9 h, 100% from 9–17 h, and 66% from 17–18 h. The internal gains are modeled as 50% convective and 50% radiative. Although not included in this work, solar gains would be an additional heat gain to the zone. However, office buildings, which are the best first candidates for LLCS implementation according to the PNNL study, are internally dominated buildings due to a small ratio of external surface to building volume. Hence, it is not anticipated that including solar gains would substantially change findings of this analysis, especially for core building zones. The only sources of latent gains during the occupied hours are loads from people of 0.144 kg/h (2 people each releasing 0.072 kg/h or 50 W). Latent loads caused by infiltration are neglected during the occupied hours since most commercial buildings are slightly pressurized to avoid infiltration. During unoccupied hours, the analysis accounts for latent loads by infiltration. According to the U.S. National Institute of Standards and Technology data base (9), the average measured airtightness of 228 commercial building (normalized by the above-grade surface area of the building envelope) is 24.8 $m^3/h/m^2$ at 75 Pa. The value recommended by ASHRAE Standard 189.1 for the Design of High-Performance Green Buildings, and also by 2012 International Energy Conservation Code is 7.2 $m^3/h/m^2$ at 75 Pa. When converted to a more typical pressure difference under ambient conditions (4 Pa), and expressed in ACH (based on the geometry for a medium-size office from DOE benchmark buildings), the average measured airtightness and recommended value are 0.37 ACH and 0.11 ACH respectively. The value used in this analysis is 0.2 ACH, between the measured and recommended value. The ventilation rate for both the VAV and LLCS system are 0.01 kg/s/person (8.5 l/s/person), according to ventilation requirements from ASHRAE Standard 62.1-2007 for office buildings.

In the comparison between the LLCS and VRF system, the simulation parameters were set to replicate the experimental measurements by Gayeski et al. (14). The simulations for the Atlanta climate assume standard office internal loads of 36 W/m^2, and for the Phoenix climate reduced loads of 22 W/m^2, representative of a high-performance building. The ventilation and dehumidification systems are not included in this analysis. Is is assumed that both the LLCS and VRF system would have an additional system for ventilation and dehumidification, and would require similar additional power for conditioning and transport of the outdoor air. Therefore, it is expected that this additional system would not have a major impact on the findings presented here.

2.2. VAV system

The VAV system delivers air at the constant supply temperature of 12.5°C (saturated air), and with airflows sufficient to remove zone sensible loads. When the outside temperature is higher than the zone's temperature, the minimum amount of fresh air for breathing is mixed with the recirculation air. When the outside temperature drops below the zone's temperature, the VAV system runs in an economizer mode, with 100% outside air.

The VAV system uses the air-to-air heat pump performance map, with the evaporator operated in a wet-coil mode, providing cooling and dehumidification. Based on outdoor and supply air conditions, the heat pump model calculates the ratio between latent and sensible cooling on the evaporator, using a wet-coil model developed from first principles according to Threlkeld (36). The evaporator airflow is dependent on the zone's sensible loads that needs to be removed, and the condenser airflow is optimized. The VAV heat pump is sized for each climate based on the cooling coil peak loads.

2.3. VRF system

The VRF system provides direct cooling to the zone, with an indoor unit placed in the zone, and the condenser cooled by outside air. Although the VRF system can also provide dehumidification, the dehumidification was not included in the comparison of the LLCS and VRF system.

The VRF system uses the air-to-air heat pump performance map, with the evaporator operated in a dry-coil mode, providing only sensible cooling. A heat pump is operated at the constant evaporator airflow and optimized condenser airflows. If the outside temperature is lower than the air temperature, the heat pump can also operate in the refrigerant-side economizer mode (2). The heat pump capacity is 3 kW, same as in the experimental measurements by Gayeski.

2.4. LLCS system

The LLCS consists of thermally activated building surfaces (TABS), a water-to-air heat pump with variable-speed drive for the compressor, and fans, and a dedicated outside air system (DOAS) for ventilation and dehumidification. The pipe spacing is 15 cm, except in the comparison of simulation and experimental measurement results (analysis of the LLCS and VRF system), where the pipe spacing is set to 30 cm. Sensible cooling through the TABS is controlled by varying the water supply temperature and pump operation (on/off mode), with the water mass flow rate being constant.

Water for the TABS is cooled by the water-to-air heat pump that operates at a constant evaporator water flow rate and optimized outside condenser airflow rate. If the outside temperature is lower than the water temperature, the heat pump can also operate in refrigerant-side economizer mode. When comparing the VAV and LLCS system, the LLCS heat pump is of the same capacity as the VAV heat pump, but is on average operating at lower part-load ratios than the VAV heat pump, increasing the average COP. The lower part-load ratios are the result of shifting cooling loads toward the night time, and providing a certain amount of sensible, and a total amount of latent cooling through the parallel system, DOAS.

2.4.1. DOAS

The DOAS assumed in this work is a variable-volume system controlled based on a room humidity sensor. The air is supplied to the room at the constant absolute humidity of 9 g/kg (which corresponds to the saturation temperature of 12.5°C), the same as the supply state for the VAV system. These conditions are chosen based on the two following criteria. First, assuming the zone's humidity setpoint of 11 g/kg, a minimum required amount of fresh air 0.01 kg/s/person, and latent loads of 0.072 kg/h/person, the supply air humidity of 9 g/kg is sufficient for the removal of latent loads using the minimum amount of fresh air required for breathing. Second, the lowest allowed air supply temperature is usually 13°C for comfort criteria and, therefore, the supply temperature in this work is chosen to prevent the need for reheat. One can argue that the lower dew-point temperature would result in reduced airflows required for dehumidification, and therefore a reduced fan power. However, it would also result in lower heat pump efficiency due to lower evaporating temperature and the need for reheat energy. To prevent possible condensation on the cold TABS, the DOAS is also operated during the night, delivering airflow rates necessary to remove infiltration latent loads and maintain the desired humidity. The DOAS does not operate during weekends.

Five dehumidification strategies shown in Figure 1 were considered. All strategies utilize the enthalpy recovery wheel, as an efficient way to recover sensible and latent heat from the return air. Although wheel efficiency will depend on its size relative to the airflow, performance characteristic, and the rotational speed, it is assumed that the total sensible and latent heat recovery efficiency is 0.8. Furthermore, it is assumed that the wheel operates only if the outside air enthalpy is higher than the return air enthalpy. System A is a typical DOAS found in a majority of analyses of a combined radiant system and DOAS. The system consists of an enthalpy recovery wheel and a cooling coil, with heat rejected to the outside air. System B has a similar configuration, but with heat transferred to the supply air stream. This configuration could possibly improve the heat pump COP due to lower condenser air temperatures. Although this adds sensible loads to TABS system, it was expected that the radiant system can remove those loads more efficiently during the night precooling. However, this system was found unfeasible for the practical implementation. Although the condenser air temperatures were lower than in the basic DOAS, the condensing refrigerant temperatures were still relatively high due to limited condenser airflows. Additional simulations were performed to analyze whether the performance of system B can be improved by increasing the condenser area. However, even with three-times-larger condenser depth, there was little or no improvement in the COP due to limited condenser airflows. Moreover, for high loads on the cooling coil, it was not always possible to reject all the heat on the condenser. After experiencing practical difficulties with system B, two variations of system B were considered, both with two parallel condensers. The first condenser is placed in the supply stream, and the second is placed in the exhaust stream after the enthalpy wheel (system C), or outside (system D). Although system D will on average have higher condenser inlet air temperatures than for system C, it will also allow for higher, optimized condenser airflow rates. System E is a variation of system A, with a run-around heat pipe used to precool the air before entering the evaporator. This can, again, have positive implications for the COP due to lower evaporator inlet air temperatures. It is assumed that the heat pipe has a constant efficiency of 0.5 (38).

The DOAS heat pump capacity is sized based on the peak latent loads and ventilation needs, and is equal for all climates. System A uses the similar performance map as the VAV system, assuming the supply temperature of 12.5°C, and the condenser placed outside. The condenser airflows are optimized for the lowest heat pump energy consumption. In systems C and D, one condenser was placed in the supply air stream, resulting in supply temperatures higher than the supply temperature of 12.5°C assumed for system A. Therefore, the supply air temperature increase across the condenser (shown for system C in Figure 2a) is calculated based on the amount of rejected heat on the first condenser. To calculate heat rejected by the first condenser, the heat pump model first needs to calculate the split between the heat rejected by each condenser. This is done assuming that the condensing pressure in both condensers are the same, since both condensers are served by the same compressor. After the air temperature increase is found, polynomials of the third order are fitted to the temperature increase curves, with fitted values being a function of the part-load ratio Q/Q_{max}, evaporator airflow rate V_e, and condenser inlet air temperature $T_{c,air,in}$. Figure 2b shows fitted values (red) to the results of the static optimization (black) for a specific evaporator airflow rate. The air temperature increase for both system C and D, as well as more detailed description of the overall model and simulation assumptions can be found in Zakula (41).

3. Results

The first analysis compares the LLCS with different dehumidification configurations against a conventional VAV system (marked in figures as 1). The conventional VAV system was operated only during the occupied hours to maintain the operative temperature at 22.5°C. The LLCS employed MPC to optimize cooling rates over the 24-hour planning horizon for the lowest electricity consumption. It allowed precooling the building during night in order to maintain the operative temperature between 20–25°C during the occupied hours. The temperature limits were set according to ASHRAE comfort standards. The analyzed DOAS configurations were: configuration A with the condenser cooled with the outside air (referred in text as LLCS 2, and marked in figures as 2); configuration C with parallel condensers, one cooled with the supply, and the other with the return air (LLCS 3); configuration D with parallel condensers, one cooled with the supply, and the other with the outside air (LLCS 4); and configuration E with the run-around heat pipe and the condenser cooled with the outside air (LLCS 5). Configuration B has not been analyzed due to the practical issues mentioned earlier.

The second analysis compares the LLCS against the VRF system for a typical summer week under the Atlanta and Phoenix climates. The LLCS room temperature limits were 19–25°C during the occupied hours, while the VRF system was controlled to maintain constant 22°C during the occupied hours. A somewhat wider temperature range than recommended by ASHRAE comfort standard was used here to analyze the influence of temperature limits on savings, and also to replicate the experimental measurements by Gayeski et al. (14).

3.1. Comparison of VAV system and LLCS for sensible cooling and dehumidification

The main difference between cooling with the VAV system and LLCS is shown in the example of a typical summer week in Phoenix. The VAV system cooled only during occupied hours (Figure 3, top graph), maintaining the steady operative temperature of 22.5°C (Figure 4, top graph). The LLCS made advantage of lower night temperatures by precooling the building over night (Figure 3, bottom graph), which caused the operative temperature to slowly rise from lower morning temperature to higher temperatures in the afternoon (Figure 4, bottom graph graph).

The cooling energy (Figure 5), electricity consumption (Figure 6) and total electricity savings (Figure 7) are shown for a typical summer week across all climates. The total LLCS cooling energy (bars marked as 2–5 in Figure 5) was lower relative to the VAV system (bar marked as 1) for mild climates (Fairbanks, Los Angeles, San Francisco and Seattle). This was expected since the LLCS had a wider temperature range than the VAV, allowing temperatures to float up to 25°C during the occupied hours. For the climates with high cooling needs, the total cooling energy consumption was higher than the VAV system due to losses inherent to thermal storage. However, despite using more cooling energy than the VAV system, the LLCS had 18–53% lower total electricity consumption across all climates (Figure 7). The electricity savings came from the reduction in the electricity for cooling (black and grey bars in Figure 6) and electricity for the transport (pink bars). The electricity savings are defined as:

$$Savings = \frac{(E_{VAV} - E_{LLCS})}{E_{VAV}} \times 100 \tag{1}$$

The intent for the LLCS 3 and LLCS 4 was to improve the DOAS heat pump performance by placing one condenser in the supply stream and cooling it with a cold air exiting the evaporator at 12.5°C. However, the DOAS in the LLCS 2 delivered a certain amount of sensible cooling to the zone (negative green bars), while in the LLCS 3 and LLCS 4, the DOAS caused sensible heating (positive green bars) and the need for an additional sensible cooling thought TABS (Figure 5, blue bars). The load shifting through the use of TABS generally has a positive impact on the energy consumption. In this case, however, the DOAS still used electricity to cool/dehumidify the fresh air, and to transport it to the zone. Consequently, although the LLCS 3 and LLCS 4 reduced the electricity consumption for the DOAS heat pump (grey bars in Figure 6), the electricity for TABS cooling (black bars) and for the transport (pink bars) increased. For example, in the Phoenix climate DOAS heat pump electricity was reduced by 46% for the LLCS 3. However, TABS cooling energy increased approximately 48%, and the electricity for the TABS heat pump increased 80% due to higher part-load ratios and more cooling during warm hours. The total transport energy also increased by 15% due to more pump-on hours. The LLCS 5 with the run-around heat pipe also reduced the amount of sensible cooling delivered by the DOAS, although not as much as the LLCS 3 and LLCS 4. The electricity for the DOAS heat pump was again reduced due to lower sensible loads on the cooling coil. However, the total electricity consumption was still somewhat higher than for the LLCS 2 due to additional cooling and transport energy used for cooling through TABS.

3.1.1. Effect of allowable room temperature excursions and precooling

The following analysis investigates the impact of wider temperature limits and the use of MPC for the VAV system, allowing the operative temperature to float between 20–25°C instead of constant 22.5°C. Furthermore, it also investigates the impact of MPC and precooling for both the VAV and LLCS.

The VAV and LLCS cooling rates were optimized using MPC for 24-hour operation and daytime-only (during the occupied hours) operation. The results suggests that the VAV system precooling does not have a significant impact on the total electricity consumption, with in increase in savings less than 3%. The VAV system with precooling shifted a certain amount of cooling toward early morning hours, but the impact was marginal because the VAV system cannot engage the building mass storage potential as effectively as the LLCS. Furthermore, the system was maintaining a relatively constant temperature of 25°C through the day, despite being allowed a wide temperature range. The impact of precooling for the LLCS was notably more pronounced, especially for hot climates such as Las Vegas (15% difference) and Phoenix (11% difference). Precooling with the LLCS resulted in a steady temperature increase from 20 to 25°C during the occupied hours, and consequently a lower average daily temperature compared to the VAV system.

Comparing the electricity consumption of the LLCS with precooling relative to the VAV system with precooling, the savings ranged from -11% (VAV used less electricity) for Los Angeles and Seattle to 29% (LLCS used less electricity) for Phoenix. The total cooling energy for the LLCS was higher than the VAV system due to losses associated with passive thermal storage. However, for most climates higher heat pump efficiency and lower transport power for the LLCS resulted in lower electricity consumption compared to the VAV system. The only climates where the VAV system performed better than the LLCS were Fairbanks, Los Angeles, Seattle and San Francisco, mild climes with the lowest cooling energy needs. Between a hot, humid climate such Miami and a hot, dry climate such as Phoenix, a humid climate showed notably less savings since more energy was required for dehumidification. In humid climates, the DOAS needs to deliver more air during night to remove latent loads caused by infiltration, resulting in higher transport and cooling energy. Nevertheless, comparing the LLCS and VAV system performance for Miami, the zone's humidity oscillations were lower for the LLCS due to a decoupling of the humidity and temperature control.

Finally, the electricity consumption of the VAV system with MPC was compared to the conventional VAV system that operated only during the occupied hours maintaining the constant temperature of 22.5°C. As expected, allowing larger temperature range resulted in significant savings of 30–50%, primarily caused by the increase in the operative temperature, and only marginally by precooling.

To estimate how the LLCS would perform relative to the VAV system over the whole spring and summer season, a 22-week period (from May 1 until September 30) was simulated for five climates with large cooling energy needs (Chicago, Houston, Las Vegas, Miami, and Phoenix). Results for the VAV system with and without precooling, and for the LLCS with and without precooling confirmed that the precooling has a significantly higher impact for the LLCS. The LLCS also performed better than the VAV system, with electricity savings 14–22% relative to the VAV system with precooling, and 43–50% relative to the conventional VAV system.

3.1.2. Effect of internal loads

Analyzing the cost savings potential of the VAV system under MPC, Henze et al. (16) noted that no real savings were achieved for a building with high internal loads. To test these findings for the LLCS and electricity consumption savings rather than cost savings, different magnitudes of internal loads were imposed, ranging from 20 W/m^2 to 60 W/m^2. The simulation results showed that the LLCS savings decreased significantly for the highest internal loads. For example, in the comparison between the LLCS and conventionally controlled VAV in Phoenix, the electricity savings decreed from 50.9% to 44.2%, when internal loads increased from 20 to 60 W/m^2. However, the significance of precooling for the LLCS increased with the increase of internal loads. For low internal loads, a large portion of sensible cooling was still provided through the DOAS. As the loads increased, the TABS cooling became predominant, with a greater opportunity for load shifting. For the same example of Phoenix, the LLCS with precooling had 7.2% savings relative to the LLCS without precooling when loads were 20 W/m^2, and 17.3% when loads were 60 W/m^2.

3.1.3. Effect of high latent loads

In case of high latent loads in the room, the DOAS airflow rates required for dehumidification might be sufficiently high to remove all sensible loads as well. The total energy consumption of the DOAS will in that case be higher than the VAV system since the DOAS operates with 100% outside air, while the VAV system mixes fresh and return air. This limiting case was identified for five climates by finding the latent loads for which all latent and sensible cooling is done by the DOAS. The limiting case is presented in Table 1 (assuming standard sensible internal loads of 36 W/m^2) in terms of the maximum latent loads in kg$_{water}$/h, equivalent ACH of infiltration, and equivalent number of people. For example, in the Chicago climate the latent load at which all latent and sensible loads would be removed solely by the DOAS was 0.4 kg_{water}/h. That latent load is equivalent to having 1 ACH infiltration rate based on the outside humidity for Chicago, or having 6 people in the room. (In Las Vegas climate, the equivalent ACH is infinite since the outside humidity is lower that the zone set point humidity). For comparison, Emmerich and Persily (9) recorded the average measured airtightness of 228 buildings commercial building of 0.37 ACH at 4 Pa, and the recommended value by ASHRAE Standard 189.1 for the Design of High-Performance Green Buildings is 0.11 ACH at 4 P. This demonstrates that the limiting case would be difficult to achieve in a typical commercial building with a typical occupant density and typical leakiness.

Table 1: Latent loads for limiting case when all latent and sensible cooling are provided by DOAS

	kg_{water}/h	**$ACH_{infiltration}$**	**No of people**
Chicago	0.4	1.0	6
Houston	0.5	1.5	7
Las Vegas	0.9	∞	12
Miami	0.5	1.5	7
Phoenix	1.0	4.5	14

3.1.4. Effect of transport power

It was observed by Krarti and Henze (23) that the fan energy can have a significant influence on the predictive controller decisions and, therefore, should not be neglected. This assumption was tested by excluding the transport power from the objective function. The optimal cooling rates for both the VAV system and LLCS were found only by minimizing the energy for cooling, after which the transport energy was added to the total cost function. Results confirmed the findings by Krarti and Henze (23) that excluding the transport power from the objective function can indeed have a significant impact on the predictive controlled decisions. When the transport power was not included in the objective function, the cooling rates for both the VAV system and the LLCS were more spread out during the day, taking the advantage of lower part-load ratios and lower air temperatures. This resulted in lower electricity for cooling, but also in higher transport energy consumption due to a larger number of fans/pump operating hours. The increase in the total electricity consumption for the VAV system ranged from less than 1% (Houston, Las Vegas and Miami, Minneapolis, Phoenix) to 150% for Helena. Similar trends were observed for the LLCS, but with differences being notably smaller (from less than 1% for Houston, Las Vegas and Miami, Minneapolis, Phoenix to 30% for Seattle. This was expected since the LLCS transport energy accounts for a smaller portion of the total energy.

3.1.5. Effect of return air flow

The return air flow in the DOAS and VAV system will be somewhat lower than the supply air flow due to building pressurization. Mumma (29) showed that for an office building with a leakage rate of 5 $m^3/(hm^2)$ at 50 Pa, and with ASHRAE Standard 62.1 ventilation requirements, the ratio of the pressurization flow to the total ventilation flow rate should be around 0.7 to achieve adequate pressurization. Mumma (29) also noted that the recovery wheel needs to work with balanced air flows, meaning that its efficiency will decrease depending on a return air flow reduction. To assess the impact of the reduced return airflow, the DOAS is tested for 50% of the return airflow relative to the supply. The results suggested that the LLCS with 50% return flow would consume between 5% (Chicago) and 11% (Phoenix) more electricity relative to the LLCS with 100% return flow. However, compared with the VAV system with precooling and to the conventional VAV system, the savings would still range from 12–20% and 42–47% respectively. For comparison, the LLCS with 100% return air showed savings of 17–23% and 45–53% relative to the VAV system and the conventional VAV system respectively.

3.2. Comparison of VRF system and LLCS for sensible cooling only

To compare LLCS savings relative to the VRF system achieved by experimental measurements and simulations, the pipe spacing here was increased to 30 cm (same as in experimental measurements). While the experimental measurements showed 25% and 19% savings for Atlanta and Phoenix respectively, the simulation results showed LLCS electricity savings of 8.9% and 9.7%. Although it is interesting that simulations showed lower savings than the measurements, this is caused by inevitable differences in modeling, especially in modeling the heat pump performance. The heat pumps used in the simulations were carefully optimized; hence, although trying to match the heat pump operation used in the experiment, the simulated heat pumps operated more efficiently than under experimental conditions. Furthermore, savings predictions are also highly sensitive to temperature setpoints, hence the small differences in temperature profiles between simulation and measurements can be an additional cause of differences in savings.

3.2.1. Effect of pipe spacing

To test the sensitivity of predicted savings on the pipe spacing, a new TABS system was simulated with the reduced pipe spacing of 15 cm, which is more common for cooling with TABS. This reduction improved the total effectiveness of TABS heat transfer, resulting in higher TABS water temperatures and significantly larger electricity savings, 24.9% and 25.3% for Atlanta and Phoenix respectively.

3.2.2. Effect of heat pump optimization and sizing

The VRF system (air-to-air heat pump) used in the experimental measurements operated with optimized condenser airflows and constant evaporator airflow. To analyze the effect of the heat pump optimization, the air-to-air heat pump with optimized evaporator and condenser airflows was implemented for the VRF system. Furthermore, the heat pump of 3 kW used for the experimental measurements was greatly over-sized for the magnitude of the imposed sensible gains. The peak cooling loads rarely exceed 1 kW, even for the VRF system operated under the conventional control for hot summer days in Phoenix. This caused both systems to run at atypically low part-load ratios, especially pronounced for the conventional VRF system. More appropriate sizing of the heat pump was done by reducing its capacity from 3 kW to 1.5 kW. After the heat pump performance for the VRF system was optimized, and the heat pumps for both systems were appropriately sized to 1.5 kW maximum capacity, the LLCS (with 15 cm pitch) electricity savings increased even further to 33.4% and 36% for Atlanta and Phoenix respectively.

3.2.3. Effect of allowable room temperature excursions and precooling

The results suggest that savings are highly sensitive to temperature limits, as shown in Table 2. For example, for the same VRF system temperature of 22°C (fifth row), the LLCS saving potential decreased from 36% (fifth row, third column) to 11.6% (fifth row, forth column) when the allowed LLCS temperature range was reduced from 19-25°C to 19-23°C.

To investigate the impact of precooling, the VRF system was allowed to float between the same temperature limits as the LLCS (19-25°C), and cooling rates were optimized using MPC over the 24-hour planning horizon. The load shift for the VRF system was somewhat more pronounced than for the VAV system, but still not as as pronounced as for the LLCS. As found for the VAV system, the VRF system was maintaining a relatively constant temperature of 25°C through the day, despite being allowed a wide temperature range of 19–25°C. The load shift for the VRF system resulted in an undesirable increase of the total cooling energy; however, even with this load increase, the VRF system under MPC was able to significantly reduce the electricity consumption relative to the system under the conventional control, performing even better than the LLCS (with 15 cm pitch). Compared to the LLCS, the VRF system consumed 18.5% less electricity in Atlanta and 10.6% less in Phoenix; however, with the higher average operative temperature than the LLCS.

Table 2: Electricity consumption savings of LLCS relative to VRF system as a function of VRF setpoints

Electricity savings for Phoenix (%)			
		LLCS	
		19–25 °C	**19–23 °C**
VRF	**21 °C**	44.4	23.2
	22 °C	36.0	11.6
	23 °C	25.4	-3.1

4. Conclusion

The presented research analyzed the performance of the novel energy efficient cooling system, termed the low-lift cooling system (LLCS), that comprises thermally activated building surfaces (TABS) and model predictive control (MPC). The LLCS was analyzed against the VAV system for sensible cooling, ventilation and dehumidification, with the LLCS and VAV system employing a water-to-air heat pump and air-to-air heat pump respectively. The analysis was done for a typical office, across 16 U.S. climates for a typical summer week and also for a 22-week spring and summer period. The VAV system was tested for both MPC and the conventional control, where the conventional control assumed maintaining the constant temperature during the occupied hours and no precooling. LLCS electricity savings were up to 23% relative to the VAV system under the MPC, and up to 50% relative to the conventional VAV system. The savings were achieved through a lower transport energy and a higher average COP, the result of higher evaporating temperatures, lower condensing temperatures, and lower part-load ratios. Savings would be even higher when replacing the VAV direct-expansion unit (air-to-air heat pump) used in this research with the water-to-air heat pump.

Five different DOAS configurations were considered for dehumidification and ventilation purposes. The basic variable-air-volume DOAS assumed the enthlapy recovery wheel, and the heat pump cooled with the outside air. Other options considered more complex configurations with the goal of improving the DOAS heat pump performance. It was shown that the basic DOAS had the lowest energy consumption across different climates. The result also suggests that the DOAS air reheat can significantly increase the energy consumption, and that the LLCS system can benefit from the DOAS delivering a certain amount of sensible cooling. Moreover, for humid climates such as Miami, the LLCS also showed lower oscillations in zone humidity.

The LLCS was also compared against the VRF system for sensible cooling only. The LLCS showed savings of 33% for Atlanta and 36% for Phoenix for a typical summer week. When the same temperature limits and precooling were allowed for the VRF sumption for Atlanta and 11% for Phoenix relative to the LLCS. Although these results imply that the VRF system controlled by MPC and with wider temperature limits could be equally, or even more promising as the LLCS, there are additional practical considerations to take into account. For example, TABS can provide more uniformed cooling since the whole surface acts as a heat exchanger area. Also, the LLCS can reduce the cost of electricity if utility rates favor night operation since the load shifting is much more effective than with the VRF system. On the other hand, the VRF system might be a better solution for retrofits since it does not require a special floor assembly with embedded pipes.

The sensitivity analysis showed that the LLCS savings are highly sensitive to internal loads, spacing between the pipes, heat pump sizing and temperature set points. However, while the increase of internal loads reduced LLCS savings relative to all-air systems, the significance of precooling for the LLCS increased since TABS cooling became predominant, with a greater opportunity for load shifting. Moreover, this work aspired to identify separate benefits of MPC and precooling from the use of TABS. The results showed that precooling did not have a notable effect on the VAV system electricity consumption, and had a somewhat more pronounced effect for the VRF system. It did, however, have a notable effect for the LLCS, especially for high internal loads, with differences in the electricity consumption up to 20%. The analysis also showed that excluding the transport power from the optimization function can significantly influence the decisions of MPC, and also notably increase the total electricity consumption with all-air systems seeing the largest increase.

Future work will extend the analysis presented here by exploring complementary technologies to the LLCS, such as ground source heat pumps and cooling towers, which could improve the performance of the LLCS system even further by lowering the condensing temperatures. Furthermore, another topic that will be explored is the use of a building with TABS for ancillary services to electricity grid operators. A building can provide the ancillary service by shredding its electricity consumption by reducing, or completely turning off the equipment for a certain period of time. It is expected that, compared to buildings with the VAV system or VRF system, buildings with the TABS could provide ancillary services for longer period due to their larger time constant. However, they may not optimally be providing cooling when ancillary services are needed. Finally, the LLCS will be tested in a real building to confirm the findings of this and previous LLCS studies, and also to additionally calibrate the simulation model.

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Table(s) with Caption(s)

Table 1: Latent loads for limiting case when all latent and sensible cooling are provided by DOAS

	$\mathbf{kg}_{water}/\mathbf{h}$	$\mathbf{ACH}_{infiltration}$	**No of people**
Chicago	0.4	1.0	6
Houston	0.5	1.5	7
Las Vegas	0.9	∞	12
Miami	0.5	1.5	7
Phoenix	1.0	4.5	14

Table 2: Electricity consumption savings of LLCS relative to VRF system as a function of VRF setpoints

Electricity savings for Phoenix (%)			
		LLCS	
		19–25 °C	**19–23 °C**
VRF	**21 °C**	44.4	23.2
	22 °C	36.0	11.6
	23 °C	25.4	-3.1

List of figures

Figure 1: DOAS configurations

Figure 2: (a) Heat pump optimization results and (b) third-order polynomial fit (red) to optimization results (black) for supply temperature increase for DOAS system C (shown for evaporator airflow rate $\Ve$ = 0.075 \mcubs). Temperature increase is a function of condenser inlet air temperature $T_{c,air,in}$ and evaporator airflow rate $\Ve$.

Figure 3: Sensible internal gains (red), VAV/TABS cooling rates (blue), and DOAS cooling rates (green) for a typical summer week in Phoenix. Top graph is for VAV system with conventional control; bottom graph for LLCS with DOAS configuration A (LLCS 2).

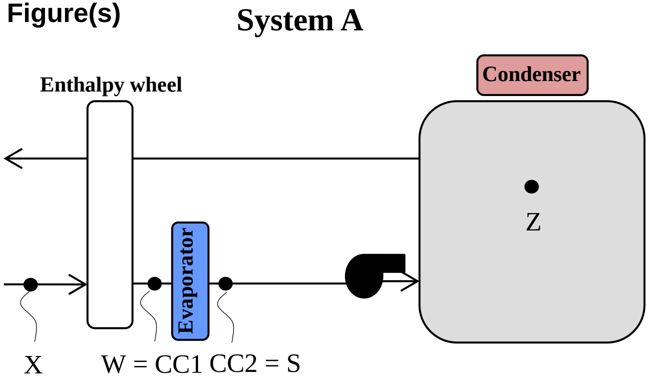
Figure 4: Operative temperatures (red) and temperature limits (black) for a typical summer week in Phoenix. Top graph is for VAV system with conventional control; bottom graph for LLCS with DOAS configuration A (LLCS 2).

Figure 5: Cooling energy delivered by VAV system (1) and LLCS with different DOAS configurations (2-5) for a typical summer week. TABS and VAV cooling is shown with blue bars, and DOAS cooling with green bars.

Figure 6: Electricity consumption for VAV system (1) and LLCS with different DOAS configurations (2-5) for a typical summer week. Electricity for TABS and VAV system heat pump is shown with black bars, for DOAS heat pump with grey bars, and for transport energy with pink bars.

Figure 7: Electricity savings of LLCS system with different DOAS configurations (2-4) relative to conventional VAV system for a typical summer week

Figure(s)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Condenser
Evaporator
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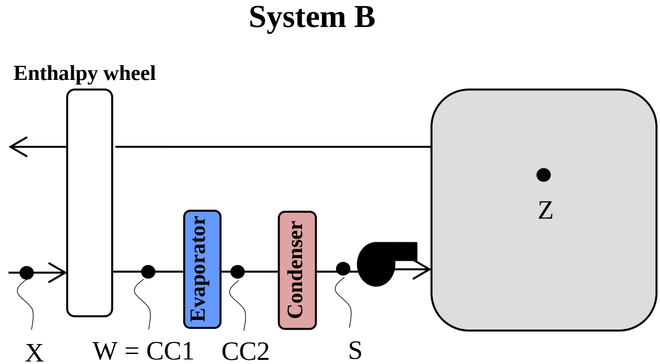
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CC2
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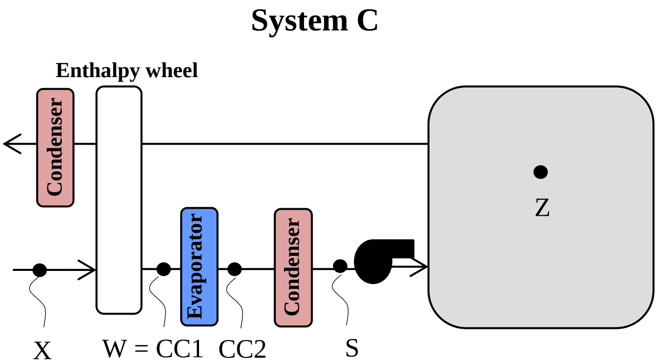
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Evaporator
Condenser
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CC2
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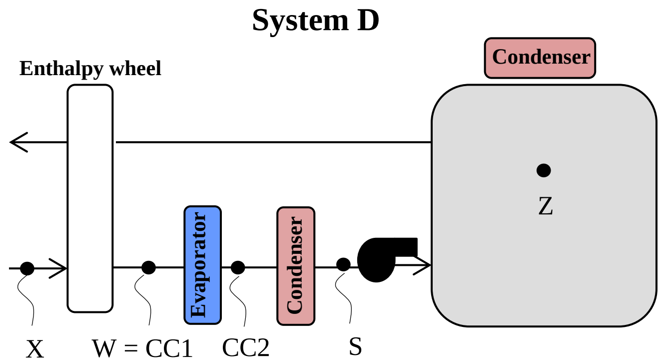
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Condenser
Evaporator
Condenser
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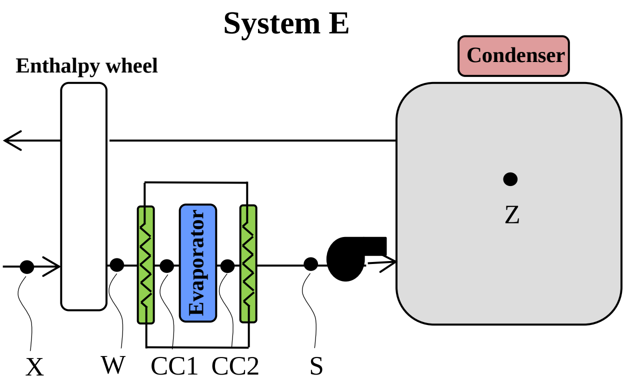
CC2
S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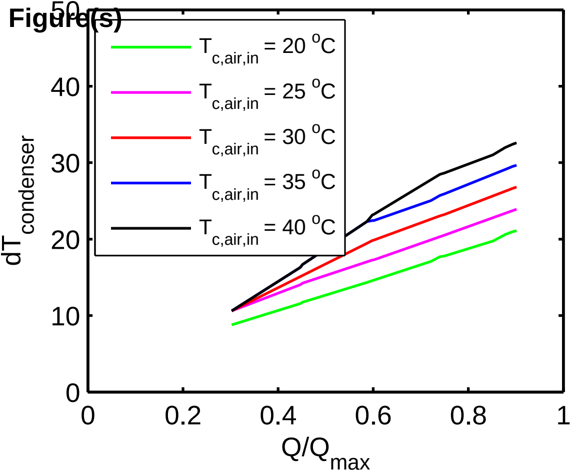
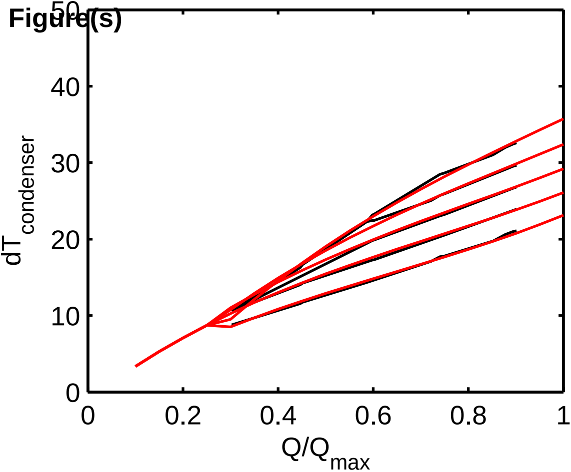
X
W
CC1
CC2
S

Figure(s)



$T_{c,air,in}$ = 20 °C
$T_{c,air,in}$ = 25 °C
$T_{c,air,in}$ = 30 °C
$T_{c,air,in}$ = 35 °C
$T_{c,air,in}$ = 40 °C
$dT_{condenser}$
Q/Q_{max}
0
10
20
30
40
50
0
0.2
0.4
0.6
0.8
1

Figure(s)



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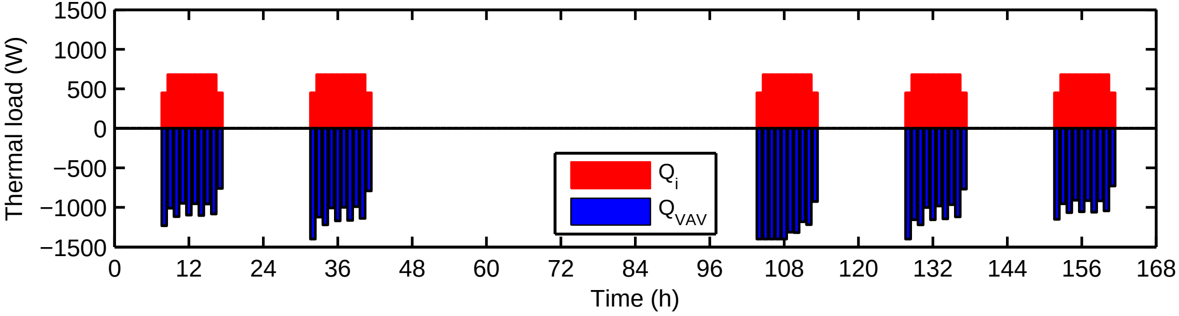
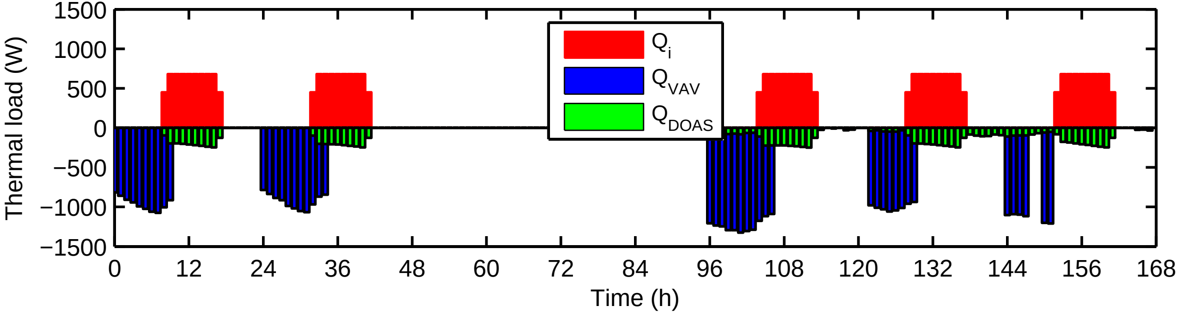
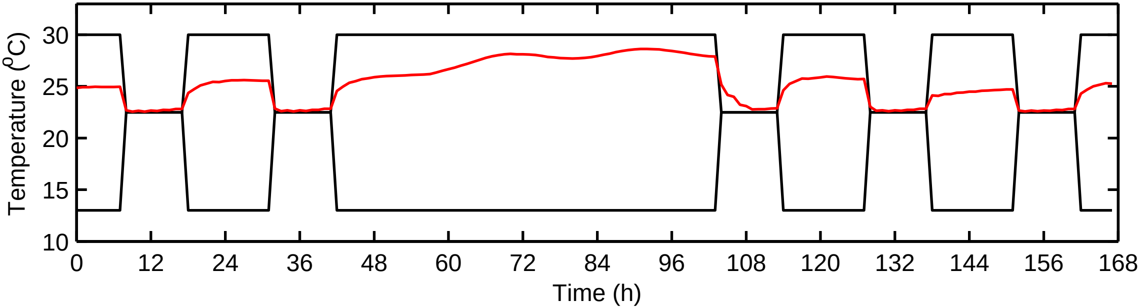


1500
1000
500
0
−500
−1000
−1500
Thermal load (W)
0
12
24
36
48
60
72
84
96
108
120
132
144
156
168
Time (h)
Q_i
Q_{VAV}

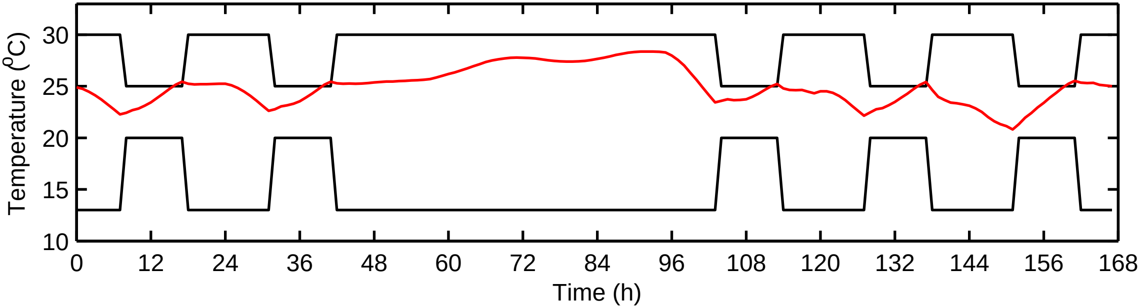


1500
1000
500
0
−500
−1000
−1500
Thermal load (W)
0
12
24
36
48
60
72
84
96
108
120
132
144
156
168
Time (h)
Q_i
Q_{VAV}
Q_{DOAS}

Figure(s)

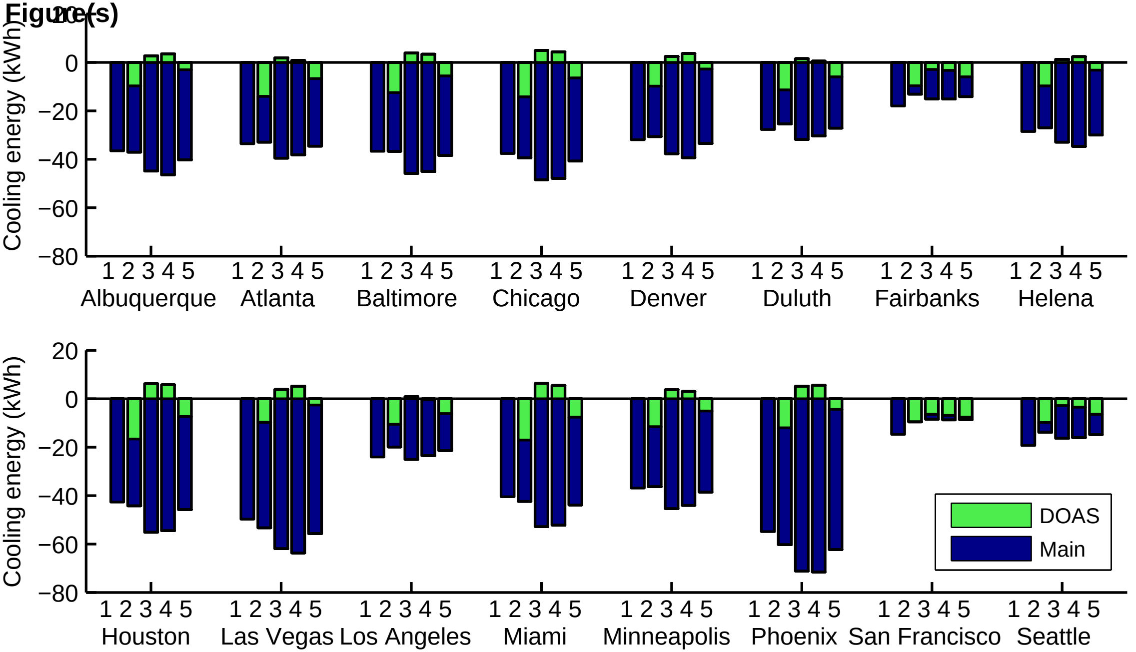


Temperature (^{o}C)
30
25
20
15
10
0
12
24
36
48
60
72
84
96
108
120
132
144
156
168
Time (h)



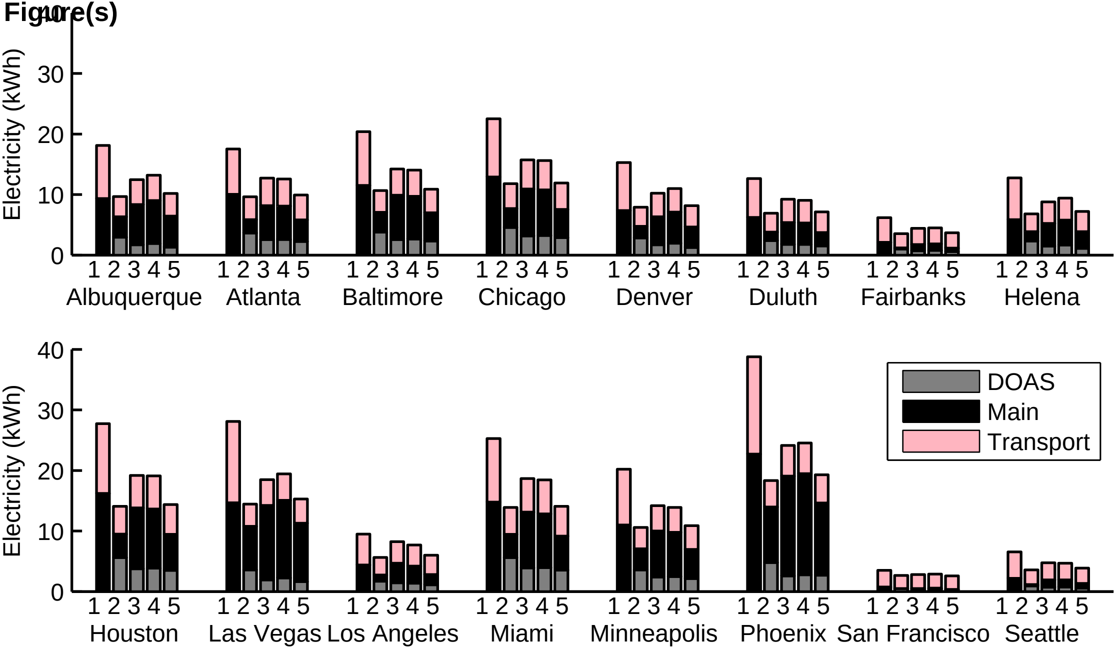
Temperature (^{o}C)
30
25
20
15
10
0
12
24
36
48
60
72
84
96
108
120
132
144
156
168
Time (h)

Figure(s)



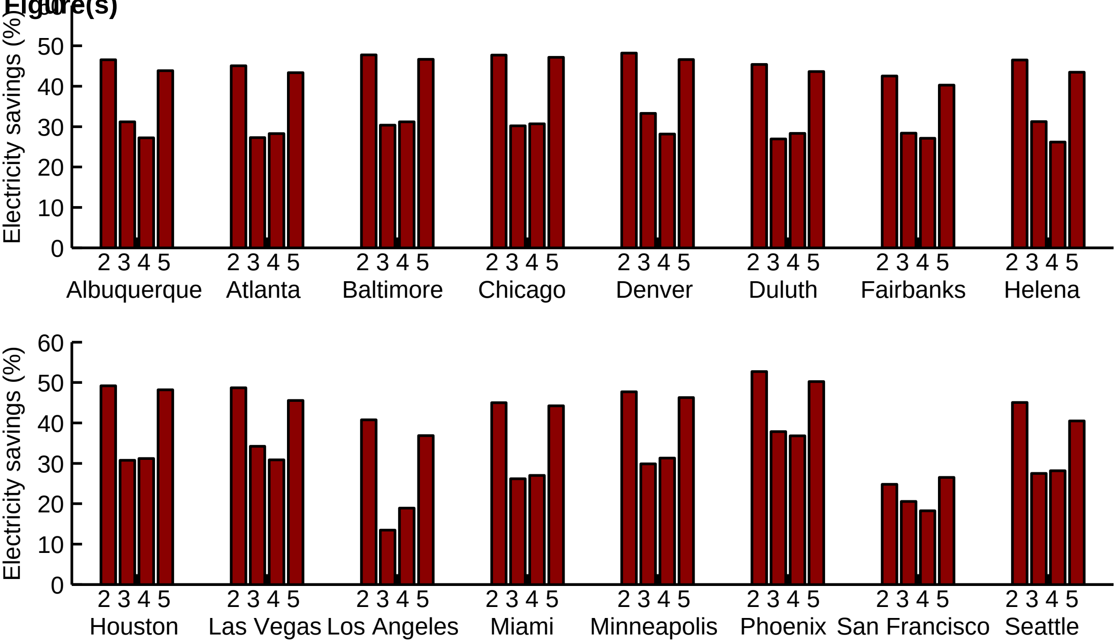
20
0
−20
−40
−60
−80
Cooling energy (kWh)
1 2 3 4 5
Albuquerque
Atlanta
Baltimore
Chicago
Denver
Duluth
Fairbanks
Helena
Houston
Las Vegas
Los Angeles
Miami
Minneapolis
Phoenix
San Francisco
Seattle
DOAS
Main

Figure(s)



Electricity (kWh)
0
10
20
30
40
1 2 3 4 5
Albuquerque
Atlanta
Baltimore
Chicago
Denver
Duluth
Fairbanks
Helena
Electricity (kWh)
Houston
Las Vegas
Los Angeles
Miami
Minneapolis
Phoenix
San Francisco
Seattle
DOAS
Main
Transport

Figure(s)



Electricity savings (%)
60
50
40
30
20
10
0
2 3 4 5
Albuquerque
Atlanta
Baltimore
Chicago
Denver
Duluth
Fairbanks
Helena
Houston
Las Vegas
Los Angeles
Miami
Minneapolis
Phoenix
San Francisco
Seattle

*Highlights (for review)

Highlights

- An advanced cooling system with model predictive control is analyzed.
- Five dehumidification strategies are analyzed.
- The sensitivity of savings on variety of parameters is analyzed.
- The impact of model predictive control is tested for the proposed system, VAV and VRF system.
- Results showed up to 50% electricity savings relative to the conventional VAV system.