Fourth National Conference of IBPSA-USA
New York City, New York
August 11 – 13, 2010

A NEW MODEL FOR CALCULATING THE CONVECTIVE AND RADIANT IMPACT OF RADIATORS AND BASEBOARDS IN ENERGYPLUS

Daeho Kang[1*] and Richard K. Strand[1]
University of Illinois at Urbana-Champaign, Champaign, IL
([*]Corresponding author email address: dkang21@illinois.edu)

ABSTRACT

Baseboards and radiators are by their very nature devices that deliver heat to a space via both convection and radiation. While it is possible to model the convective effects of these devices fairly easily in many whole building energy simulation programs, the presence of a radiation source from HVAC equipment is an impossible task for most programs since they do not use strict surface heat balance techniques derived from the First Law of Thermodynamics. One program that does at least have the capacity to deal with radiant heat transfer from HVAC equipment is EnergyPlus. EnergyPlus already has models for low and high temperature radiant systems that account for the radiation impact of these systems on each surface within the space. In the past, the radiator and baseboard model in EnergyPlus has been purely convective. This paper details the capabilities of the new radiant/convective baseboard models in EnergyPlus and provides a comparison to the convective-only model that to date has been the only model available in the program.

INTRODUCTION

Radiators and baseboard heating units have been widely used for many years, especially in Europe. Those systems are almost always attached to a wall, and the wall surfaces within the space are heated by these heating devices not only via natural convection but also radiation. A portion of the radiant energy transfer from the systems is delivered directly to people in the space, so this direct radiation incident on people affects thermal comfort in the zone. To correctly model baseboard heaters, an algorithm should account for the radiation impact on the surfaces as well as thermal comfort in the zone. Almost all baseboard heater models, however, including the existing models in EnergyPlus are convective-only models where all heat generated is assumed to be convected to the space, thus only affecting the air temperature in the zone.

To remedy this deficiency, new baseboard models have been developed and implemented in EnergyPlus so that these models appropriately handle both convective and radiant heat additions from a baseboard heater or radiator to the space. In these models, the radiant heat gain from a baseboard heater is distributed to the surfaces in the space and people by user-defined fractions, so that the surface heat balances as well as thermal comfort level are impacted. The heat from the surfaces and the convective heat gain from a heater are then delivered to the surrounding air via convection.

The most important contribution of this work is a significant improvement in the predictions of both the thermal comfort and the system response of these units. Convective-only baseboard models have no direct impact on the surface temperatures of the space. The new models presented in this paper do impact surface temperatures which then play a role in thermal comfort predictions, heat loss calculations, and system response. As a result of this new calculation method, the new models are able to calculate the total convective system impact by summing the additional convective heat gain from the surfaces as a result of receiving radiation from the baseboard unit and the convection to the surrounding air while typical convective-only models include only convection to the surrounding air. As a result of this new calculation algorithm, the prediction of thermal comfort, surface and air temperatures, system response, and heat loss through the building envelope is more accurate.

This paper will present a brief introduction to the deficiencies of existing models, a description of new model, and a demonstration of the new model capabilities. In addition, a comparison between the convective-only water baseboard model and the new radiant/convective water baseboard model will also be presented to demonstrate the advantages of the new model.

EXISTING BASEBOARD MODELS

Simulation algorithm of convective-only models

There are two different convective-only baseboard models in EnergyPlus. The current convective-only water and electric baseboard and radiator models assume that all heat from the heaters is delivered to the space via convection, while, in fact, a portion of heat is really radiated to surfaces and people. All baseboard models in EnergyPlus are intended to meet any remaining zone load that other heating systems have not met at each time step. As for the water baseboard model, the heating media is supplied from the primary system, and the water flow is controlled depending on the remaining zone load. Heat from the heaters is convected from the water to the surrounding air within the space, so that the zone air heat balance is impacted. The models do not include any transient heat storage in the unit itself and thus will only impact the space when it is operating.

The heat transfer between the water and the zone air is determined by an effectiveness-NTU heat exchanger model. Since only the inlet water and air temperatures are known, this methodology is well suited to determine the performance of the heater. The heater is assumed to be cross flow with both fluids unmixed. The model requires the user to input a UA (U-factor times Area) value and maximum design water flow rate. These parameters are also auto-sizable in EnergyPlus when the user is uncertain of their values. The water and air temperatures at the inlets are assumed to be the outlet temperature of the plant in the primary heating system and the zone temperature, respectively. The water mass flow rate is controlled depending on the remaining load in the space. The air mass flow rate is currently assumed to be twice the water mass flow rate due to the complexity of the prediction. An accurate prediction of the air mass flow rate is complicated because it is very sensitve to the other fluid’s conditions as well as the space conditions. While this assumption may lead to inaccuracy in the prediction of the actual conditions of the fluid, the model allows this uncertainty to be adjusted in the temperatrue prediction. An algorithm that predicts more accurate air mass flow rate will probably developed.

- Water baseboard model

Once the initial conditions are set, the model determines the effectiveness as follows:

$$C_w = c_{p,w}\dot{m}_w \quad (1)$$

$$C_a = c_{p,a}\dot{m}_a \quad (2)$$

These capacitances provide the maximum and minimum capacitances for the model. The capacitance ratio Rc, NTU, and effectiveness are then determined as:

$$R_c = \frac{C_{\min}}{C_{\max}} \quad (3)$$

$$NTU = \frac{UA}{C_{\min}} \quad (4)$$

$$\varepsilon = 1 - \exp\left\{\frac{NTU^{0.22}}{R_c}\left[\exp(-R_c NTU^{0.78}) - 1\right]\right\} \quad (5)$$

Once the effectiveness value is known, both the air and water outlet temperatures are obtained from the following expressions:

$$\varepsilon = \frac{C_a(T_{a,out} - T_{a,in})}{C_{\min}(T_{w,in} - T_{a,in})} \quad (6)$$

$$C_w(T_{w,in} - T_{w,out}) = C_{\min}(T_{w,in} - T_{a,in}) \quad (7)$$

The heating load that the heater meets at each time step is then obtained from the following expression:

$$q = C_w(T_{w,in} - T_{w,out}) \quad (8)$$

- Electric baseboard model

The heating capacity of the electric heating unit is the remaining design heating load to be met by this device in the space. The model, however, assumes the heating capacity to be the rated maximum capacity of the unit, i.e. nominal capacity, when the design load is greater than the maximum. The actual heating capacity is thus determined by the following expression, if the nominal capacity is greater than the design heating load:

$$q = \frac{q_{design}}{\eta} \quad (9)$$

If the design heating load is greater than the nomnal capacity, the actual heating capacity is:

$$q = \frac{q_{nom}}{\eta} \quad (10)$$

Limitations

Baseboard heaters or radiators are typically attached to walls in perimeter zones through which the majority of heat loss to the exterior environment takes place. They radiate an amount of heat to the other surfaces and people in the space, so that the surface temperatures and thus the surrounding air temperature and therrmal comfort of the people in the space are impacted. The convective-only baseboard and radiator model in EnergyPlus as well as almost all the other models, however, affect only zone mean air temperature. Those models thus overestimate the heating demands that the heaters should meet as well as energy use. The convective-only models do not fully account for impact of modified surface temperatures on the thermal comfort level in the space since heat radiated from the heater improves it. To handle appropriately the actual system impact, a model should include the impact of radiant heat addition as well as convective heat addition from the heater to the space.

The current convective-only water baseboard model in EnergyPlus requests users to input a UA value that is not rated or provided by manufacturers and this UA value significantly affects the heating capacity of the heaters. The model does autosize a design UA value. However, it predicts the design UA value from design inlet conditions, i.e. water temperature at the primary system outlet and zone air temperature. The temperature differences between the water and air vary with the surface area of the pipe through the heaters. An appropriate mean temperature is thus necessary to calculate the UA value and thus the heating capacity of the unit. The log mean temperature difference (LMTD) methodology is necessary to resolve the gap in this model.

FEATURES OF NEW MODELS

Model Enhancements

New baseboard models in EnergyPlus that handles both radiant and convective heat transfer are able to handle systems that are based on water, steam, and electrical input. They are intended to improve the limitations of the existing models described above and the accuracy in the calculation of the actual system impact. These new baseboard models include the convective impact to the surrounding air as well as radiant heat transfer to the surfaces and people. The actual system impact in a space is the sum of the additional convective heat transfer from the surfaces to the zone air after they have been heated by the heater as well as radiant heat transferred to people and the convective heat transfer to the space from the heater. This actual convective heat tries to meet any remaining heating requirement in the space in each system time step. The model thus improves the accuracy of heat loss calculation through the surfaces, thermal comfort predictions, and system responses because the new baseboard models allow these units to directly impact the surfaces within the space via radiation from the unit.

The water baseboard model determines an standard UA value based on standard rating information available in the literature. Almost all baseboards and radiators are rated under standard conditions such as water flow rates and average water temperatures. The heating capacity of these heating devices can thus be obtained from manufacturer's information or rating document such as the I=B=R rating document while current convective-only model asks the user to input an UA value or autosizes it from the design conditions. The new model calculates the UA value for the unit using the log mean temperature difference (LMTD) method so that the heating output from the heater can be determined more correctly than the convective-only water baseboard model. The new model only requests standard rating values that can be easily obtained in the literature and the fraction of radiant heat from the heater to the surfaces as well as people in the space. The usability and accuracy of the calculation in the UA value is thus greatly improved.

Simulation algorithm

- Water baseboard model

The model initializes all the conditions at the inlet node such as mass flow rates, temperatures, and specific heat capacities. It then calculates a UA value using the LMTD method. Since the heating capacity, the average water temperature, and the standard water mass flow rate are known from the user inputs, the standard inlet and outlet water temperatures are determined from the following expressions:

$$T_{w,in} = \frac{q_{std}}{2\dot{m}_w c_{p,w}} + T_{w,avg} \quad (11)$$

$$T_{w,out} = 2T_{w,avg} - T_{w,in} \quad (12)$$

The model then assumes the air mass flow rate is twice the rated water mass flow rate. Since the inlet air temperature of 18℃ and the rated heating capacity are known from the standard rating information, the outlet air temperature is thus:

$$T_{a,out} = \frac{q_{std}}{2\dot{m}_w c_{p,a}} + T_{a,in} \quad (13)$$

All the temperatures at each node are now known. The mean temperature difference is thus obtained from the following expressions:

$$\Delta T_1 = T_{w,in} - T_{a,out} \quad (14)$$

$$\Delta T_2 = T_{w,out} - T_{a,in} \quad (15)$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\log\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (16)$$

The UA value of a specific unit is thus:

$$UA = \frac{q_{std}}{\Delta T_{lm}} \quad (17)$$

Once the UA value is determined, the model employs an effectiveness-NTU heat exchanger method to determine the heat transfer between the heater and the air in the space during the simulation by using equations (1) through (8).

- Steam baseboard model

The basic methodology of the steam baseboard model is the same as the water baseboard model except that the steam model involves a phase change on the steam/water side. Like the water baseboard model, this model calculates the mass flow rate of steam based on the heating demand in a space. This model then determines the heating capacity from the sum of the latent heat transfer and sensible cooling of water using following expression:

$$q = \dot{m}_s (h_{fg} + c_{pw}\Delta t) \quad (18)$$

- Electric baseboard model

While the electric baseboard does not have fluid flowing through it and thus does not require a UA value, the rest of the features of the new radiant/convective algorithm as far as handling the radiation exchange between the unit and the surfaces and people within the space being conditioned are the same.

- Radiant heat calculation

The new models include an algorithm for calculating radiant heat addition from the baseboard heaters to the space. These models request user-defined fractions to model the portion of radiant heat transferred to the individual surfaces within the space. The user must also specify the fraction distributed to people. The radiant heat falling on people impacts the prediction of thermal comfort of the zone occupants. It, however, would get lost from the heat balance though this radiant heat is added to the space. To maintain overall energy balance, it is assumed to be convected to the space so that the heat balance can include this heat addition.

Once the heating capacity of the unit is determined, the radiant heat additions are:

$$q_{rad} = q \cdot F_{rad} \quad (19)$$

$$q_{conv} = q \cdot F_{conv} \quad (20)$$

$$q_{people} = q_{rad} \cdot F_{people} \quad (21)$$

$$q_{surf} = q_{rad} \cdot \Sigma F_{surf,i} \quad (22)$$

These radiant heat sources are then distributed to people and appropriate surfaces in the space so that the surface heat balances are altered. To calculate the impact of the radiant heat addition to the surfaces, the model determines the difference in convection between surfaces heated by radiant heating devices and unheated surfaces to the air. The actual system impact rate is thus expressed as:

$$q_{req} = (q_{surf} - q_{surf,z}) + q_{conv} + q_{people} \quad (23)$$

CASE STUDY DESCRIPTION

Case studies have been performed to investigate the simulation capability as well as the enhancements in the predictions of indoor thermal environment and system response. Main indoor environmental factors such as mean radiant temperature (MRT), zone mean air temperature (MAT) as well as energy consumption were predicted and compared between the convective-only water baseboard model and new water baseboard model in EnergyPlus. The simulations were run with the EnergyPlus version 4.0 October through March for the long-term simulations using the Chicago, IL TMY3 weather data. The simulation time step was set to 15 minutes in all simulation runs.

The medium office benchmark model in EnergyPlus was chosen for these case studies. This three story rectangular building has a total area of 4,932m^2, an aspect ratio of 1.5, and window to wall ratio of 0.33 on all the outside walls. Each floor is identical and divided into 4 perimeter zones named zone 1 through 4, and 1 core zone named zone 5, as shown in Figure 1. The total length and width of the building is approximately 50m and 33m, respectively. Each zone has a ventilation rate of 2.5 L/s per person, heat gain from lights of 10.76w/m^2, and an electric plug load of 8.07w/m^2. Approximately 11 people and infiltration of 0.0411m^3/s were applied to the perimeter zone 1 and zone 3. The number of people and infiltration rate in the perimeter zone 2 and 4 were approximately 7 and 0.028m^3/s, respectively. The number of people in the core zone 5 was set to 53. Variables for the thermal comfort models were a metabolic rate of 120W/person, a work efficiency of zero, a clothing level of 1.0clo, and an air velocity of 0.2m/s. The building is set to be occupied from 8AM to 5PM while the fraction of the occupancy varies with time. The building HVAC systems include a multi-zone VAV system with reheat coils in the core zones and with no reheat coils in the perimeter zones with a backup of water baseboard heaters. The HVAC systems are scheduled to be operated from 6AM to 10PM. The heating setpoint is 15.6℃ during unoccupied period and 21℃ during the occupied. The zone averaged scheme for the zone mean radiant calculation was chosen. The fractions radiant of 0.5, 0.4, and 0.3 were set to electric equipment, internal lights, and people in each zone, respectively.

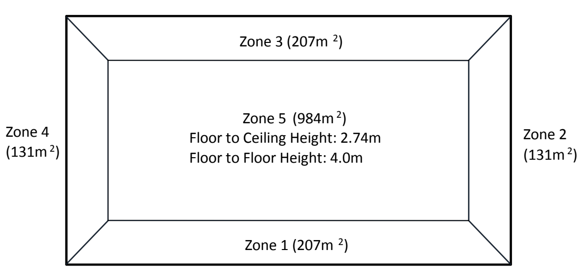

Figure 1 Space zoning and areas in the benchmark medium office building

A medium-capacity baseboard heater in the I=B=R rating document was chosen as for the inputs of the new model. Baseboard heaters were set as a secondary heating system in the perimeter zones which meets any remaining heating loads in the zones. All baseboard heaters were set to have the standard water mass flow rate of 0.063kg/s and the standard average temperature of 82.2℃. The maximum water flow rate of 0.005m^3/s was entered. The fraction of the total radiant heat from the heater was 0.3, and the fraction for the heat being incident on people within the total radiant heat was set to 0.3. The fractions of radiant energy to the surfaces were set to 0.4 for the surfaces where the heater is installed, and 0.1 for the other three vertical walls. The actual lengths of the heaters were set 40m for the zone 1 and 3 and 25m for the zone 2 and 4. The heating capacities were thus 9957.6 Watts for zone 1 and 3 and 4978.8 Watts for the zone 2 and 4, respectively. In addition, the UA values obtained by the new model were put in the input of the convective-only model.

DISCUSSION

Thermal environment

Indoor thermal environment in the meduim office benchmark building were estimated by the convective-only water baseboard model and new water baseboard model. As a representative case, the variation of the MRTs in the five zones on the first floor was shown in Figure 2. The fraction radiant of 0.3 was set to the new baseboard model for this analysis. The MRTs estimated by the two models were very close each other between 1AM and 6AM because the baseboard heating rates were fairly low thus the radiant heat source from the baseboard heaters in each zone has a little impact to the surfaces within the zones. In the four perimeter zones on the first floor, considerable differences in MRTs between the two models were observed from 6AM due to the significant increase of the baseboard heating requirements. Maximum differences in the MRTs between 6AM and 7AM appeared as approximately 0.73℃ in the zone 1 and 3 and 0.9℃ in the zone 2 and 4. Thosee differences decreased as the radiant heat sources from the heaters decreased. On the other hand, small differences in MRTs in the core zone during the occupied appeared. It is because that the interior inter zonal surfaces in zone 5 has little impact from the radiant heat sources about 10% of the total radiant heat to the surfaces. The maximum difference in MRT in this core zone was 0.21℃ at 9AM. It is likely that the interior surfaces were not only heated by the radiant heat sources but also affected by the solar radiantions incident on the south zone 1. Accordingly, as the remaining heating demands that the primary heating system did not meet increased, the radiant heat sources increased, so that the surface heat balances were impacted thus increased MRTs in the zones.

Maximum differences of approximately 0.9℃ in the zone 2 and 4, and of approximately 0.21℃ appeared in the MAT predictions by the two basebaord models. Small differences in MAT in all zones during the ouccupied appeared because baseboard units were trying to meet the heat setpoint temperature based on the remaining heating loads in the zones. On the other hand, the number of hours that meet the heating setpoint 21℃ during the occupied in the convective model predictions was greater than in the new model predictions as shown in Table 3, and this number increased along with the increase of fractions radiant. It is because that the convective baseboard model impacts only the zone air temperatures thus alters the MATs with no interaction with the surface heat balances. The MATs in the zone 1 and 5 the new model predicted were approximately 0.1℃ higher than ones the convective model predicted with no big difference. However, a different tendency appeared in the zone 2 and 4. The time that did not meet the heating setpoint in the zone 2 was between 10AM and 4PM while that in the zone 4 was between 1PM and 5PM. The heating setpoint was met throughout the entire hours in the north zone, zone 3, and the west zone, zone 4, in the morning. Accordingly, the heat balance calcualtions in the new model predictions included various radinat heat sources at each time step, so that the radiant heat additions in the zones altered the surface temperatuers as well as the zone temperatures.

Energy end use & influence of fraction radiant

The new baseboard models requires the user to specify the fracton radiant, which determines the radiant heat sources distributed to surfaces and people. In fact, accurate calculation or estimation of radiant heat from HVAC equipment, which impacts the overall heat balances is difficult. To investigate the effects of the fractional value, simulations October through March with different fractions ranging from 0.1 to 0.5 with intervals of 0.1 were carried out.

The annual heating loads in the five zones on the first floor decreased as the fractional values increased, as shown in Table 1. Typically the heating loads of a core zone in a cnotrolled space are small enough, and oftentimes cooling loads appear during occupied hours. In the zone 5, heat losses occur largely through the floor and the ceiling. They also appeared through inter-zonal heat transfer between the core zone and the other perimeter zones when the heating system starts to operate during setback hours. As a result, this zone has little radiant heat impact from the baseboared heaters. The heating load of the core zone in the fraction of 0.1 case decreased approximately 5.97% less than that the convective model predicted, and decreased up to 12.79% in the fraction of 0.5 case. When the fractions were set to 0.1 and 0.2, the reduction percent in the heating loads differed from each perimeter zone. In the fraction of 0.1 case, percents varied 8.19% in the zone 1, 1.24% in the zone 2, 5.9% in the zone 3, and 1.79% in the zone 4. These differences of the percents among zones decreased as the fractions increased. The percents increased according to the fractions up to 27.61% in the zone 4. Accordingly, the fractions significantly affects the heat loss calculations, and the heating loads on the first floor of the medium office benchmark building decreased up to 27.61% in the fraction of 0.5 case.

Table 2 showed baseboard heating energy for the entire simulation period in the four perimeter zones on the first floor, which was predicted by the two different water baseboard models in EnergyPlus. The convective basebaord model predicted 14008.17MJ, 13367.41MJ, 20596.13MJ, and 14126.07MJ in the zone 1, 2, 3, and 4, respectively. The predictions by the other algorithm at the fraction radiant of 0.1 represented 13888.66MJ, 13666.62MJ, 20594.21MJ, and 14487.01MJ in the zone 1, 2, 3, and 4, respectively. The differences in the baseboard heating energy predictions between the two models were less than 2.5% in this case. However, they increased up to 14.73% in the zone 4 at the fraction radiant of 0.5. The differences in the percent reductions among zones also varied with the fractions. It was turned out that while the new modeling algorithm affects the heat loss calculations thus decreased the primary HVAC system loads, baseboard heating energy increased as the radiant heat additions to the space increased. It is because that the actual convective system impacts determined by the new baseboard models were almost always smaller than the baseboard heating energy, which then included to the overall heat balances in the zone. As a result, the overall HVAC system heating energy increased as the fractions increased as shown in Table 3. The radiant heat additions from the basebaords has small effects to the overall HVAC system energy consumptions while the convective model underestimated the basebaord heating capacities.

CONCLUSION

Multiple baseboard heater and radiator models that predicts the actual convective system impact of the units by impacting surface heat balances were developed and added to EnergyPlus. An analysis of comparisons between the convective model affecting only zone air temperatures and the new model developed was conducted to demonstrate overall impact of the radiant heat additions predicted by the new model. The new baseboard models significantly improves the heat loss calculations, delivering the radiant heat to the surfaces thus leading to reductions in heating load in a space and overall HVAC system loads. In addition, the purely convective baseboard model underesimates overall baseboard heating capacity due to the direct imact to the air heat balances with no alteration of the surface heat balances. The value of fraction that the user must input significantly affects overall system performance, and further study is necessary to figure out relationships between the fractions and the performance of the baseboard heaters. While the new baseboard models that include an existing algorithm for modeling radiant heat transfer in EnergyPlus require additional computation time, it is able to accounts for the actual effects of these heating devices.

NOMENCLATURE

C_a : capacitance of air
C_w : capacitance of water
$c_{p,a}$: specific heat capacity of air (J/kg· K)
$c_{p,w}$: specific heat capacity of water (J/kg· K)
h_{fg} : heat of vaporization of steam (J/kg)
$\dot{m}_a$: mass flow rate of air (kg/s)
$\dot{m}_s$: mass flow rate of steam (kg/s)
$\dot{m}_w$: mass flow rate of water (kg/s)
F_{people} : fraction radiant to people
F_{rad} : fraction radiant
$F_{surf,i}$: fraction radiant to ith surface
R_c : capacitance ratio
q : heating capacity of a heater (W)
q_{conv} : convective heat from a heater (W)
q_{design} : design heating load (W)
q_{nom} : nominal heating capacity of a heater (W)
q_{rad} : radinat heat source from a heater (W)
q_{people} : radinat heat to people from a heater (W)
q_{req} : actual convective heating load (W)
q_{std} : rated heating capacity of a heater (W)
q_{surf} : sum of convection from surfaces (W)
$q_{surf,Z}$: convection from surfaces with no radiation (W)
$T_{a,in}$: inlet temperature of air (℃)
$T_{a,out}$: outlet temperature of air (℃)
$T_{w,in}$: inlet temperature of water (℃)
$T_{w,out}$: outlet temperature of water (℃)
$T_{w, avg}$: rated average temperature of water (℃)
η : efficiency of a heater
ε : effectiveness of a heater
ΔT_{lm} : log mean temperature difference
Δt : degree of sub cooling (℃)

ACKNOWLEDGMENT

This material is based upon work supported by the Department of Energy National Energy Technology Laboratory under award number DE-FC26-06NT42768.

DISCLAIMER

REFERENCES

Incropera and DeWitt, Fundamentals of Heat and Mass Transfer, Chapter 11.3 and 11.4, eq. 11.15, 11.17, and 11.33.

Li Lianzhong and M. Zaheeruddin (2006), Dynamic modeling and simulation of a room with hot water baseboard heater, Int. J. Energy Res.; 30, p427–445

I=B=R Ratings for Boilers, Baseboard Radiation, Finned Tube (Commercial) Radiation, and Indirect Fired Water Heaters, January 2009 Edition

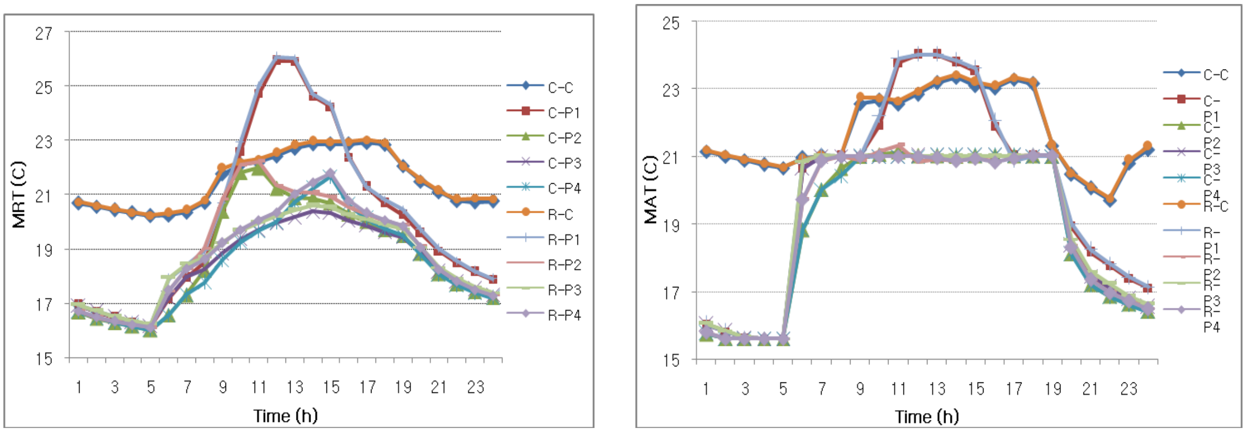

Figure 2 Variation of zone mean radiant temperatures and zone mean air temperatures on a winter day

Table 1 Zonal heating loads(MJ) on the first floor on a winter day

Zone	Convective	F - 0.1	F - 0.2	F - 0.3	F- 0.4	F - 0.5
1	9441.93	8668.33	8614.39	8138.78	7639.04	7129.39
2	7344.34	7252.98	7034.35	6536.58	6042.92	5508.96
3	13566.9	12766.59	12220.3	11452.4	10665.2	9822.73
4	7986.56	7843.66	7464.53	6902.23	6351.6	5781.29
5	3238.17	3044.81	2996.15	2944.85	2888.37	2824.17

Table 2 Baseboard heating energy (MJ) during the entire heating season

Zone	Convective	F - 0.1	F - 0.2	F - 0.3	F- 0.4	F - 0.5
1	14008.17	13888.66	14222.40	14574.43	14971.05	15409.79
2	13367.41	13666.62	14030.69	14412.26	14810.17	15252.77
3	20596.13	20594.21	21222.78	21913.18	22659.4	23452.79
4	14126.07	14487.01	14884.83	15292.85	15736.1	16206.35

Table 3 Annual sums of HVAC system performance during heating period

Overall HVAC Air System Loads (MJ)						
	Convective	F - 0.1	F - 0.2	F - 0.3	F- 0.4	F - 0.5
VAV1	68042.6	67264.6	67022.3	66755.4	66461.4	66137.8
VAV2	89487.8	87294.2	86502.6	85581.1	84584.8	83504.6
VAV3	218797	213624	211871	210011	208036	205924
Overall HVAC System Energy for Heating (MJ)						
-	635778	632750	634782	639335	644214	649416
Time set point not met during occupied heating hours (hrs)						
-	321.5	358.75	378.25	389.5	415.5	430.5