SIMULATION ANALYSIS OF BUILDING HUMIDITY CONTROL AND ENERGY CONSUMPTION FOR DIFFERENT SYSTEM CONFIGURATIONS

Chaoqin Zhai[1], Xiaohong Liao[2], Reinhard Radermacher[2], Khee Poh Lam[1], Volker Hartkopf[1]

[1]School of Architecture, Carnegie Mellon University, Pittsburgh, PA 15213, USA
[2]Department of Mechanical Engineering, University of Maryland, College Park, MD 20742, USA

ABSTRACT

In this paper, four different system configurations:

- a conventional stand alone direct expansion (DX) cooling coil,
- an enthalpy wheel together with a DX coil,
- a desiccant wheel followed by an enthalpy wheel together with a DX coil,
- and an enthalpy wheel followed by a desiccant wheel as well as a heat pipe together with a DX coil,

are simulated and compared in terms of their energy consumption and dehumidification performance using two approaches. The results from the first approach, which compares the energy performance of different configurations, while attempting to maintain similar level of indoor temperature and humidity control, show that the desiccant unit has the potential to provide better indoor humidity conditions. The results also indicate that the desiccant unit could be more energy efficient than conventional sub-cooling and reheating approach in order to provide similar level of indoor humidity control. The second approach compares the energy performance as well as resulting indoor thermal conditions, provided that they maintain the same supply air temperature (SAT). In this case, the conventional stand alone cooling coil turns out to have the lowest energy consumption; however it creates the poorest indoor humidity conditions.

INTRODUCTION

In a traditional HVAC system, dehumidification is achieved by cooling the supply air below its dew-point temperature and thus condensing the water on the cooling coil. In order to provide appropriate indoor temperature control, the sub-cooled supply air is usually reheated before it gets into the space. It is believed that this dehumidification method results in extra energy consumption due to the sub-cooling and reheating, and poor indoor humidity control due to the dehumidification performance of the cooling coil. With the requirement of better indoor humidity conditions and the increasing concern of potential mold growth, desiccant and enthalpy wheels are deployed, together or individually, to boost or replace the dehumidification of the cooling coil (Harriman et al., 1999; Popovic et al., 2002). Sometimes the desiccant wheel is also coupled with a sensible heat exchanger (e.g., heat pipe) to transfer some of the heat to the building exhaust air and thus to reduce the temperature of the dry air coming out of the desiccant wheel (Radermacher, 2003; Mumma, 2001). It is commonly suggested that both vapor compression (VC) chillers and direct expansion (DX) cooling coils will perform better due to their higher evaporation temperature. The cooling coil will also have better heat transfer performance due to its dry coil condition. However, extra energy is required by the operation of the desiccant and/or enthalpy wheels, and by pushing air through the wheels. The comparison of energy performance, as well as resulting indoor thermal conditions across various dehumidification systems using different control strategies is the focus of this paper.

METHODOLOGY

An existing office building at the University of Maryland is selected as the study building for analysis. It is a four-story building housing several university administrative offices. The building was constructed in 1991 and measures 39 m in length, 29.3 m in width, and 15.2 m in height, as shown in Figure 1. During occupied hours, there are around 200 employees in the building and they perform a light-duty office work (Marantan et al., 2002).

Principally, the HVAC system consists of two conventional roof top units (RTUs), which are DX air conditioning systems, and variable air volume (VAV) boxes throughout the building. Each RTU serves one zone, which consists of two floors of the building. VAV boxes with electrical terminal reheat are used in the perimeter areas, while no reheat is provided for the interior areas.

Four different system configurations are tested based on the simulation results:

- The base configuration, which consists of a RTU as mentioned above, is illustrated in Figure 2. The outside air is mixed with the return air in the outside air mixer. The mixed air is then pulled through the DX coil, where it is cooled and dehumidified. Finally the air, at a certain temperature and humidity, is supplied to the VAV boxes located in the space. Due to the outside air intake, a certain amount of air has to be exhuasted in order to relief the air pressure in the space.

- An enthalpy wheel, which works between the building relief air and the outside air stream, is deployed to recover the sensible and latent heat from the relief air, as shown in Figure 3. Due to the energy recovery, the inclusion of an enthalpy wheel can presumably reduce the sensible and latent cooling load on the DX coil and thus provide better indoor thermal conditions and building energy efficiency.
- A desiccant wheel is added after the enthalpy wheel to further boost the dehumidification performance of the system, as shown in Figure 4. Due to the adsorption process occuring on the desiccant wheel, the temperature of the air increases as it is dehumidified. As an effective result, the enthalpy of the air might increase, depending on the thermal characteristics of the desiccant wheel. However, when the desiccant wheel is integrated with the enthalpy wheel, the combined system has the potential for enthalpy reduction of the outside air due to the energy recovery capability of the enthalpy wheel. The combined system also has the potential to further improve indoor humidity control due to the inclusion of the desiccant wheel.
- A heat pipe (sensible heat exchanger), which works between the relief air stream from the enthalpy wheel and the outside air stream from the desiccant wheel, is included based on the previous configuration, as illustrated in Figure 5. The dry but hot outside air coming from the desiccant wheel can be cooled by the heat pipe and therefore the sensible cooling load on the DX coil can be reduced.

In each configuration, the conditioned air is distributed into the spaces through VAV boxes, and electrical terminal reheat is provided only in perimeter areas. The HVAC system runs from 5am to 10pm, Monday through Sunday, to maintain the indoor air temperature between 21.1 and 23.3 ^{o}C. The outside air volume is kept the same for each configuration, which is constantly 1.4 m^3/s, roughly 20% of the total supply air flow. The air pressure drops across the enthalpy wheel, desiccant wheel and heat pipe are assumed to be 200, 200 and 150 Pa, respectively, based on the product catalogue and certain engineering experience. Natural gas is used to regenerate the desiccant wheel.

The new generation building energy modeling program EnergyPlus (Version 1.1.0.012) is used for the hourly simulation of the annual operation of different system configurations. Accordingly, the results obtained from the study reflect the system performance and energy consumption across all seasons.

Two approaches are used to compare the performance of these four system configurations. The first approach compares the energy performance of different configurations, provided that they maintain similar level of indoor temperature and humidity conditions. The second approach compares the energy performance, as well as the resulting indoor thermal conditions, provided that they maintain the same supply air temperature (SAT).

While the first approach implies an atypical control strategy, the second is commonly found in engineering practice. The various systems are compared in these two different approaches, with the intention to give some useful information for the design of building control systems, especially in renovation projects. In some cases, enthalpy and/or active desiccant wheels are included in the systems, as part of the renovation, in order to meet the building codes and improve building energy efficiency. But the control systems remain unchanged, still maintaining the same SAT. As a result, the renovated systems might not perform as expected. This paper tries to use building simulation to demonstrate why this situation may happen and how the systems can perform better with some changes in building controls.

These two approaches are detailed in the next two sections.

COMPARISON OF ENERGY PERFORMANCE BASED ON SIMILAR INDOOR THERMAL CONDITIONS

In this approach, the energy performance, in terms of annual electricity and natural gas consumption, of different systems is compared based on similar indoor thermal conditions in relation to temperature and humidity. The main system parameters, such as SAT and maximum outlet air humidity ratio from the desiccant wheel (MOAHR), are adjusted, through trial-and-error, in the simulation in order to obtain the desired indoor thermal conditions.

The indoor humidity control level is indicated by the number of hours the indoor relative humidity (RH) is above 50%. With this indicator, the excess humidity control performance of different systems is reflected, since dehumidification, rather than humidification, is the focus of this paper. The comparison of annual energy consumption is based on approximately 2% of annual system operating hours (i.e., 124 hours) this humidity condition occurs.

In the simulation, SAT and MOAHR are the control variables for the DX cooling coil and the desiccant wheel, respectively. The settings of these two variables, together with the resulting indoor humidity level indicated by the number of hours the RH is above 50%, are shown in Table 1.

As seen from Table 1, in order to obtain similar level of indoor humidity control, the SAT required in the base case is the lowest and that required in the configurations with the desiccant wheel is the highest. The reason for this difference is obvious. In the base configuration, the total dehumidification load is provided by the DX coil. In the second configuration, the outside air is presumably cooled and dried by the enthalpy wheel before it is pulled through the DX coil. The dehumidification performance of the system is boosted by the enthalpy wheel. In the third and fourth configurations, the inclusion of the active desiccant wheel further increases the dehumidification capability of the system. In order to achieve the desired humidity control level, the supply air in the base configuration must be cooled to a much lower temperature than required in the other system configurations, so that the moisture in the air can condense out.

Table 1 also shows that the required settings of SAT and MOAHR in the third and fourth configurations are the same. This can be explained by the fact that the heat pipe reduces the sensible load of the cooling coil, but it has little impact on the latent load.

Figures 6 and 7 plot the comparisons of annual electricity and natural gas consumption required by the regeneration of the desiccant wheel. As seen in Figure 6, the base configuration has the highest electricity consumption for both cooling and terminal reheat due to its lowest SAT. However, it has the lowest power consumption for fan because of the smaller supply air volume and the pressure drops caused by the enthalpy wheel, desiccant wheel and heat pipe in other configurations. Overall, it is the configuration with the highest electricity consumption. The third configuration has the lowest electricity consumption, since it has the highest SAT and requires the lowest amount of terminal reheat. The electricity consumption of the fourth configuration is slightly higher than the third one, though the cooling electricity in the former is lower. The comparision of electricity consumption between the third and fourth configurations indicates the energy benefit and associted penalty resulting from the inclusion of the heat pipe.

Figure 8 plots the comparison of annual total and latent cooling load of the DX coil. As seen from the plot, the base configuration has the highest total and latent cooling load on the DX coil, because it requires the lowest SAT. The sensible and latent load reduction resulting from the enthalpy wheel is obvious from the comparison of the first and second configurations. Compared with the second configuration, the latent load of the DX coil is lower in the third one, because of the dehumidification provided by the desiccant wheel; the lower sensible and total cooling load can be explained by the lower SAT in the third configuration. It is also seen that the third and fourth configurations have the same amount of latent load on the coil, while the fourth configuration has the lowest total cooling load, due to the sensible load reduction of the heat pipe.

In addition, Figure 8 also indicates that the DX coil still provides part of the dehumidification load in the systems with desiccant wheel. This can be explained by the insufficient moisture suppression capacity of the wheel (Liao, 2004). In this system, the desiccant wheel does not have the capability to handle the entire latent load of the space, when it deals with only the outside air, which accounts for roughly 20% of the total supply air flow. It is also noticed that the cooling coil still provides dehumidification when the SAT is above its dew-point tempearure, which can be explained by the non-uniform condition of the air leaving the cooling coil and the on-off cycling of the coil, which is not in sync with the simulation timestep.

As noted above, in configurations with desiccant wheel there is an additional control variable MOAHR. Different combinations of the two control variables could result in similar indoor thermal conditions. For example, in the third configuration, when MOAHR is 10 g/kg dry air and SAT is 17.8 ^{o}C, the number of hours the indoor RH is above 50% is the same as when MOAHR is 12 g/kg dry air and SAT is 16.7 ^{o}C. In terms of energy consumption, the cases with higher SAT are more efficient compared to those with lower SAT, due to less cooling and reheat energy required.

COMPARISON OF PERFORMANCE BASED ON THE SAME SUPPLY AIR TEMPERATURE

In this approach, the energy performance, in terms of annual electricity and natural gas consumption, as well as the resulting indoor RH, of different system configurations is compared, based on the same SAT and MOAHR, where applicable. The settings of SAT and MOAHR, which are determined based on certain engineering practice, together with the number of hours the resulting indoor RH is above 50%, are shown in Table 2.

As seen from this table, the stand alone DX coil provides very poor indoor humidity control, when the SAT is 15.6 ^oC. The inclusion of the enthalpy wheel significantly improves the indoor humidity conditions, reducing the number of hours the indoor RH is above 50% from 1284 to 132. The addition of the desiccant wheel further decreases the number of hours to 99, a 92% improvement compared to the base configuration.

Figures 9 and 10 plot the comparisons of annual electricity and natural gas consumption, respectively, based on SAT 15.6 ^oC and MOAHR 8.5 g/kg dry air.

As indicated in Figures 9 and 10, when the SAT is kept the same, the base configuration is the most energy efficient, while the third and fourth configurations are the least efficient. The amount of electricity required for terminal reheat is approximately the same for different configurations due to the same SAT. Different from the results obtained in the first approach, the cooling electricity required in the second configuration is higher than that in the base configuration, which could be explained by the "negative" contributions made by the enthalpy recovery. Depending on the conditions of the outside air and the building exhaust air, and the requirement for heating or cooling by the building, the deployment of enthalpy recovery could result in extra cooling load of the DX coil. The cooling electricity in the third configuration is even higher, due to the operation of the desiccant wheel as mentioned previously. Compared to the third configuration, the cooling electricity in the fourth one is slightly lower, since the heat pipe reduces the temperature of the dry air coming from the desiccant wheel. The fan power consumption in the fourth configuration is the highest, due to the pressure drops associated with the enthalpy and desiccant wheels, and the heat pipe.

As seen from Figure 10, the amount of natural gas required in the third and fourth configurations is the same, which could be explained by the same amount of dehumidification provided by the desiccant wheel in both configurations.

Figure 11 plots the comparison of total and latent cooling load of the DX coil under the same conditions of the previous two plots. As indicated in this plot, the total cooling load of the DX coil is the lowest in the base configuration and the highest in the third one, due to the reasons presented above.

CONCLUSION

In this paper, the energy and dehumidification performance of four system configurations are simulated and compared using two different approaches: one based on similar indoor thermal conditions, and another based on the same SAT. As observed from the simulation results, control strategies play a significant role in the operation of the mechanical systems. The building control systems have to be modified accordingly when the mechanical systems are renovated, so that the expected energy and thermal performance of the systems could be realized.

The comparison of the four different systems also indicates that both enthalpy and desiccant wheels have the potential to improve the dehumidification performance of the system and thus provide better humidity conditions in the space. In order to obtain the same level of indoor humidity control, the deployment of a desiccant unit together with a cooling coil could be more energy efficient than the conventional sub-cooling and reheating approach. This is due to higher SAT and thus less cooling and reheat energy requirement as in the former configuration. However, if the SAT is kept the same across different systems, the stand alone cooling coil turns out to have the lowest energy consumption, with the poorest indoor humidity conditions.

The results also show that energy recovery could make "negative" contributions to the system energy efficiency, depending on the conditions of outside air and building exhaust air, and the requirement for heating or cooling by the building.

In addition, it is indicated in the results that the extra fan power consumption resulting from the pressure drops of the various system components has a significant impact on the energy performance of the systems with desiccant units. For example, the extra fan power required to push air through the heat pipe could outweigh the benefit in reducing the sensible load of the cooling coil.

ACKNOWLEDGMENT

The authors would like to thank Mr. Matthew Cowie for his valuable help with this paper.

REFERENCES

Harriman L., Czachorski M., Witte M., and Kosar D. 1999. Evaluating Active Desiccant Systems for Ventilating Commercial Buildings, ASHRAE Journal, October, pp. 28-34.

Liao, X. 2004. The Integration of Air-cooled Absorption Chiller in CHP Systems, PhD Thesis, University of Maryland, USA.

Marantan, A., Popovic P., and Radermacher R. 2002. The Potential of CHP Technology in Commercial Buildings – Characterizing the CHP Demonstration Building, ASHRAE Transactions, 108 (1), pp.1025-1031.

Mumma, S. 2001. Dedicated Outdoor Air-Dual Wheel System Control Requirements, ASHRAE Transactions, 107 (1), pp.147-155.

Popovic P., Marantan A., Radermacher R., and Garland P. 2002. Integration of Microturbine with Single-Effect Exhaust-Driven Absorption Chiller and Solid Wheel Desiccant System, ASHRAE Transactions, 108 (2), pp. 660-669.

Radermacher, R.. 2003. The Impact of Waste Heat Activated Desiccant Systems on Buildings, International Conference on Building Systems and Facilities Management, Singapore.

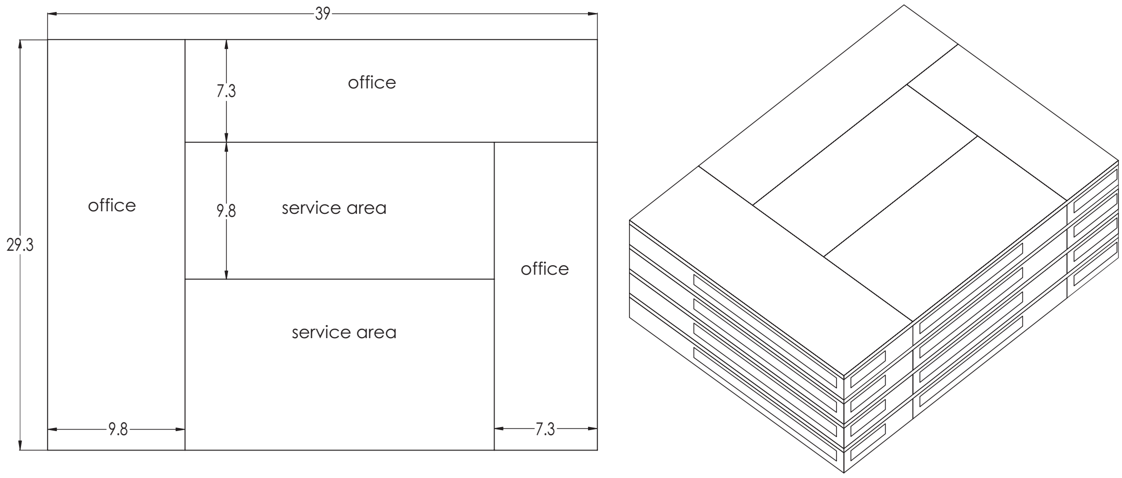

Figure 1 The study building (unit: m)

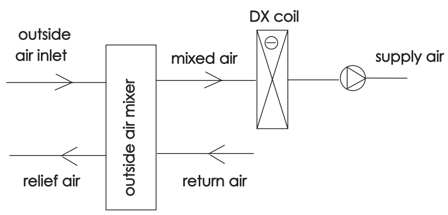

Figure 2 Air loop diagram of base configuration

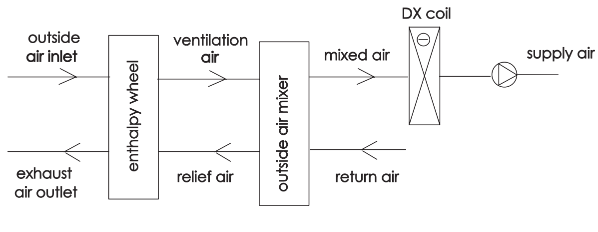

Figure 3 Air loop diagram of enthalpy wheel system

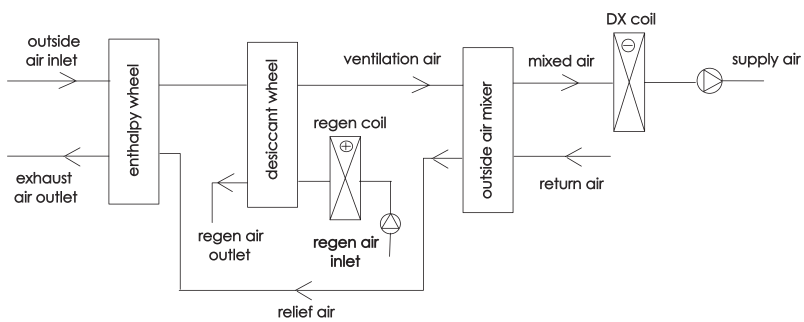

Figure 4 Air loop diagram of enthalpy + desiccant wheel system

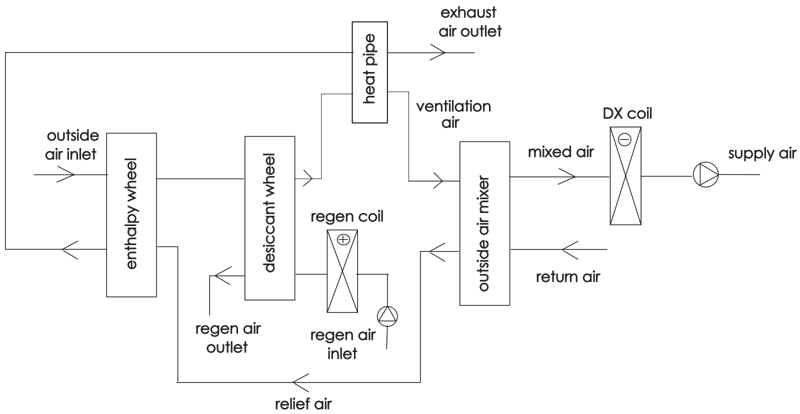

Figure 5 Air loop diagram of enthalpy + desiccant wheel + heat pipe system

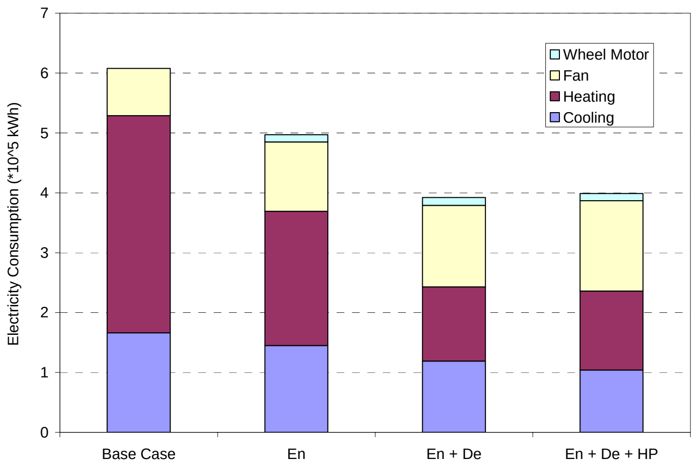

Figure 6 Comparison of annual electricity consumption based on the same indoor thermal conditions

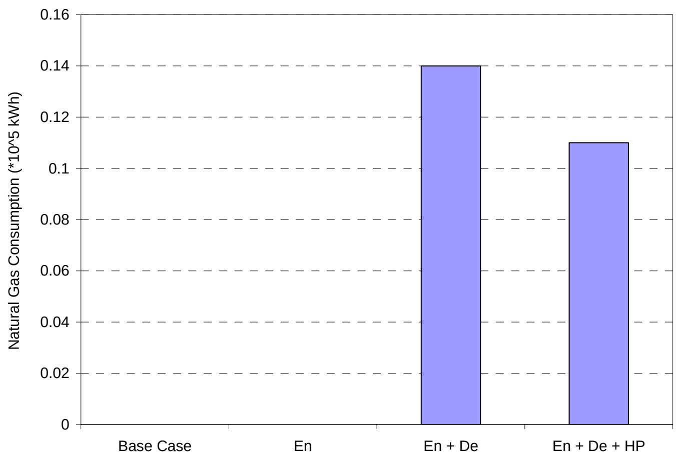

Figure 7 Comparison of annual gas consumption based on the same indoor thermal conditions

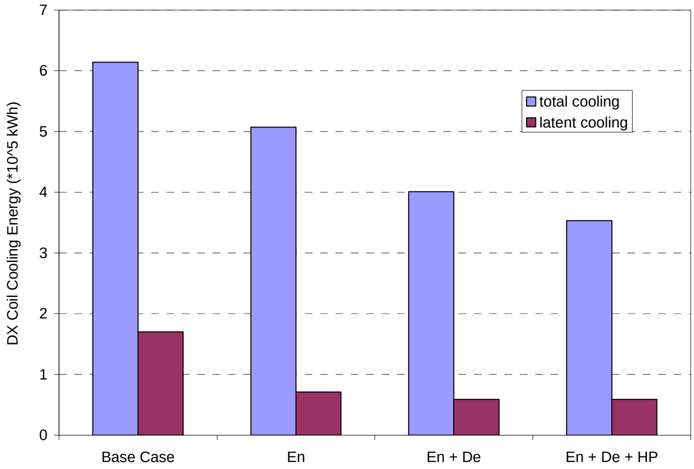

Figure 8 Comparison of annual DX coil cooling load based on the same indoor thermal conditions

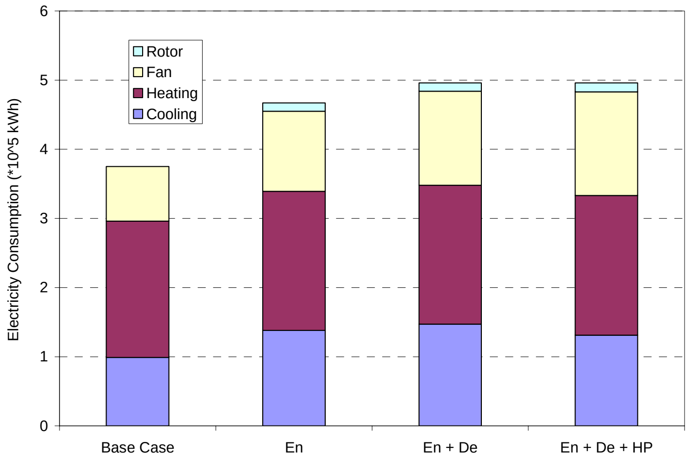

Figure 9 Comparison of annual electricity consumption based on the same SAT

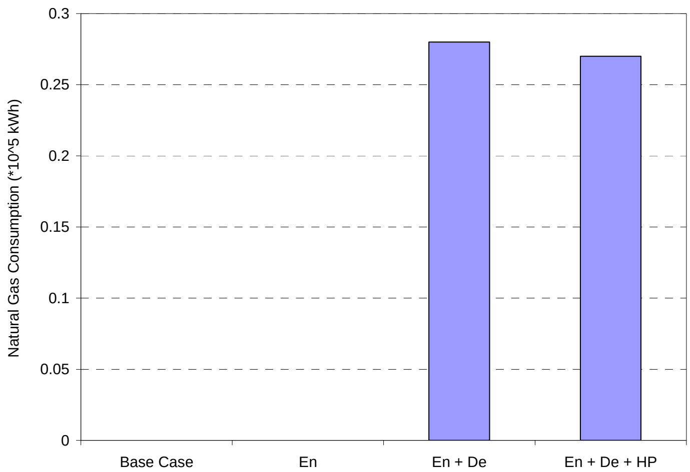

Figure 10 Comparison of annual gas consumption based on the same SAT

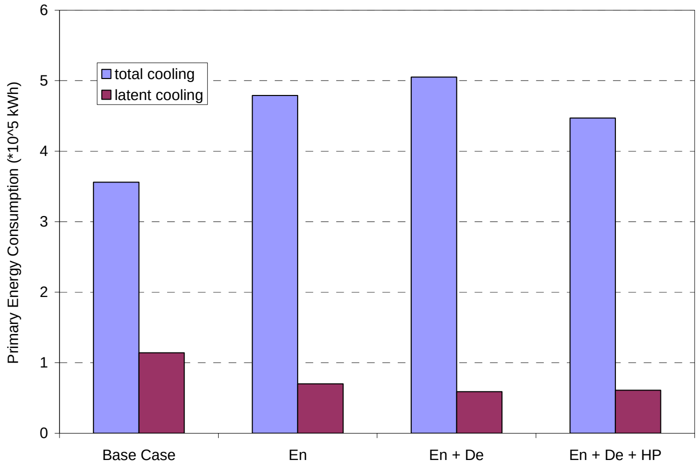

Figure 11 Comparison of annual DX coil cooling load based on the same SAT

Table 1 Settings of control variables and resulting indoor humidity conditions in the first approach

Configuration	SAT (^oC)	MOAHR (g/kg dry air)	Indoor humidity control level (hrs)
1. base case (stand alone DX coil)	10	--	147
2. enthalpy wheel + DX coil	15	--	125
3. enthalpy + desiccant wheel + DX coil	17.8	10	122
4. enthalpy + desiccant wheel + heat pipe + DX coil	17.8	10	125

Table 2 Settings of control variables and resulting indoor humidity conditions in the second approach

Configuration	SAT (^oC)	MOAHR (g/kg dry air)	Indoor humidity control level (hrs)
1. base case (stand alone DX coil)	15.6	--	1284
2. enthalpy wheel + DX coil	15.6	--	132
3. enthalpy + desiccant wheel + DX coil	15.6	8.5	99
4. enthalpy + desiccant wheel + heat pipe + DX coil	15.6	8.5	99