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Published in:
Proceedings of the 15th IBPSA Conference

DOI (link to publication from Publisher):
10.26868/25222708.2017.219

Publication date:
2017

Document Version
Publisher's PDF, also known as Version of record

Analysis of control strategies for a novel HVAC system equipped with a room-temperature water loop

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Abstract

This article presents a simulation-based study that investigated four control strategies to regulate a novel two-pipe system for office buildings. The main characteristic of the system is its ability to provide simultaneous heating and cooling by operating a single water circuit with supply temperature of about 22°C. The study was conducted by considering a typical office building model in two different climates, Chicago (US) and Copenhagen (Denmark). Simulations were performed in Dymola, a commercial simulation environment for Modelica models.

The four control strategies ranged from linear SISO (single input, single output) to PI (proportional-integral) feedback controllers with detailed information about the indoor climate in the building.

Results showed that all the four strategies were able to maintain comfortable levels of indoor air temperatures in the zones. With regard to the annual energy use, savings of about 7-10% were achieved when comparing PI feedback controllers with SISO strategies.

Introduction

Anthropogenic greenhouse gas (GHG) emissions have increased since the pre-industrial era, driven largely by economic and population growth. According to the Intergovernmental Panel on Climate Change (IPCC) their effects have been the dominant cause of the observed warming of land and ocean surfaces since the mid-20th century. Continued emission of GHG will cause further warming and long-lasting changes in all components of the climate system, increasing the likelihood of severe, pervasive and irreversible impacts for people and ecosystems (IPCC, 2014).

Building energy-related activities are responsible for the 19% of GHG emissions worldwide. This energy use and related emissions may double or potentially even triple by mid-century due to the increased access for billions of people in developing countries to adequate housing, electricity, and improved cooking facilities. Therefore, research in the field of energy efficiency in buildings is crucial in order to reduce the global GHG emissions and mitigate the effects of the climate change.

Low-exergy building energy systems are defined as systems that provide heating and cooling at a temperatures close to the room temperature (Hepbasli, 2012). This allows the employment of low-valued energy, which can be delivered by sustainable energy sources such as waste heat, river/lake water, solar energy, geothermal applications and heat pumps, with a high coefficient of performance. Therefore, the use of low-exergy systems can reduce the environmental impact of buildings by providing energy with low GHG emissions.

Various studies about low-temperature heating and high-temperature cooling systems have been carried out in the past years (Hesaraki and Holmberg, 2013; Hasan et al., 2009; Lehmann et al., 2007; Zhao et al., 2013). Generally, a minimum temperature of about 30°C is used in low-temperature heating applications, and a maximum temperature of about 18°C is found in high-temperature cooling applications.

By lowering the hot water temperature even more in heating systems, and raising the chilled water temperature even more in cooling systems, further advantages could be achieved in terms of maximizing the use of sustainable energy sources.

The limiting case of lowering heating water temperature, and raising cooling water temperature, can be seen as a system where heating and cooling circuits have the same supply water temperature, somewhere between the indoor comfort limits of 20°C and 25°C.

The design, modeling and energy savings potential of a system with such a characteristic were investigated in previous works (Maccarini et al., 2014, 2016). In this article, focus is given to the analysis of four different control strategies for the regulation of the system in two different locations, Chicago and Copenhagen. The study was carried out by using Dymola, a simulation environment for Modelica models.

Concept

Existing large commercial buildings may require simultaneous heating and cooling demand. Interior zones of a building tend to overheat due to waste heat generated by internal factors (people, lighting and equipment), while perimeter zones require heat-

https://doi.org/10.26868/25222708.2017.219

ing due to heat losses through windows, walls and infiltration.
To deal with this situation, the distribution water circuit of conventional systems is typically designed with two separated hydronic loops (four-pipe configuration), which includes two supply and two return pipes. As a consequence, some zones can receive cold water while other zones receive hot water, meaning that heating and cooling can be provided simultaneously.
The novel HVAC system is able to handle heating and cooling loads simultaneously by operating only one hydronic loop (two-pipe configuration) that operates water that is near the room temperature. Active beams are used as terminal units. Figure 1 shows the schematic diagram of the two-pipe system. This configuration shows two main benefits:

- Waste heat from warm zones in a building can be transferred to cold rooms, reducing the total energy use.
- The use of water temperatures of about 22°C in the hydronic circuit facilitates the integration of sustainable energy sources. Therefore, savings in terms of primary energy use can be obtained with a consequent reduction of GHG emissions.

An added benefit of operating a system with water temperatures close to room temperature is that its rate of heat transfer is very sensitive to changes in room temperature. This is known as a self-regulation effect. This effect can be exploited to simplify the control strategy, as individual room-temperature feedback controllers are no longer required (Afjei et al., 1996).

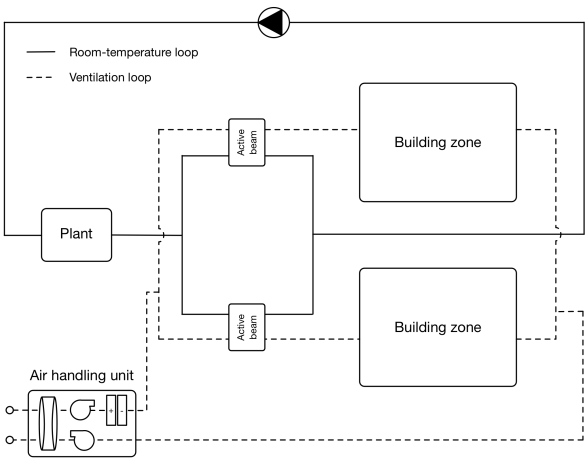

Figure 1: Diagram of the two-pipe system.

Methodologies

Building model

The simulation study was performed through the integration of the two-pipe system in a reference three-storey building model. The geometry of the building model is illustrated in Figure 2. Each floor consists of four perimeter thermal zones and one interior thermal zone. The total floor area is 4980 m^2 with an aspect ratio of 1.5.

This fifteen-zone model is representative of the medium office building prototype, as described in the report, U.S. Department of Energy Commercial Reference Building Models of the National Building Stock (Deru et al., 2011). The report characterizes 16 prototype buildings for 16 climate zones covering the majority of the US commercial building stock. These building models have been developed to serve as a starting point for energy efficiency research, as they represent fairly realistic buildings and typical construction practices.

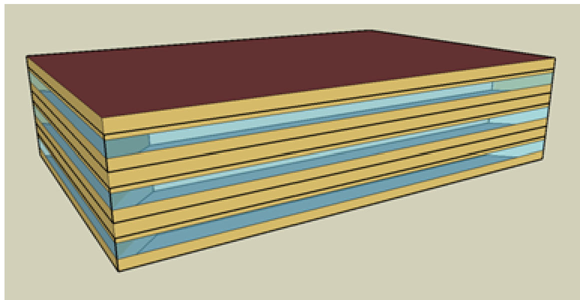

Figure 2: Geometry of the reference building model.

Table 1 illustrates the thermal properties of the building elements. These are compliant with ASHRAE 90.1 2010 standard. The roof and the external walls were exposed to the outdoor environment while the floor to a set of monthly average ground temperatures. The glazing surfaces were evenly distributed along all facades with a window-to-wall ratio of 0.33 and consist of double pane windows with solar control properties. In accordance with the prototype office building models, no shading devices were applied to the windows. An average infiltration rate of 0.1 ACH was selected. Internal gains due to people, lighting and equipment were, respectively, 6.5 W/m^2, 9.7 W/m^2 and 8 W/m^2. Working hours were assumed to be between 8 AM and 5 PM on weekdays.

Inter-zone air flow was modelled by placing doors into the building. The door model allows for bi-directional air flows through adjacent zones. The bi-directional air flow is modeled based on the differences in static pressure between adjacent rooms at a reference height plus the difference in static pressure across the door height as a function of the difference in air density (Wetter, 2006).

As previously mentioned, simulations were run for two climate locations, Chicago and Copenhagen. Figure 3 shows the yearly outdoor air temperature profiles according to the corresponding weather files (EPW files).

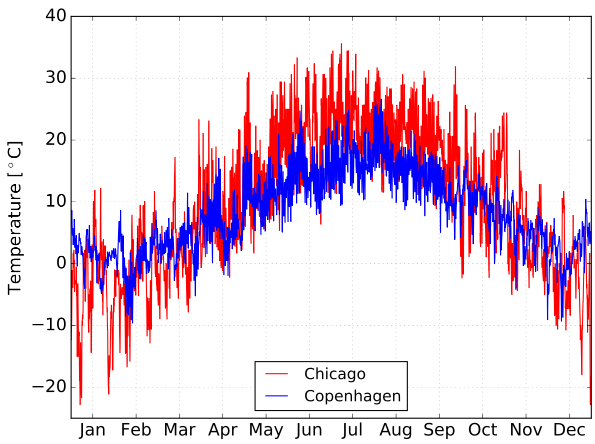

Figure 3: Yearly outdoor air temperature profiles.

Table 1: Thermal properties of the building elements for ASHRAE 90.1 2010 prototype office model.

Building element	**U-value** [W/m^{2}K]
External Walls	0.36
Roof	0.27
Floor	1.82
Windows	2.6 (SHGC=0.4)

System model

The layout of the Modelica model developed in Dymola is shown in Figure 4. It includes the building model (previously described) and the HVAC system model. Models from the Modelica Buildings library were used (Wetter et al., 2014).

The HVAC system consists of a water loop for space heating and space cooling and an air loop for ventilation.

A central plant provides thermal energy to the water circulating in the loop. The plant includes a reversible air-to-water heat pump and a dry cooler to take advantage of free cooling conditions.

The ventilation loop is a constant air volume (CAV) system with constant supply air temperature of 20°C. Air is delivered to the terminal units by an air handling unit (AHU) comprising supply and return fans, heating and cooling coils and a heat recovery unit. The heating and cooling coils are served by secondary water loops connected, respectively, to a heat pump and a chiller.

The system runs at full capacity during operating hours (between 6 AM and 10 PM). During night operation, the water loop is turned off, while the air mass flow rate delivered through the ventilation system is reduced by 50%.

As previously mentioned, active beams were used as terminal units. A comprehensive description and validation of the active beam model used in this work is provided in Maccarini et al. (2015). The model encapsulates empirical equations derived by a novel active beam terminal unit that operates with low-temperature heating and high-temperature cooling systems.

Heating and cooling peak loads of the building were pre-calculated by running dedicated simulations in Dymola. Two sizing calculations were performed in respect to the climate locations. The number of active beams placed in each thermal zone was a consequence of the pre-calculated loads, chosen design parameters and ventilation requirements. Table 2 shows the design parameters for the heating and cooling mode. The values of the these parameters were selected according to the REHVA chilled beam application guidebook (Virta et al., 2004) and manufacturers recommendations. The ventilation requirements were selected according to the Danish standard DS/EN 15251 (REHVA, 2007).

At the nominal flow rate, the total pressure drop in the water loop was assumed to be 35 kPa, and the total pressure drop in the ventilation loop was assumed to be 500 Pa. For lower flow rates, e.g. during night operation, these values are reduced, as the simulation model computes flow friction as a function of the flow rate.

Table 2: Design parameters. These values refer to a single active beam unit.

	Cooling	**Heating**
Room air temperature	24°C	20°C
Primary air temperature	20°C	20°C
Supply water temperature	20°C	23°C
Nozzle pressure drop	100 Pa	100 Pa
Primary air mass flow rate	0.03 kg/s	0.03 kg/s
Water mass flow rate	0.04 kg/s	0.04 kg/s
Length	3 m	3 m
Total capacity	520 W	297 W

Control strategies

Since the ventilation system is designed to have constant air mass flow rate and constant supply temperature, the indoor climate in the building is regulated by acting on the room-temperature water loop.

Four different control strategies for regulating the water loop were investigated. Their level of complexity spans from linear SISO with standard inputs to PI feedback controllers with detailed information about the disturbances acting on the building. Table 3 summarizes the control strategies analyzed. These were named according their most peculiar features.

All the four strategies were designed to operate with a supply water temperature range between 20°C and 23°C. In winter, a temperature of 23°C would be able to heat a perimeter zone at 20°C and cool an interior zone at 25°C. In summer, both interior and perimeter zones generally require cooling. Since there is no direct control on each zone, a minimum temperature of 20°C ensures that the air temperature in each zone will be at least at around 20°C, even if the zone becomes unoccupied.

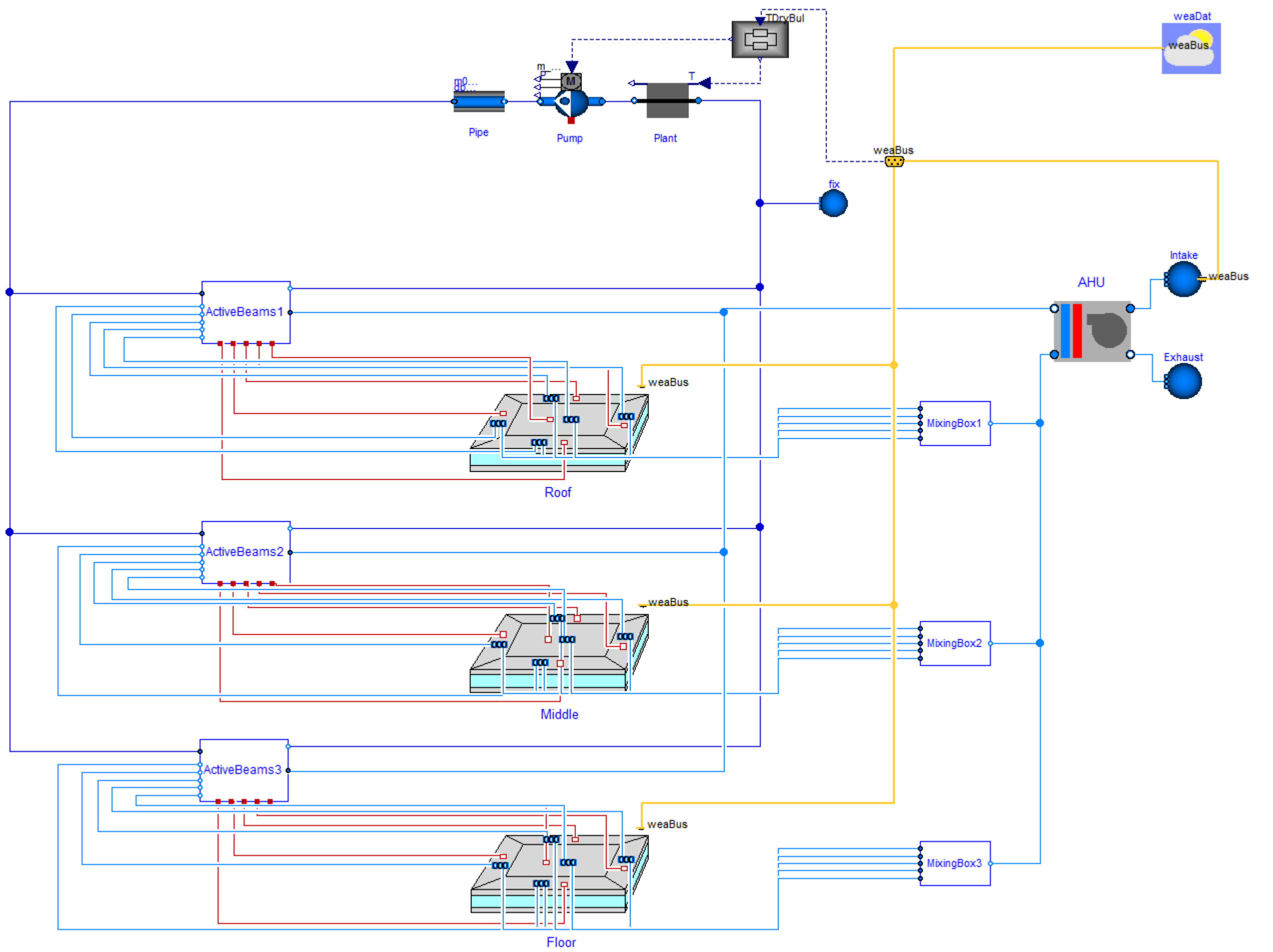

Figure 4: Modelica model developed in Dymola. Light-blue lines represent air streams, dark-blue lines represent water streams, red lines represent convective heat exchange, and dashed blue lines represent control signals.

The OATS presents constant water mass flow rate and variable supply water temperature. The value of the supply water temperature is function of the outdoor air temperature, as shown in Figure 5. The maximum supply water temperature of 23°C was set in correspondence of the coldest design outdoor air temperature. The minimum supply water temperature of 20°C was set in correspondence of the warmest design outdoor air temperature.

As above, also the EATS presents constant water mass flow rate and variable supply water temperature. However, in this case, the supply water temperature is set based on the exhaust air temperature, as illustrated in Figure 6. This represents an average of the air temperatures in the zones. The maximum supply water temperature of 23°C was set in correspondence of an exhaust air temperature of 20°, which is the heating design room air temperature . The minimum supply water temperature of 20°C was set in correspondence of an exhaust air temperature of 24°C, which is the cooling design room air temperature.

The linear correlation for the OATS and the EATS was chosen since this form is in accordance to how

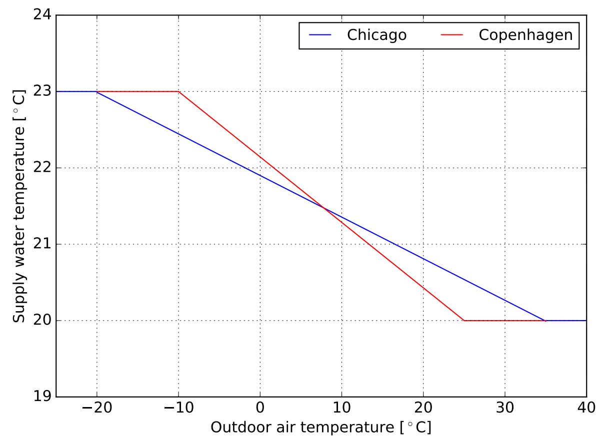

Figure 5: Supply water temperature vs. outdoor air temperature.

the considered input signals affect the heating and cooling demand in the building.

The FWTS implements temperature sensors in each zone. A constant water mass flow rate is circulated in the loop, while the supply water temperature is adjusted based on the actual room air temperatures.

Table 3: Summary of the control strategies analyzed.

Name	Water mass flow rate	Supply water temperature
Outdoor air temperature strategy (OATS)	Constant	$f(T_{out})$
Exhaust air temperature strategy (EATS)	Constant	$f(T_{exh})$
Feedback water temperature strategy (FWTS)	Constant	PI controller
Feedback water flow strategy (FWFS)	PI controller	$f(T_{out})$

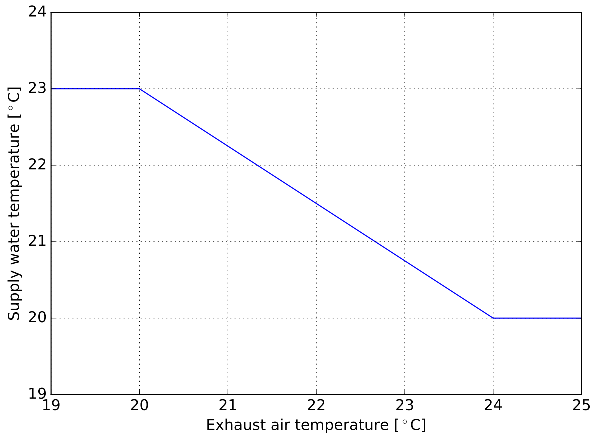

Figure 6: Supply water temperature vs. exhaust air temperature.

The supply water temperature can be expressed by the following equation:

$$T_{sup} = T_{ret} + k_{hea} - k_{coo} \tag{1}$$

Where T_{ret} is the return water temperature and k_{hea} and k_{coo} are offsets able to adjust the return water temperature based on current air temperatures in the rooms and set-point temperatures.

The controller is fed by the signals of actual air temperatures in the fifteen zones and return water temperature. The minimum air temperature among the fifteen zones is an input for the PI heating controller, where it is compared with the heating temperature set-point. If the minimum air temperature is above the set-point, k_{hea} is equal to 0. Otherwise, the PI controller evaluates the value of k_{hea} to be added to T_{ret} to meet the heating set-point.

The maximum air temperature among the fifteen zones is an input for the PI cooling controller, where it is compared with the cooling temperature set-point. If the maximum air temperature is below the set-point, k_{coo} is equal to 0. Otherwise, the PI controller evaluates the value of k_{coo} to be deducted from T_{ret} to meet the cooling set-point. As a consequence, whenever all the zones in the building present air temperatures within the heating and cooling set-point range, k_{hea} and k_{coo} are equal to 0 and, therefore, the supply water temperature is set equal to the return water temperature, requiring for no energy in the thermal plant.

The FWFS presents variable water mass flow rate and supply water temperature adjusted using the relationship described in Figure 5. Due to the absence of valves at zone levels, the water mass flow rate must be prescribed at system level. The actual water mass flow rate can be expressed by the formula

$$\dot{m} = k * \dot{m}_{nom} \tag{2}$$

where $\dot{m}$ is the actual water mass flow rate, $\dot{m}_{nom}$ is the nominal mass flow rate and k is coefficient in the range of 0.25 and 1. The value of 0.25 was chosen as the minimum water mass flow rate allowed in the circuit during operation hours. As in the FWTS, the controller is fed by the signals of the actual air temperatures in the fifteen zones. Whenever the minimum and the maximum zone air temperature are within the heating and cooling set-points, k is equal to 0.25, and therefore, the minimum amount of energy is required. Otherwise, the value of k is increased just enough to meet the set-points.

Results and Discussion

In this work, simulations were used to investigate four different control strategies in two climate locations for the regulation of a room-temperature HVAC system. Two control strategies were refereed as simple strategies due to the need of only a single input signal and the absence of temperature sensors in the building zones. The other two control strategies were refereed as complex strategies due to the need of temperature sensors in each building zone and the use of PI controllers.

The results in the following text were calculated in terms of annual electricity use for the air-to-water heat pump (space heating and space cooling), heating and cooling AHU coils, fans and pumps.

In Figure 7 and 8 , the energy use of the two-pipe system is presented for the Chicago and Copenhagen climate respectively. Generally, due to more extreme outdoor conditions (see Figure 3) the two-pipe system in Chicago requires more energy than the two-pipe system in Copenhagen.

The ranking provided by sorting the alternative strategies according to their energy performance was consistent for the two climate locations. That is, the FWTS has the highest energy performance with annual electricity energy use of 25.2 and 11.3 kWh/m^2 respectively for Chicago and Copenhagen, subsequently followed by the FWFS (25.4

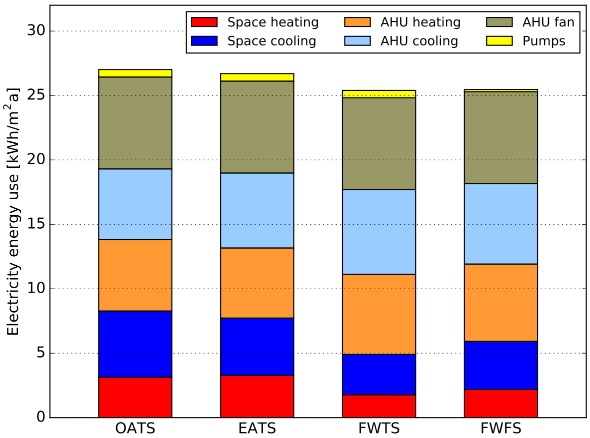

Figure 7: Annual electricity use (Chicago).

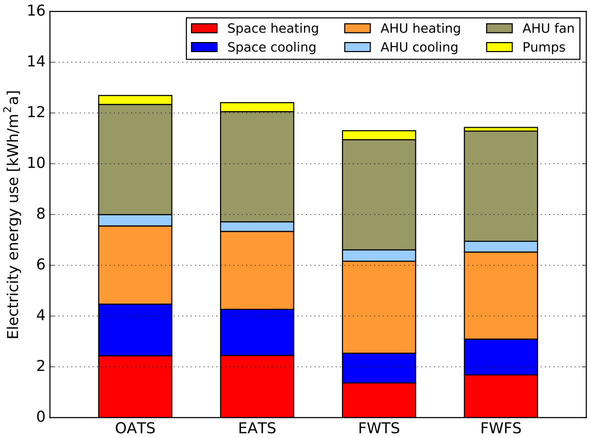

Figure 8: Annual electricity use (Copenhagen).

and 11.4 kWh/m^2 respectively), EATS (26.7 and 12.5 kWh/m^2 respectively) and OATS (27 and 12.6 kWh/m^2 respectively).

No significant difference is noticed when comparing the two SISO strategies. The relative difference is approximately 1% for both climates. Similarly, the two PI feedback control strategies present analogous values of total annual electricity use. Also in this case, the relative difference is approximately 1% for both climates.

Larger differences are noticed between the SISO strategies and the PI feedback strategies. In particular, when comparing the most efficient strategy (FWTS) with the least efficient strategy (OATS), energy savings of approximately 7% and 10% were achieved, respectively, for Chicago and Copenhagen.

As expected, since all the control strategies were assumed to deliver the same constant amount of air mass flow rate for ventilation, the energy use for fans is equal among the four strategies.

The energy use for pumps is equal for the OATS, EATS and FWTS as these strategies have constant water mass flow rate. The FWFS presents a lower value of energy use for pumps as this strategy allowed modulating the water mass flow rate in the loop according to the building loads.

When comparing the two SISO control strategies with the two PI feedback strategies, it is worth highlighting that the former present higher energy use for space heating and cooling, but lower energy use for AHU heating and cooling coils. This can be explained by illustrating the indoor air temperatures for a typical winter and summer day in the two climates.

Figure 9 illustrates the minimum indoor air temperature for a typical winter day in Chicago and Copenhagen. This represents the minimum value of indoor air temperature among all the fifteen zones. It is noticed that, due to the presence of PI feedback controllers, the FWTS and FWFS can strictly meet the heating set-point of 20°C. Conversely, the OATS and EATS present indoor air temperature higher than 20°C. As a consequence, the FWTS and FWFS are able to minimize the heating energy required by the thermal water plant. However, more energy is required in the AHU heating coil due to colder exhaust air temperature from the zones, and as a consequence, lower heat exchange in the heat recovery unit.

A similar explanation can be provided for the cooling season. Figure 10 shows the maximum indoor air temperature for a typical summer day. Also in this case, the FWTS and FWFS achieved indoor air temperature matching the set-point (24°C). On the other hand, colder indoor air temperature are noticed for the OATS and EATS. Therefore, the use of PI controllers lead to lower cooling energy in the thermal plant, but higher energy use for the AHU cooling coil.

Overall, the ability of the PI feedback strategies to set a value of the supply water temperature (FWTS) or water mass flow rate (FWFS) that minimizes the energy required by the reversible air-to-water heat pump allowed reducing the total electricity use.

Figure 9 and 10 also show that all the four control strategies analyzed in this work were able to maintain comfortable levels of indoor air temperatures in both winter and summer conditions.

Conclusions

In this work, different control strategies to regulate the operation of a novel two-pipe HVAC system were investigated through simulations with Modelica. A total of four control strategies were considered, spanning from simple SISO to more complex PI feedback controllers.

The behavior of the different control strategies was evaluated for a reference office building model in two different climates, Chicago (US) and Copenhagen (Denmark). Standard thermal properties of the building envelope and internal heat gains were selected.

Results from the simulations showed that all the four control strategies were able to provide comfortable

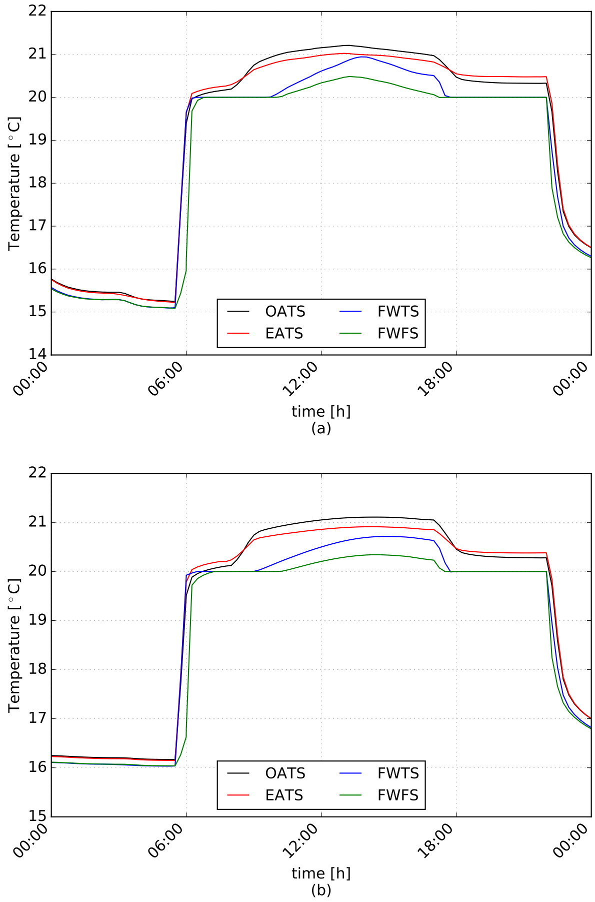

Figure 9: Minimum air temperature among the building zones, Chicago (a) and Copenhagen (b).

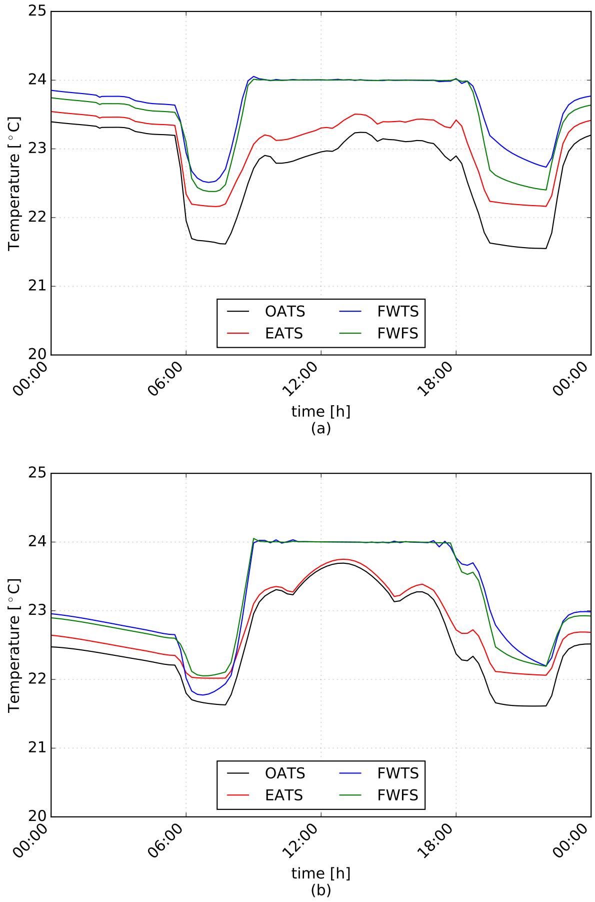

Figure 10: Maximum air temperature among the building zones, Chicago (a) and Copenhagen (b).

levels of indoor air temperature in the building for both climates.

Annual energy savings of approximately 7% and 10% were achieved respectively for Chicago and Copenhagen when comparing the most efficient control strategy (FWTS) with the least efficient control strategy (OATS).

In conclusion, both linear SISO and PI feedback control systems can be used to regulate the room-temperature water loop. PI feedback control strategies require temperature sensors in each building zone, leading to higher installation cost. However, air temperatures in the zones can be kept closer to desired set-points and the operating cost for energy use is reduced. On the other hand, SISO strategies are simple and cheap, but the annual operating cost for energy resulted slightly higher.

A first real-life implementation of the two-pipe system is currently under monitoring in an office building located in Jönköping (Sweden). At present, the system operates with the OATS described in this work. On-site measurements of indoor climate and energy performance are future tasks.

Acknowledgement

This study was financially supported by ELFORSK, a research and development program administrated by the Danish Energy Association.
This research emerged from the Annex 60 project, an international project conducted under the umbrella of the International Energy Agency (IEA) within the Energy in Buildings and Communities (EBC) Programme. Annex 60 will develop and demonstrate new generation computational tools for building and community energy systems based on Modelica, Functional Mockup Interface and BIM standards.

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