Internship Report

Master of Science in System, Control and Information Technology

Modeling, Identification and Control of Experimental Platform For Energy Management in Intelligent Buildings

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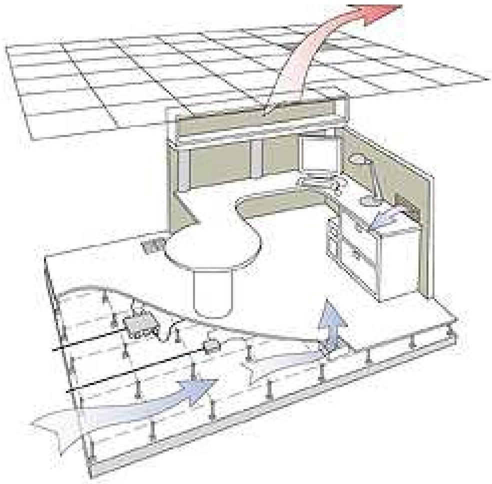
1 Introduction



Underfloor air distribution (UFAD) is a method of delivering space conditioning in offices and other commercial buildings that is increasingly being considered as a serious alternative to conventional ceiling-based air distribution systems because of the significant benefits that it can provide. This technology uses the open space (underfloor plenum) between the structural concrete slab and the underside of a raised access floor system to deliver conditioned air directly into the occupied zone of the building. Air can be delivered through a variety of supply outlets located at floor level (most common), or as part of the furniture and partitions. UFAD systems have several potential advantages over traditional overhead systems, including improved thermal comfort, improved indoor air quality, and reduced energy use. By combining a building's heating, ventilating, and air-conditioning (HVAC) system with all major power, voice, and data cabling into one easily accessible service plenum under the raised floor, significant improvements can be realized in terms of increased flexibility and reduced costs associated with reconfiguring building services. These raised floor systems are particularly appropriate for office buildings housing today's businesses with their typically extensive use of information technologies and high churn rates. [1]

Energy consumption in buildings (both commercial and residential) has reached up to 40% of the total energy use in developed countries and is rapidly increasing with the growing demand in comfort. This fact motivates many countries to make energy savings in buildings a priority, as mentioned by the European Energy Performance of Buildings Directives. As a result, a substantial amount of work has been done in the past decades toward intelligent buildings. This has been particularly the case for climate regulation in buildings, with research on modeling, simulation and control of Heating, Ventilating and Air Conditioning (HVAC) systems, to improve comfort and energy efficiency. On these matters, the UnderFloor Air Distribution (UFAD) solution has shown some interesting results compared to more traditional ceiling ventilation. [2]

In University of Joseph Fourier, there is a small-scale building represents a real building. The goal of this project is to present an appropriate model, which follow the behavior of small-scale experimental building in all the possible ways.

2 Experimental Building

Experimental building that is shown in figure1, Divided by three area; Ceiling Plenum, Rooms and Underfloor Plenum. There are 4 Rooms with different area and volume. In each room, there are two doors that designed as an interface to the adjacent rooms and also one fan mounted on the floor to flow the air from underfloor plenum to each rooms and one manual open/close exhaust in the interface layer with ceiling plenum to let the air out. One air channel completes the air circulation with connecting the ceiling plenum to underfloor plenum, with one fan, which is in the inlet of the channel to force the circulation. Material of all the walls, ceiling, and floor (all the enclosures) are PVC. For making this experimental model close to real building, we have some lamps in the rooms, which are representative of external heat source such as electrical device and human body in reality.

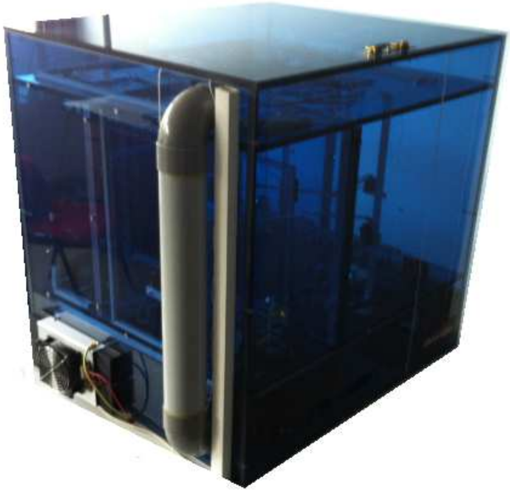


Figure 1- small-scale building

2.1 Programmable Controller and User interface software

The experimental model equipped by National Instruments CompactRIO programmable automation controller, which is able to read all the data coming from the building and also send appropriate commands with some dedicated I/O modules.

In order to have control on the experimental building such as sending commands to the actuators, monitoring the sensor's values, implementation of the controller, data gathering in order to analyze them, we will need a user interface area. Here, with using of **LabVIEW™** software we created such this interface.

2.2 Sensors and Actuators

In order to study on the thermal behavior of the experimental model we should have several sensors, which can measure the values of different areas. Here we should have six temperature sensor which represent the temperature of each rooms and plenums. Meanwhile, we should have possibility to control the fans, doors and heat sources as final element and actuators in each room. There are some servomotors dedicated for Opening and Closing the doors. The heat sources work with voltage supply of 12 VDC. The fans nominal supply voltage is 12 VDC, but here, we run this fan in several voltages to achieve different flow rate.

2.2.1 Calibration

At the preliminary tests that I have done to be more familiar with the model, and to know how it works, I encountered to some problems. The main problem that I have seen in the first look was the accuracy of data received by temperature sensors. Lots of oscillation and noise appeared in the received data from the temperature sensors.

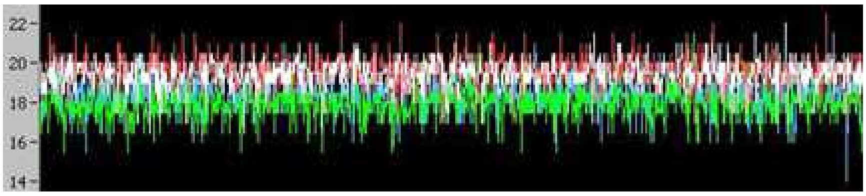


Figure 2- noise in temperatur sensors

As we can see in the figure above, I put all sensors value in a same plot but with different colors. The oscillation of each sensor is around 4 degree of centigrade, which is not acceptable. The worst case is when we turn ON the heat sources.

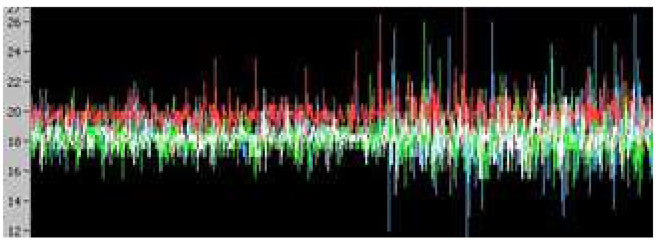


Figure 3- Increasing the noise with turning ON one of the heat sources

When all the Lamps are on, the noise level increase dramatically.

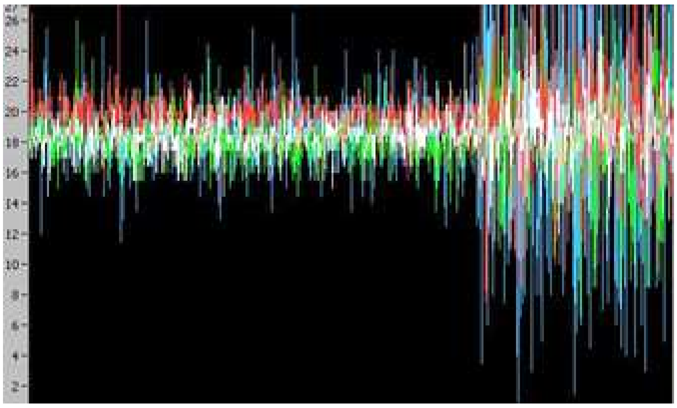


Figure 4- Increasing the noise with turning ON all the heat sources

We should discover source of noise to improve the quality of the read analog values. With presence of such a big noise, it was not possible to keep on testing procedures.

In the experimental building, for reading the analog signals such as temperature, humidity and pressure, we use $NI9201$ analog input module. This module dedicated for reading the analog signal in range of $-10V$ to $+10V$ with the good accuracy given by the table below.

Measurement Conditions	Percent of Reading (Gain Error)	Percent of Range* (Offset Error)
Calibrated typ (25 °C, ±5 °C)	±0.04%	±0.07%
Calibrated max (–40 to 70 °C)	±0.25%	±0.25%
Uncalibrated typ (25 °C, ±5 °C)	±0.26%	±0.46%
Uncalibrated max (–40 to 70 °C)	±0.67%	±1.25%
* Range equals 10.53 V		

Table 1- NI 9201 accuracy (excludes noise)

Analog modules operate well, But the problem occur when we use just a little part of our data band and using the improper sensors.

For further wireless sensors network, an electronic board Previously had been designed that receives thermocouple signal and delivers output in mV to another board, that convert mV signal to analog signal between $0V$ to $1.2\ V$. We know that with adding elements between sensors (final elements) and processor, we will decrease the SIL (safety integrity level) of system. With such this strategy, temperature measurement should be scaled between $0V$ to $1.2\ V$. It will decrease the accuracy 20 time less than the original one, because resolutions of analog input modules are constant and limited and if we want to measure a wide range of temperature in just a small range between $0V$ to $1.2\ V$ actually we will lose lots of accuracy

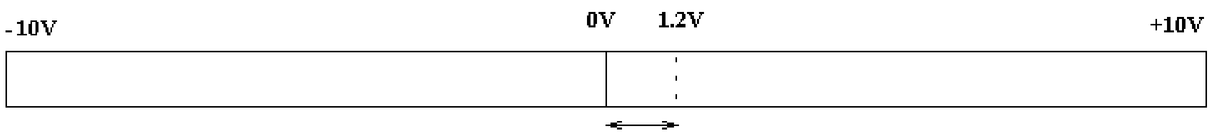
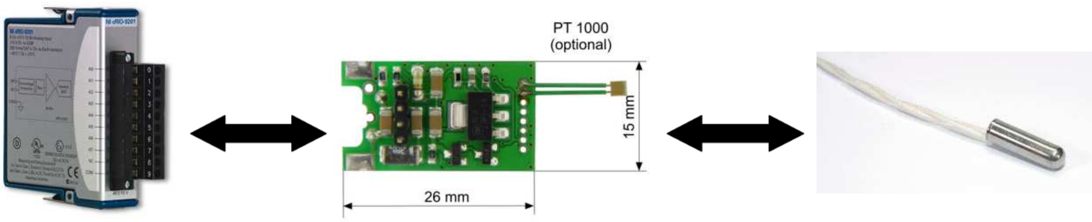


Figure 5- range of analog module in comparison of the range we already used

Therefore, the solution would be to change the method of signal translation.

I believed that we should not loss our NI modules accuracy with reducing the data band. We should use all $-10V$ to $+10V$ or at least $0V$ to $10V$ range, then we can easily use standard sensors ($-10V$ to $+10V$ or $0V$ to $10V$ are standard range that all the manufacturer use for their products), and then use special modules for translate it to wireless data.

Using RTD sensors such as PT-100 or PT-1000 instead of Thermocouple was the other solution to improve temperature signals quality. Thermocouple is not suitable for low temperatures measurement[1]. I change all the rooms' thermocouple sensor with RTD type sensors (PT-1000) and change all the electronic interface boards with standard transducer to read the PT-1000 data and change it to $0V$ to $10V$ as output. In this case, all the thermal sensors are the same, all the transducers are the same and this similarity, will made a better references to evaluate data.



Analog Input Module | Transducer PT1000 to 0-10V | PT1000

Divide the wires roots of analog signals from digital signals was the other solution to decrease the noise effect on the analog sensors. Because of the inductive effects, we are not allowed to cross the wires that carry two different type of signal.

After doing all these modifications, we eliminate all the noise effect on the analog data. Now, we can trust all the data that we receive from the sensors.

[1] Wikipedia: The main limitation with thermocouples is accuracy; system errors of less than one degree Celsius (°C) can be difficult to achieve.

3 Theoretical Model

The theoretical model that we are going to describe is base on the Law of Conservation of Energy and Mass. At first, we make some assumption as follow:

We use a 0-dimensional[2] model to describe the temperature dynamics in a building equipped with Under Floor Air Distribution. The air is moving at a reduced speed, therefore we will be considering the air to be incompressible: $\rho_i = \rho_{air}$. The kinetic and potential energies will be neglected due to the reduced speed and low mass of the gas.

3.1 Initial model

The model of the temperature behavior in the room i will be derived from the first law of Thermodynamics:[3]

$$\frac{dE_i}{dt} = \dot{Q}_i + \sum \dot{m}_{in_i} h_{tot,in_i} - \sum \dot{m}_{out_i} h_{tot,out_i}$$

Where $E_i = \rho C_v V_i T_i$ is the room energy, $\dot{Q}_i$ the heat exchanges, $\dot{m}_{in_i}$ and $\dot{m}_{out_i}$ are the input and output mass flow rates for the room i and $h_{tot} = C_p T$ the total enthalpy, C_p and C_v are the constant pressure specific heat and constant volume specific heat respectively, and V_i is the volume of the room.

We will consider three types of energy associated with heat exchanges:

- The conduction (Fourier's law) from room j to room i through a wall: $\dot{Q}_{cond} = -kA(T_i - T_j)/\Delta x$, where k is the conductivity of the wall (in $[W/m.K]$), A its surface of conduction and Δx its thickness.
- The radiation from heat source s in the room i : $\dot{Q}_{rad} = \epsilon\sigma A_s(T_s^4 - T_i^4)$, where ϵ is the emissivity, $\sigma = 5.67 * 10^{-8}\ Wm^{-2}K^{-4}$ is the Stephan-Boltzmann constant, and A_s is the surface of the source;
- the energy associated to the mass flow rates $C_p \dot{m}\, T$ (where T is the temperature of the room from which the flow is coming), which can be divided into three groups:
  - $C_p \dot{m}_{pl_i} T_{pl}$: the mass flow forced by the underfloor fan, which is an input of the system ;
  - $C_p \dot{m}_{d_{ij}} \max\left(T_i, T_j\right) = C_p * \rho A_d \sqrt{2R\left|T_i - T_j\right|} * \max\left(T_i, T_j\right)$: associated with the mass flow rate going from a high temperature room to a low temperature room when the door between them is open. ρ is the air density, A_d the surface of the door and $R = C_p - C_v$.
  - the last term will correspond to the mass flow rate going from the room i to the ceiling, resulting from the mass conservation: $C_p \dot{m}_{c_i} T_i = C_p * \left(-\dot{m}_{pl_i} - \sum_j \dot{m}_{d_{ij}}\right) * T_i$

[2] In 0-dimansion model, we make an assumption that all the area in an enclosure have unique temperature.

For a given room i, we will consider the following convention: all mass flow with index i is positive if entering the room. Therefore, the mass conservation equation for the room i will be written:

$$\dot{m}_{pl_i} + \dot{m}_{c_i} + \sum_j \dot{m}_{d_{ij}} = 0$$

With $\dot{m}_{pl_i} > 0$, $\dot{m}_{c_i} < 0$ and $\dot{m}_{d_{ij}}$ positive if $T_i < T_j$ and negative otherwise.

We have two Boolean variables to introduce in the model. Let's note $\delta_{d_{ij}}$ the Boolean corresponding to the opening of the door between rooms i and j, and $\delta_{s_i}$the one for the state of the source s in room i.

The expression of the heat exchanges, applied to the first law of thermodynamics will give us, for room$i$:

$$\dot{E}_i = \rho C_v V_i \frac{dT_i}{dt} = -\alpha_{pl_i}(T_i - T_{pl}) - \alpha_{c_i}(T_i - T_c)$$

$$- \sum_{j \in \{iw_i\}} \alpha_{i,j}(T_i - T_j) - \sum_{k \in \{ow_i\}} \alpha_{i,k}(T_i - T_{out})$$

$$+ C_p \dot{m}_{pl_i} T_{pl} + C_p \dot{m}_{c_i} T_c + \sum_j \delta_{d_{ij}} C_p \dot{m}_{d_{ij}} \, max\,(T_i, T_j)$$

$$+ \sum_{s_i} \delta_{s_i} \epsilon \sigma A_{s_i} (T_{s_i}^4 - T_i^4)$$

Where $\{iw_i\}$ are the inside walls surrounding the room i, $\{ow_i\}$ the outside walls and $\alpha_x = k_x A_x / \Delta x_x$. On the right side of the equation above, the terms on the first and second lines correspond to the conduction (plenum, ceiling, inside and outside walls), the third line involves the mass flow rates, and the last term is for the heat sources.

$$\dot{m}_{pl_i} + \dot{m}_{c_i} + \sum_j \delta_{d_{ij}} \dot{m}_{d_{ij}} = 0$$

If we use the new hybrid mass conservation equation above and the expression of the mass flow rate going through a door in the expression of the heat exchanges, we obtain the final hybrid equation:

$$\dot{E}_i = \rho C_v V_i \frac{dT_i}{dt} = -\alpha_{pl_i}(T_i - T_{pl}) - \alpha_{c_i}(T_i - T_c)$$

$$- \sum_{j \in \{iw_i\}} \alpha_{i,j}(T_i - T_j) - \sum_{k \in \{ow_i\}} \alpha_{i,k}(T_i - T_{out})$$

$$+ C_p \dot{m}_{pl_i}(T_{pl} - T_i) + \sum_j \delta_{d_{ij}} C_p \rho A_{d_{ij}} \sqrt{2R} \delta_{i \le j} (T_j - T_i)^{3/2}$$

$$+ \sum_{s_i} \delta_{s_i} \epsilon \sigma A_{s_i} (T_{s_i}^4 - T_i^4)$$

Where we introduced the Boolean $\delta_{i\leq j}$ to simplify the notations: $\delta_{i\leq j} = 1$ when $T_i \leq T_j$, and 0 otherwise. Thus, the term involving the mass flow rate going through the door will only appear if the door is open and the temperature of the room i considered is smaller than the temperature of the room j connected. In that case, it will become: $C_p \rho A_{d_{ij}} \sqrt{2R}(T_j - T_i)^{3/2}$

4 Comparison between theory and reality

For the beginning, I simplified the model just considering the heat exchange through the PVC walls.

$$\dot{E}_i = \rho C_v V_i \frac{dT_i}{dt} = -\alpha_{pl_i}(T_i - T_{pl}) - \alpha_{c_i}(T_i - T_c)$$

$$- \sum_{j\in\{iw_i\}} \alpha_{i,j}(T_i - T_j) - \sum_{k\in\{ow_i\}} \alpha_{i,k}(T_i - T_{out})$$

In this case, all the energy stored in the room i will be exchanged through the walls. For practically test this part, we close all the doors, then heat up or cool down one of the rooms, after stopping the heat source or cold air source, we start data gathering and wait until room temperature return to the equilibrium value.

At the first look, with putting the real values of ρ, C_v , V_i and α ,we see that the rate of increase of decrease of the temperature in reality is too much slower than what we see in the model. In the following figure, we will find that, the time constant of initial model is pretty much lower than what we expect.

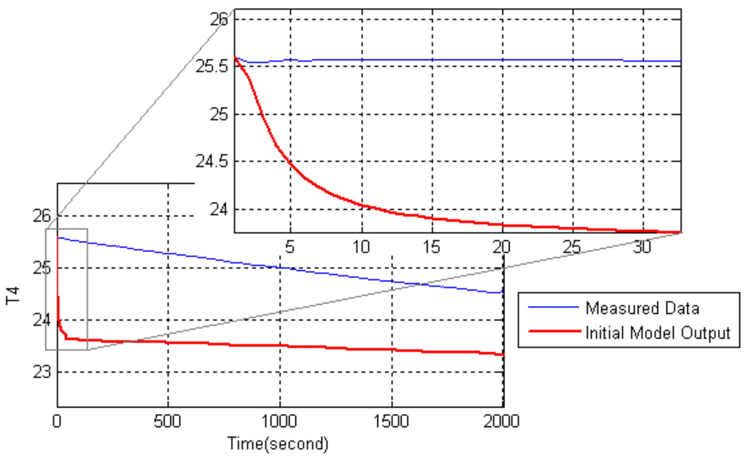


Figure 6- compression between time constant of initial model and reality

In figure6, temperature of room 4 at the starting of the test is 25.6 degree of centigrade, the initial model says that it will reach to 23.3 degree of centigrade just in 30 seconds, but the real data that we measured is totally different and decrease gradually.

If we want to keep our initial model, the only parameter that can be expected to change is Conductivity factor of the walls, because the other parameters such as geometrical values and specific heat constant and density are fixed values. To have the time constant same as what we saw in figure6, the value of k should be around 2×10^5 times bigger than what we expect. (Conductivity factor of PVC=$0.19[W/m.K]$). So, we should find another way to satisfy this high time constant.

5 Modified Theoretical Model

Based on our initial model, rate of losing or gaining the energy in the rooms, just in the case that all the heat source and fans are OFF and the doors is also closed, is depend on the k; conduction of the walls. In our modified theoretical model, we will consider two ideas as follow:

As we know, heat can be transferred from one place to another by three ways, conduction, convection and radiation. To justify the different time constant between our model and reality, we will express our model, with adding Convection term in the heat transfer equation.

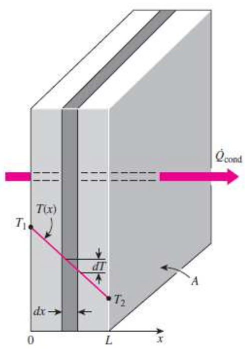
In addition, as I said before in our rooms' equation, all the energy stored in the room is $E_i = \rho C_v V_i T_i$, and it just depends on the volume of the room. It is believed that, not only the air stored in the room, but also all the elements inside of the room, even a some percentage of the walls thickness, absorb energy and should take into consideration in dynamics of the rooms.

With these two hypotheses, we rewrite rooms' equations.

- With the new definition of the room energy, we can say that:

Energy of the walls + Energy of the Air + Energy of other components inside of the room = Total Energy of the room

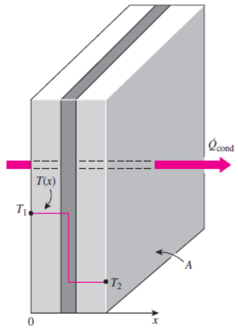
We can neglect energy of all components inside of the room and take it into the wall energy, because in our experimental flat, we just have one or two lamp and a servomotor with an aluminum crank to open/close the door, Estimation of how much they absorb energy is not simple.



$\rho_{air} V_{air} C_v T_{air} + \rho_{wall} V_{wall} C_p T_{wall}$ = Total E belong to the room

Total Energy of the room will go out through the wall.

$$Q_{cond,wall} = -kA\frac{dT}{dx}$$



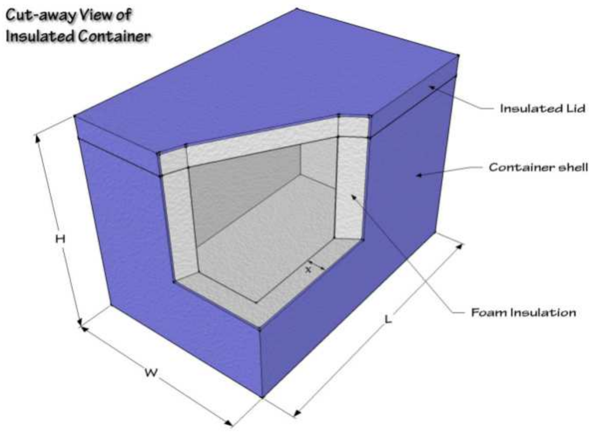
Where the rate of conduction heat transfer $Q_{cond,wall}$ and the wall area A are constant. Thus $\frac{dT}{dx}$ = constant, which means that the temperature through the wall varies linearly with x. That is, the temperature distribution in the wall under steady conditions is a straight line.

For elimination of a new state as wall temperature, with a good approximation we can divide the wall in two equal part such that each part has the temperature same as the nearest room temperature.

$$\rho_{air}V_{air}C_vT_{air} + \rho_{wall}\frac{V_{wall}}{2}C_pT_{air} = (\rho_{air}V_{air}C_v + \rho_{wall}\frac{V_{wall}}{2}C_p)T_{air} = \text{Total E}$$

- Heat Gain or loss through the walls of an insulated container can be defined in such a way that both conduction and convection factors are involved.[4]

$$Q = A\left[\frac{T_o - T_i}{\frac{x}{k} + \frac{1}{H_i} + \frac{1}{H_o}}\right]$$



$\boldsymbol{Q}$ is the heat lost or gained in watts
$\boldsymbol{x}$ is the thickness of insulation in meters (m)
$\boldsymbol{k}$ is the Thermal Conductivity of the insulation material in watts/meter K (or C)
$\boldsymbol{A}$ is the outside surface area of the container in meters squared (m^2)
$\boldsymbol{H_i}$ is the Heat Transfer Coefficient of the surface material inside of container in watts/meter^2 C[3]
$\boldsymbol{H_o}$ is the Heat Transfer Coefficient of the surface material outside of container in watts/meter^2 C
$\boldsymbol{T_o}$ is the Outside ambient air temperature in C
$\boldsymbol{T_i}$ is the Inside temperature in C

[3] Natural or Free Convection is essentially still to slightly stirred air with **H** values ranging from 1 to25. Forced Convection is air moved by a fan or other active method. **H** Values range from 25 to100.

The final equation with applying two hypotheses for a single container will be:

$$\left(\rho_{air}V_{air}C_v + \rho_{wall}\frac{V_{wall}}{2}C_p\right)\frac{dT_{in}}{dt} = -\frac{A_{wall}}{\frac{x}{k}+\frac{1}{H_i}+\frac{1}{H_o}}(T_{in} - T_{out})$$

$$\frac{dT_{in}}{dt} = -\frac{A_{wall}}{\left(\frac{x}{k}+\frac{1}{H_i}+\frac{1}{H_o}\right)\left(\rho_{air}V_{air}C_v+ \rho_{wall}\frac{V_{wall}}{2}C_p\right)}(T_{in} - T_{out})$$

6 Experimental building modification

In the other test that was focused on fans operation analysis, we confront with another matter that actually had a bad effect on the final model that we will construct. In the test, as we can see in the figure7, we turned ON the rooms' fans respectively and studied on the changing of each rooms' temperature.

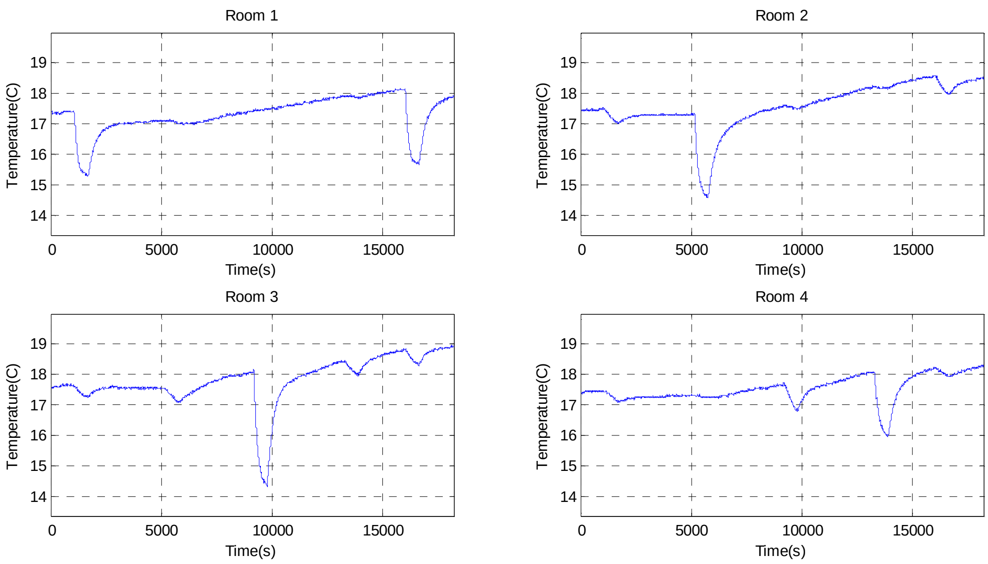


Figure 7-effect of the fans on the other rooms

As it is obvious in the plots above, with turning ON the fan, the corresponding room temperature decreases. With decreasing the room temperature, temperature of the adjacent rooms will decrease. But, the problem is decreasing the non-adjacent room. For instance when the fans of room 1 is running and temperature of room 1 is decreasing, according to our initial model, we should not see any decreasing effect in the temperature of room 3 same as the effect in rooms 2 and 4.

With closed look at the flow direction in the building, we will find out that when one of the rooms fan are running, it forces the air from underfloor to the room and in the other hand the air will be exhausted from the open hole mounted in the room ceiling. Therefore, the ceiling temperature will decrease. If we leave all the exhausts hole of the rooms open, naturally the cold air in the ceiling would come down to the hot rooms through the exhaust because of natural or free convection[4]. That is why we have seen small drop of temperature in non-adjacent room. To avoid from being more complex and keep the initial model structure we do not model the airflow comes from ceiling through the exhaust hole. The solution will be a proportional control on the exhaust opening, such that when the fan is running the corresponding exhaust should be open and when the fan is OFF the exhaust should leave closed.

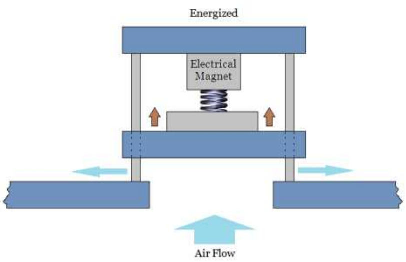


Figure 8- solution for preventing unwanted flow comes through the ceiling

The other issue that I took into account was interstice and gap between different parts of our experimental building such as ceiling and walls and also the gap between door and wall when door supposed to be quite close. Those cause some unwanted airflow from one room to the other, so I tried to do insulation for having accurate results in measuring. Doors, gaps between ceiling and walls, conjunctions of walls and gaps between walls and floor, have been insulated by special material that often used for building insulation. After controlling the exhausts caps and isolation the gaps, I have re-done the test with fans and the result was more justifiable.

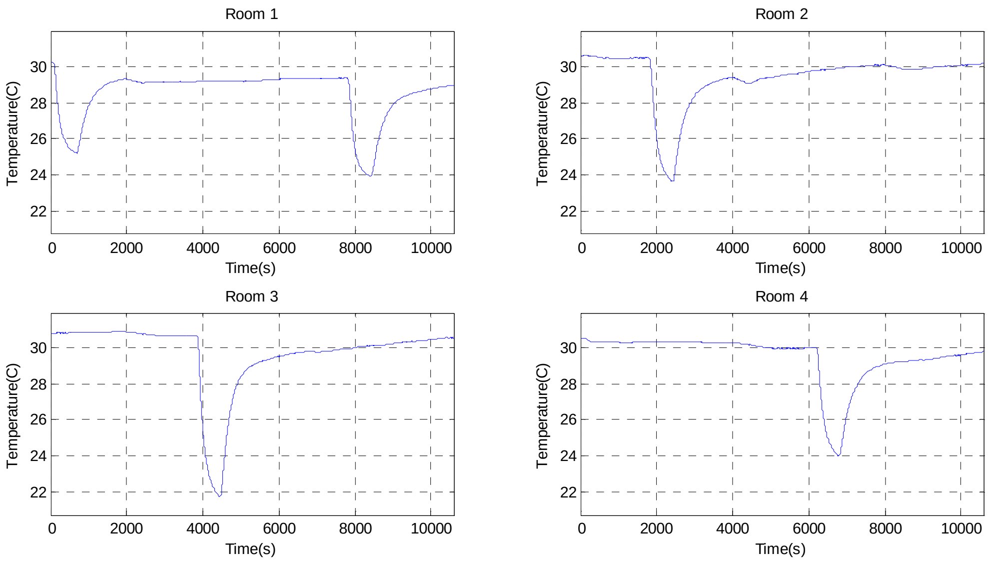


Figure 9- fan effect after modification

[4] Natural convection is caused by buoyancy forces due to density differences caused by temperature variations in the fluid. At heating the density, change in the boundary layer will cause the fluid to rise and be replaced by cooler fluid that also will heat and rise. This continues phenomena is called free or natural convection

As we can see in the figure9, all the extra effects have been eliminated. There is no extra flow from the ceiling to effect on the rooms' temperature.

7 Optimization of unknowns

After doing all the modification, for being sure about the data that we will receive from the building, it is the time to show that our new theoretical model can follow the experimental building behavior in all various conditions. In this step, with using of MATLAB® software, we will try to find the optimal parameters such that having the best fit to the reality. Here, we will change continuous time format to discrete time formats as follow:

$$\int_0^{N\Delta t} (T_i - T_c)dt \approx \frac{1}{\Delta t}\sum_{h=0}^{N-1} T_i(h\Delta t) - T_c(h\Delta t)$$

In all the tests that we have done, sampling time was one second. So $\Delta t = 1$.

Room's equation in continuous time form is:

$$R_i \frac{dT_i}{dt} = -\alpha_{ipl}(T_i - T_{pl}) - \alpha_{ic}(T_i - T_c) - \alpha_{iow}(T_i - T_{ow}) - \sum_{j\in\{iw_i\}} \alpha_{ij}(T_i - T_j)$$
$$+ C_p\dot{m}_{pl_i}(T_{pl} - T_i) + \sum_j \delta_{d_{ij}} C_p \rho A_{d_{ij}} \sqrt{2R}\delta_{i\le j}(T_j - T_i)^{3/2}$$
$$+ \sum_{s_i} \delta_{s_i}\epsilon\sigma A_{s_i}(T_{s_i}^4 - T_i^4)$$

$$\alpha_{ab} = \left[\frac{A_{ab}}{\frac{x}{k} + \frac{1}{H_a} + \frac{1}{H_b}}\right] \quad , \qquad R_i = (\rho_{air} V_i C_v + \rho_{wall}\frac{V_{wall_i}}{2} C_p)$$

After taking integral in both side of above equation we will have: (For instance for room 1)

$$R_1(T_1(t) - T_1(0)) =$$
$$-\left[\alpha_{1pl}\int_0^t (T_1 - T_{pl})\,dt\right] - \left[\alpha_{1c}\int_0^t (T_1 - T_c)dt\right] - \left[\alpha_{1ow}\int_0^t (T_1 - T_{ow})dt\right] - \left[\alpha_{12}\int_0^t (T_1 - T_2)dt\right]$$
$$-\left[\alpha_{14}\int_0^t (T_1 - T_4)\,dt\right] + \left[C_p\dot{m}_{pl_i}\int_0^t (T_{pl} - T_1)\,dt\right] + \left[\delta_{d_{12}} C_p\rho A_{d_{12}}\sqrt{2R}\delta_{1\le 2}\int_0^t (T_2 - T_1)^{3/2}dt\right]$$
$$+\left[\delta_{d_{14}} C_p\rho A_{d_{14}}\sqrt{2R}\delta_{1\le 4}\int_0^t (T_4 - T_1)^{3/2}dt\right] + \left[\delta_{s_1}\epsilon\sigma A_{s_1}\int_0^t (T_{s_1}^4 - T_1^4)dt\right]$$

With changing to discreet format, we will have:

$$R_1\big(T_1(t) - T_1(0)\big) \approx$$

$$-\left[\alpha_{1pl}\sum_{h=0}^{t-1}\big(T_1 - T_{pl}\big)\right] - \left[\alpha_{1c}\sum_{h=0}^{t-1}(T_1 - T_c)\right] - \left[\alpha_{1ow}\sum_{h=0}^{t-1}(T_1 - T_{ow})\right] - \left[\alpha_{12}\sum_{h=0}^{t-1}(T_1 - T_2)\right]$$
$$-\left[\alpha_{14}\sum_{h=0}^{t-1}(T_1 - T_4)\right] + C_p\dot{m}_{pl_i}\sum_{h=0}^{t-1}\big(T_{pl} - T_1\big) + \delta_{d_{12}}C_p\rho A_{d_{12}}\sqrt{2R}\delta_{1\le 2}\sum_{h=0}^{t-1}(T_2 - T_1)^{3/2}$$
$$+\delta_{d_{14}}C_p\rho A_{d_{14}}\sqrt{2R}\delta_{1\le 4}\sum_{h=0}^{t-1}(T_4 - T_1)^{3/2} + \delta_{s_1}\epsilon\sigma A_{s_1}\sum_{h=0}^{t-1}\big(T_{s_1}^4 - T_1^4\big)$$

7.1 Grey box optimization

Regardless of the coefficients that we obtained in previous section, we consider whole experimental building as a grey box with associated Input and Output. Outputs are $[T_1, T_2, T_3, T_4]^T$ Temperature of the rooms the controlled inputs are $[\dot{m}_{pl_1} \dots \dot{m}_{pl_4}]^T$ and the exogenous inputs are $[T_{pl}\ T_c\ T_{out}]^T$. We can define $[\dot{m}_{d_{12}} \dots \dot{m}_{d_{41}} \dots\ T_{s_1} \dots T_{s_4}]^T$ as disturbances.

$$T_i = f(T_{pl}, T_c, T_{out}, T_i, \dot{m}_{pl_i}, T_{s_i}, \dot{m}_{d_{ij}})$$

With just knowing the theoretical relation between inputs and outputs we can write the grey box equations as follow:

$$\dot{T}_1 = \alpha_1\big(T_1 - T_{pl}\big) + F_1\beta_1\big(T_1 - T_{pl}\big) + \gamma_1(T_1 - T_c) + \delta_1(T_1 - T_{out}) + \varphi_1(T_1 - T_2) + \theta_1(T_1 - T_4) + S_1\mu_1\big(T_{s_1}^4 - {T_1}^4\big) + D_{12}\omega_1(T_2 - T_1)^{3/2} + D_{14}\tau_1(T_4 - T_1)^{3/2}$$

$$\dot{T}_2 = \alpha_2\big(T_2 - T_{pl}\big) + F_2\beta_2\big(T_2 - T_{pl}\big) + \gamma_2(T_2 - T_c) + \delta_2(T_2 - T_{out}) + \varphi_2(T_2 - T_1) + \theta_1(T_2 - T_3) + S_2\mu_2\big(T_{s_2}^4 - {T_2}^4\big) + D_{21}\omega_2(T_1 - T_2)^{3/2} + D_{23}\tau_2(T_3 - T_2)^{3/2}$$

$$\dot{T}_3 = \alpha_3\big(T_3 - T_{pl}\big) + F_3\beta_3\big(T_3 - T_{pl}\big) + \gamma_3(T_3 - T_c) + \delta_3(T_3 - T_{out}) + \varphi_3(T_3 - T_2) + \theta_3(T_3 - T_4) + S_3\mu_3\big(T_{s_3}^4 - {T_3}^4\big) + D_{32}\omega_3(T_1 - T_3)^{3/2} + D_{34}\tau_3(T_4 - T_3)^{3/2}$$

$$\dot{T}_4 = \alpha_4\big(T_4 - T_{pl}\big) + F_4\beta_4\big(T_4 - T_{pl}\big) + \gamma_4(T_4 - T_c) + \delta_4(T_4 - T_{out}) + \varphi_4(T_4 - T_3) + \theta_4(T_4 - T_1) + S_4\mu_4\big(T_{s_4}^4 - {T_4}^4\big) + D_{43}\omega_4(T_3 - T_4)^{3/2} + D_{41}\tau_4(T_1 - T_4)^{3/2}$$

F_i , S_i and D_{ij} are three Boolean value. F_i is "one" when the Fan in room i is running and "zero" when Fan is not working. S_i is "one" when the Heat source in room i is ON and "zero" when it is OFF. D_{ij} is "one" when Door between room i and j is open and also temperature of room j is bigger that temperature of room i and "zero" when relevant Door is closed or temperature of room j is not bigger that temperature of room i.

With dividing $S_i\mu_i\big(T_{s_i}^4 - {T_i}^4\big)$ to two different terms, $S_i\mu_i\big(T_{s_i}^4\big)$ and $S_i\vartheta_i\big({T_i}^4\big)$ we can also have temperature of the heat sources as one of optimized parameters.

We have 40 unknown parameters that should be identified.

$$\begin{matrix} \alpha_1 & \beta_1 & \gamma_1 & \delta_1 & \varphi_1 & \theta_1 & \mu_1 T_{s_1}^4 & \vartheta_1 & \omega_1 & \tau_1 \\ \alpha_2 & \beta_2 & \gamma_2 & \delta_2 & \varphi_2 & \theta_2 & \mu_2 T_{s_2}^4 & \vartheta_2 & \omega_2 & \tau_2 \\ \alpha_3 & \beta_3 & \gamma_3 & \delta_3 & \varphi_3 & \theta_3 & \mu_3 T_{s_3}^4 & \vartheta_3 & \omega_3 & \tau_3 \\ \alpha_4 & \beta_4 & \gamma_4 & \delta_4 & \varphi_4 & \theta_4 & \mu_4 T_{s_4}^4 & \vartheta_4 & \omega_4 & \tau_4 \end{matrix}$$

After taking the integral on above system, the Matlab optimization will be done using `lsqlin(C,d,[],[])` to solve $C * x = d$ with x the vector of the 40 unknown parameters, and $d = \begin{pmatrix} T_1(t_f) - T_1(t_0) \\ T_2(t_f) - T_2(t_0) \\ T_3(t_f) - T_3(t_0) \\ T_4(t_f) - T_4(t_0) \end{pmatrix}$, $T_i(t_f)$ Is value of T_i at the end of interval and $T_i(t_0)$ is value of T_i at the beginning of interval (We divide T_i to several linear parts, each part has its own equation). And C is the 4×40 matrix such that $C * x$ is equal to the integration of the right-hand part of the system of equation.

We solve this optimization problem with a recursive least-squares algorithm initialized with a set of values based on known physical parameters and observations.

The theoretical model is adapted to the measured behavior of our experiment with the following identification strategy. [2]

We run several experiments on the flat to capture the main behaviors modeled in grey box equation. In these tests, the outside temperature T_{out} is varying around 30 degree of centigrade, and the underfloor temperature T_{pl} is regulated at 17 degree of centigrade using a PID controller. Our experiments are quickly summarized as follows, with the desired physical phenomena.

- To observe the heat radiation, in each room, we turn ON a lamp, wait for an equilibrium to be reached, then turn the lamp OFF;
- for the air flow coming from the underfloor, in each room, we turn ON a fan, wait for an equilibrium to be reached, then turn the fan OFF;
- for the air flow through a door, we heat up one of the neighbor room with a lamp to create a temperature gradient, then open the door;
- Similarly to the previous test, we create a temperature gradient at the door by cooling down a neighbor room using a fan, then open the door. This test is also added to check the influence of the fans on the direction of the air flow through the door;
- in each room, we alternatively turn ON and OFF both fans and lamps to generate a data set that is sufficiently representative of the different operating conditions.

All these tests aim to quantify heat transfers due to heat radiation and the exchange of air flows as described above, as well as the conduction in the walls. parameters that obtained after optimization highly follow the behavior of the rooms' temperatures.

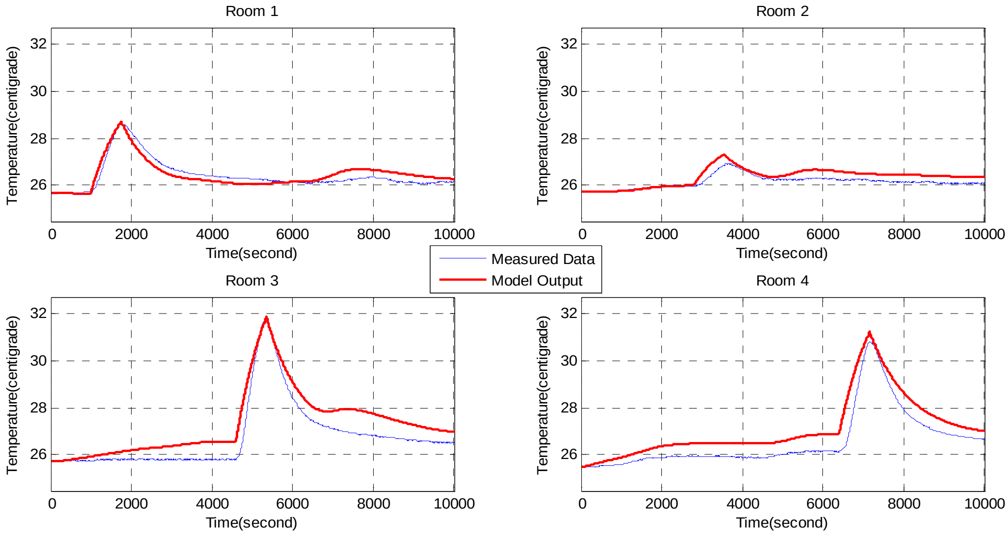


Figure 10- Data set 1(heat source effects)

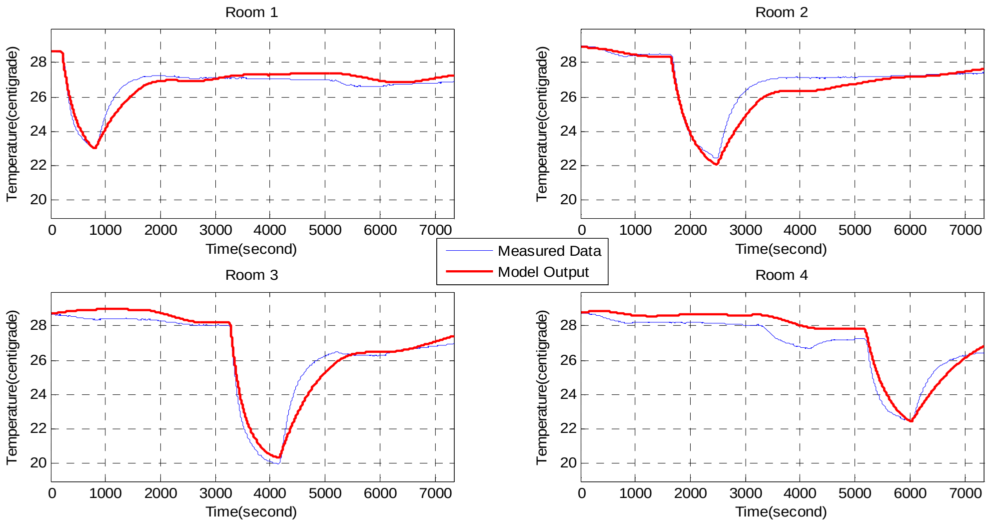


Figure 11-Data set 2(fans effects)

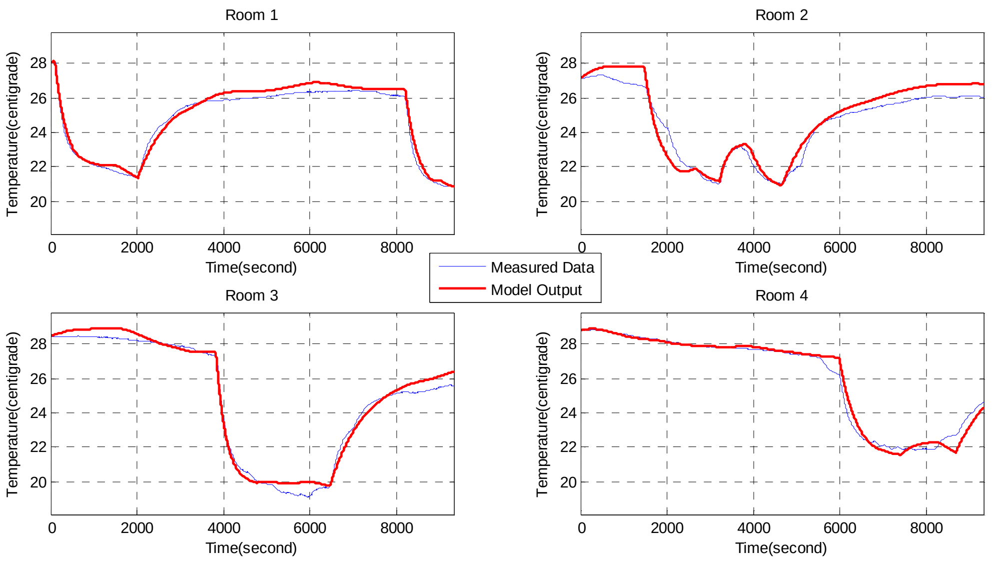


Figure 12-Data set3(fans and doors effects)

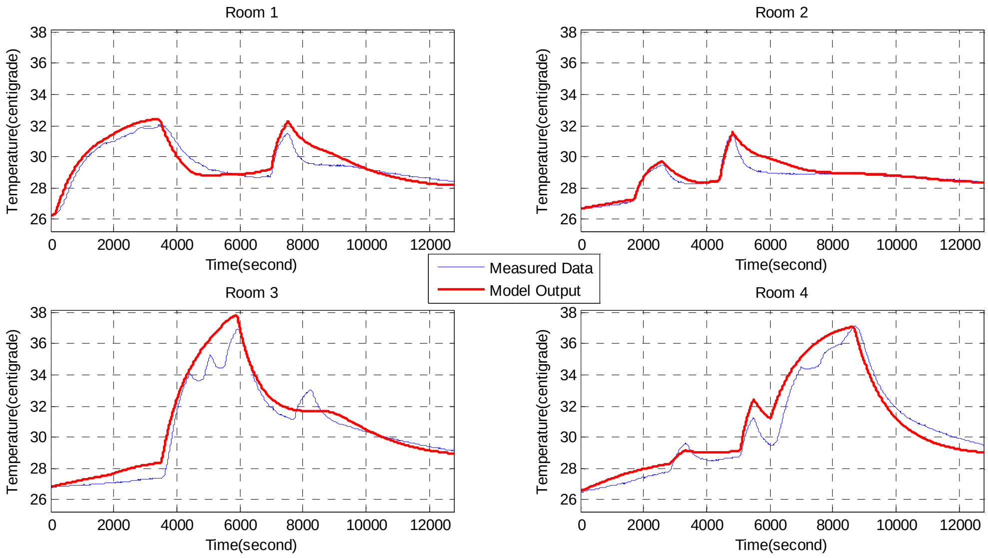


Figure 13-Data set4(heat source and doors effects)

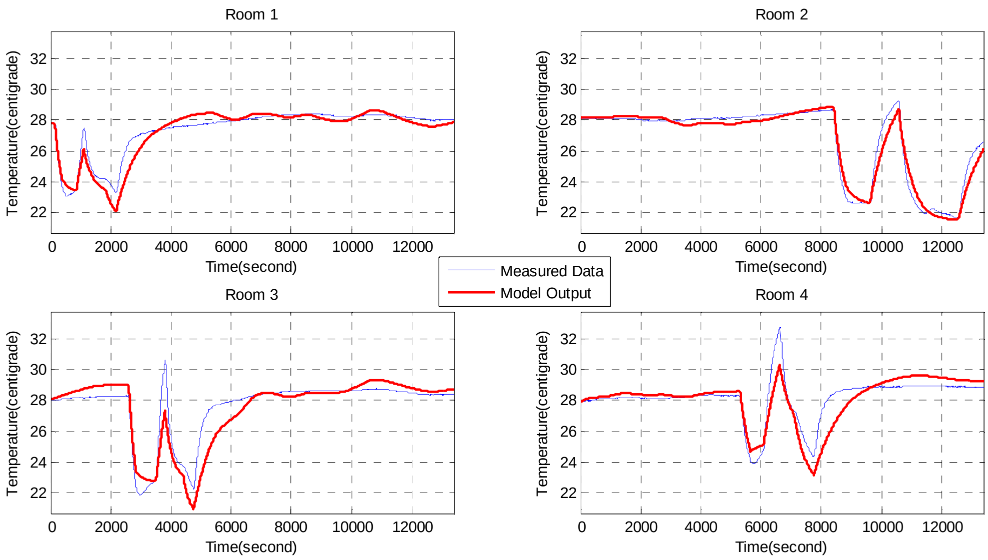


Figure 14-Data set5 (doors ,heat source and fans effects)

8 Model evaluation

The identified model is evaluated on experimental scenarios not included in the identification data set. An example of such scenario is described below, where we start with all lamps and fans off, and all doors closed.[2]

- After 120 seconds, the lamp in room 1 and the fan in room 3 are turned ON;
- After 420 seconds, the lamp in room 3 is turned ON and the door between 1 and 4 is opened;

- After 240 seconds, the fan in room 4 is turned ON;
- After 120 seconds, the lamp in room 3 is turned OFF and the door between 1 and 2 is opened;
- After 120 seconds, the door between 1 and 4 is closed.

This data set gathers the main situations that can happen in the building: conduction alone (rooms 2 and 4); a lamp (room 1); a fan (room 3); an open door (rooms 2 and 4); a lamp and a fan (room 3); a lamp and an open door (room 1); a fan and an open door (room 4).

The data measured during this experiment (noisy, blue) is displayed and compared to the model (dashed, red) in Figure15. The vertical lines correspond to the transitions described in the previous paragraph: solid when the switched element has a direct link to the corresponding room and dashed otherwise. For the most part, we can see in Figure15 that the model fits reasonably well the data. For this data set of 1211 points, the mean error between the data and the model is 0.06°C, with a standard deviation of 0.42.

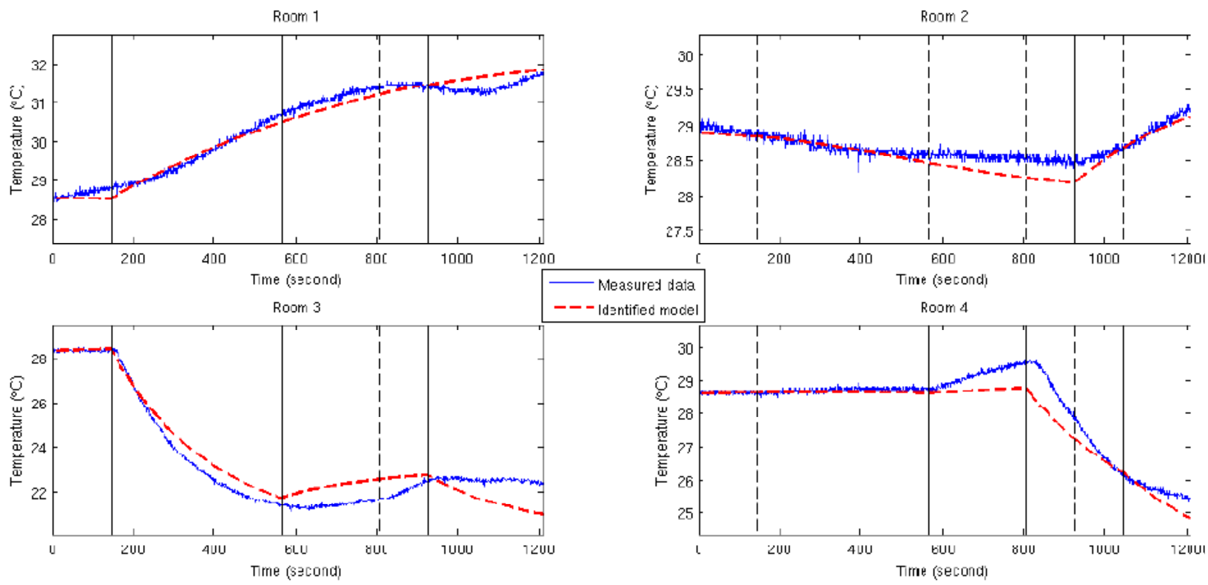


Figure 15-Comparison between the identified model (dashed, red) and the verification data set (noisy, blue), with the transitions (vertical lines: solid when linked to the room, dashed otherwise)

9 Do optimized values have Physical meaning?

From the grey box optimization, we have 40 optimized parameters; we can find values of convection and flow rates of the fans.

9.1 Convection

Convection can be obtained form α_i:

$$\alpha_{ij} = \frac{\frac{A}{(\frac{\Delta x}{k} + \frac{1}{H_i} + \frac{1}{H_j})}}{(\rho_{air} V_i C_v + \rho_{wall} \frac{V_{wall_{ij}}}{2} C_p)}$$

We can find some values for H to fit the alpha, but these values are not unique, it can be found several results that fit to alpha value. For evaluate the values of convection, we can refer to the standards tables as follow:

The Institute of Heating and Ventilation Engineers' 1970 edition Guide Book A (IHVE, 1970) provides a single expression for the calculation of forced convection. The original correlations were valid for wind speeds below 5 m/s but the guidebook makes no reference to this limitation. For buoyancy-driven convection in the absence of forced air movement, the guide recommends that the fixed values shown in Table 2 should be used.
$h_c = 5.8 + 4.1V$ (V: wind speed)

Surface orientation	h_c (W/m^{2}K)
Vertical surfaces	3.0
Top side of horizontal surfaces	4.3
Underside of horizontal surfaces	1.5

Table 2-Values for natural convection (source:IHVE, 1970)

Tables of internal and external surface resistances are also provided for 'sheltered', 'normal' and 'exposed' surfaces. These external values are based on the assumption that wind speeds are two-thirds of those measured at the roof height of the building. It is interesting to note that the effect of emissivity (i.e. radiation) is not significant for exposed locations such as the upper floors of high-rise buildings.

Building element	Emissivity	$h_r + h_c$ (W/m^{2}K)		
		Sheltered	Normal	Exposed
Wall	High	12.5	18.2	33.3
	Low	9.1	14.9	33.3
Roof	High	14.3	22.2	50.0
	Low	11.1	18.9	50.0

Table 3- Combined heat transfer coefficients for outside surface (source:IHVE, 1970)

British Standard gives the most comprehensive range of data for the calculation of rates of convection from buildings surfaces. At external surfaces, the following equation is provided where 'V' is the wind speed adjacent to the surface (in m/s)

$$h_c = 4 + 4V$$

At internal surfaces, three values are presented for heat flow upwards and downwards from horizontal surfaces, and heat flow from vertical surfaces. These values are given in Table 4 below and it can be seen that the values are different from those presented in previous sources (see for example Table 4). A reference for the source of these new values is not provided.

Surface orientation	h_c (W/m²K)
Vertical surfaces	2.5
Top side of horizontal surfaces	5.0
Underside of horizontal surfaces	0.7

Table 4-Values for natural convection (source: BSI,2007)

After we apply these standard constraints for obtain the value of convection in each area, we will find out that convection values can almost follow the standards. For instance, value of convection for the walls of room4 $[W/m^2K]$:

$H_{4pl} = 2.27$ (Floor)

$H_{4c} = 5$ (Ceiling)

$H_{4ow} = 3.33$ (Vertical walls common with outside)

$H_{43} = 1.44$ (Vertical walls common with room3)

$H_{41} = 0.96$ (Vertical walls common with room1)

9.2 Fans flow rate

Flow rate of the fan can be obtained form β_i:

$$\beta_i = \frac{C_p \dot{m}_{pl_i}}{(\rho_{air} V_i C_v + \rho_{wall} \frac{V_{wall}}{2} C_p)}$$

According to the data sheet of the fans, flow rate of the fans when supplied with $12VDC$ in room 2, 3 and 4 are 0.006 kg/s and in room 1 is 0.008 kg/s. In our experimental platform, the maximum output voltage that can supplies the fans is $10V$ (limitation of digital output modules). And also these kind of fans don't work with voltage supply less than $3V$.

After we apply values of fans flow rate that mentioned in data sheets of the fans as a constraints for obtain the value of fans flow rates in each rooms, we will find out that the values are in the operation range. (Note: Because of the big size of the fan in compression of the rooms' volume in the experimental building, we avoid to apply supply voltage more than 6VDC, and we assumed that the voltage and flow rate have linear relation $[3V - 6V]$ = $[0 - \text{max flow rate}]$)

Maximum Fans Flow Rate $[kg/s]$ (when supplied with 6VDC):

$Room1 = 0.0049$
$Room2 = 0.0028$
$Room3 = 0.0028$
$Room4 = 0.0020$

With applying the all real values, optimized convection values and fans flow rates in our model and compare with the verification Data set, we will see such a close behavior of parametric model and model with real values, as it shown in figure16.

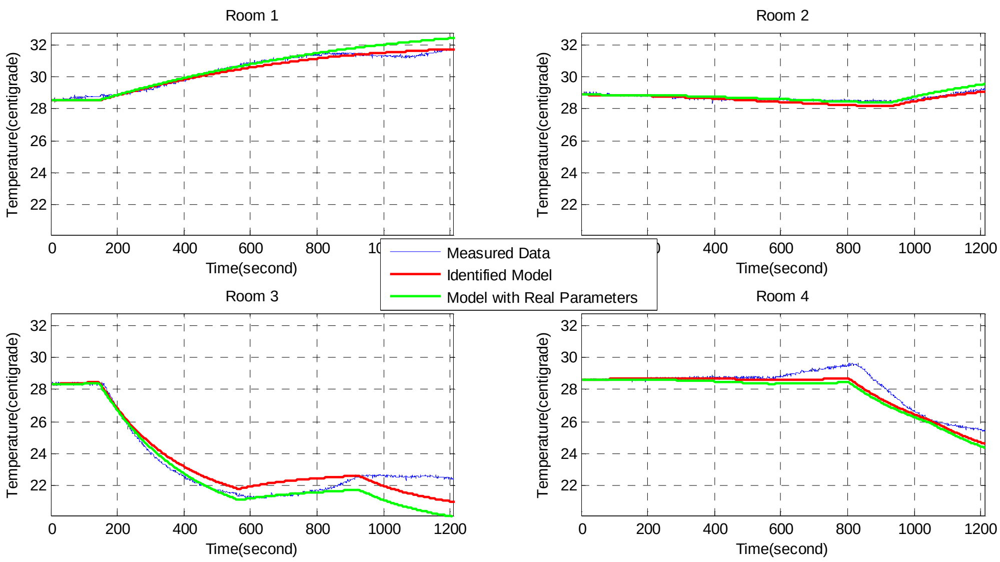


Figure 16-comparison between identified model, model with real Parameters and verification Data set

10 Controller

The control strategy that we applied on the experimental building was based on Robust Controlled Invariance [2]. Main goal is studying the possibility of a controller to keep the state of the system (T_i) inside a given interval for any external conditions (sources, doors, and exogenous temperatures). If the model of the building verifies the monotonicity property [5] the considerations on Robust Controlled Invariance are very simplified, such that we can only consider the extremal values of each variables. So when we obtained the parameter values fitting the real behavior of the building, we can use the obtained model to get theoretical limits $[\underline{T_i}, \overline{T_i}]$ for Robust Controlled Invariance. If an interval is chosen in these limits, any controlled should be able maintain the state in this interval (even in the worst conditions). We applied this Decentralized Linear Saturated (DLS) strategy to control the fan voltages:

$$\begin{cases} T_i \leq \underline{T_i} & V_i = 0 \\ \underline{T_i} \leq T_i \leq \overline{T_i} & V_i = \overline{V_i} \times \frac{T_i - \underline{T_i}}{\overline{T_i} - \underline{T_i}} \\ T_i \geq \overline{T_i} & V_i = \overline{V_i} \end{cases}$$

We aim to control the experiment so that each temperature stays in its target interval [$\underline{T_i}$, $\overline{T_i}$]. To ensure that the system encounters several disturbance situations, including both extremal conditions, we apply the following plan, starting when all doors are closed and all lamps are OFF:

t1) lamps 2 and 3 ON;
t2) doors 1-2 and 2-3 open;
t3) lamp 4 ON, door 3-4 open;
t4) lamp 3 OFF, doors 2-3 and 3-4 closed;
t5) lamp 4 OFF, door 4-1 open;
t6) lamp 1 ON, door 4-1 closed;
t7) all lamps ON, all doors open;
t8) all lamps OFF, all doors closed.

The results are displayed (in blue) in Figure17. The horizontal black lines are the boundaries of the target interval. The vertical lines represent the switching times in the schedule described above: solid lines when the switched component (door or lamp) is link to the room; dashed otherwise. In addition, the red dashed curves correspond to the simulation of this control using the identified model from Section7.

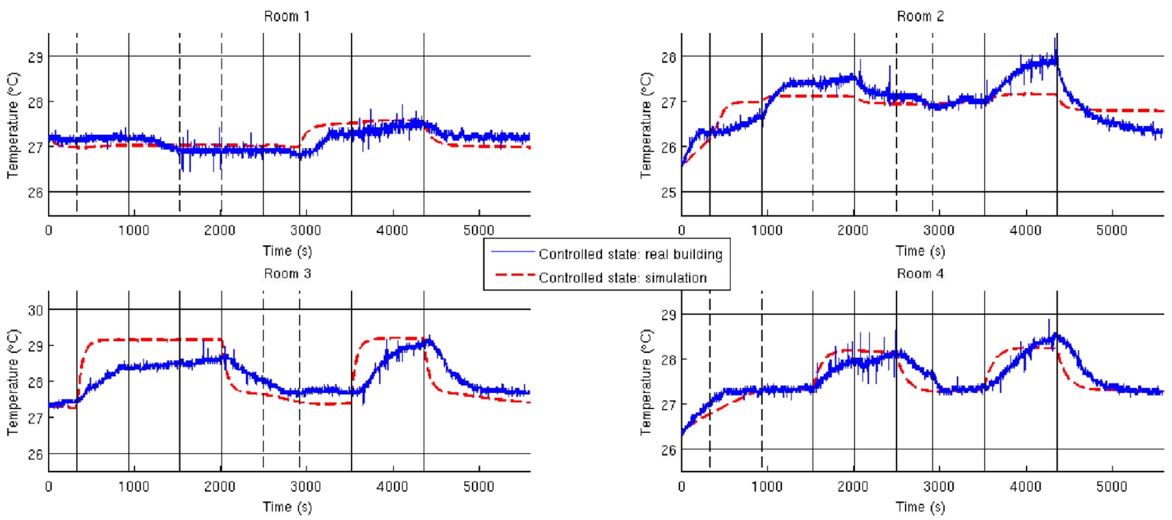


Figure 17- Controlled state, using DLS controller, for the experiment (blue, noisy) and the simulation (red, dashed), with the transitions (vertical lines: solid when linked to the room, dashed otherwise)

It is clear that the designed controller keeps the rooms temperatures inside of the target interval. For more detail, refer to [2].

11 Conclusion

In this report, I tried to express my major activities during my internship. Although I spend a lot of time to make this experimental building operational, but finally I succeed to do more than 100 tests in different condition from the building. For the optimization part, number of the tests that covers all the possible conditions was very important. I did the optimization with 7 different tests (Data set) to find reasonable values, which could follow the verification Data set behavior. Perhaps, with some other Data sets, we may obtain better results; though we should know that our model cannot 100 percent follow the data of the temperature sensors, Because this building made such a way that each room had not same effect from plenum area, even from outside area. For instance the return pipe outlet which brings the hot air from ceiling plenum, mounted so close to the fan inlet of room4, while we assumed that the inlet temperature of all fans are exactly same as the underfloor temperature.(this difference in some cases take more than $6°C$). Meanwhile, we totally ignored the sunlight effects coming through the windows; actually, this effect is not equal for all the temperatures sensors. This phenomenon can rise the measuring temperature up to $4°C$. However, I believe that the optimized parameters can represent our experimental building as well, beside, the implemented controller that was depending on these parameters, proof it.

12 References

[1] http://cbe.berkeley.edu/underfloorair/techoverview_pr.htm

[2] P.-J. Meyer, A. Girard, E. Witrant and H.Nazarpour Robust Controlled Invariance for UFAD Regulation. In Proceedings of the Conference on Buildsys, 2013.

[3] A Hybrid Model and MIMO Control for Intelligent Builings Temperature Regulation over WSN», By Emmanuel Witrant, Stéphane Monacu, Olivier Sename, Control Systems Department, UJF/ING, Université de Grenoble, 2009

[4] http://www.customthermoelectric.com/heatequations.html

[5] D. Angeli and E. D. Sontag. Monotone control systems. IEEE Transactions on Automatic Control, 48(10):1684–1698, 2003.