# CLIMA 2016 - proceedings of the 12th REHVA World Congress

*volume 8*

Heiselberg, Per Kvols

*Publication date:*
2016

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

Link to publication from Aalborg University

*Citation for published version (APA):*
Heiselberg, P. K. (Ed.) (2016). *CLIMA 2016 - proceedings of the 12th REHVA World Congress: volume 8*. Department of Civil Engineering, Aalborg University.

# Air Handling Unit Faults Impact on Thermal Comfort, Energy Consumption and Indoor Air Quality in an Office Building

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## Abstract

*Widely existing in heating, ventilating, and air conditioning (HVAC) systems, faults always lead to inefficient energy consumption, undesirable indoor air conditions (thermal discomfort and/or poor air quality), and even damage to the mechanical components. Although faults in HVAC systems can lead to disastrous consequences on buildings, this is a topic still little studied in the literature. In this work, the impact of three air handling unit faults is evaluated on the thermal comfort, the energy consumption and the indoor air quality in an office building. Due to their frequency and repeatability, the offset of the temperature sensor in the building, the leakage of the valve connected to cooling/heating coils, and the air leakage through the ducts are modeled and implemented in Dymola, using Modelica language. The obtained results are compared to a reference model of an AHU (Air Handling Unit) without faults, previously implemented in Dymola.*

***Keywords - HVAC; faults; thermal comfort; energy consumption.***

## 1 Introduction

The building sector represents 37 % of the total European Union energy consumption, [1]. Heating Ventilation and Air Conditioning (HVAC) is the most energy consumer. For a specific building in certain environmental condition, the energy consumed in HVAC depends on the type of the system and the correct operation. The incorrect operation of the HVAC systems and subsystems has effects not only on the energy consumption but also on the indoor air quality, [2]. Air Handling Unit (AHU) is one of the HVAC components where air properties are adjusted and then distributed to the whole space. With the growing interest of reducing buildings energy consumption, it is now important to focus on the study of the HVAC system faults. The major part of the research on the faults of the HVAC systems in general, and the AHU in particular comes with the objective of detecting and diagnosing these faults [3, 4, 5, 6, 7]. Due to their frequency and repeatability, we have chosen to study the following faults: the offset of the temperature sensor in the building, the leakage of the valve connected to cooling/heating coils, and the air leakage through the ducts. The impact of these three AHU faults is evaluated on the thermal comfort, the energy consumption and the indoor air quality in an office building. The research methodology is based on comparing a reference model without faults with different fault models. The reference model, composed of a building model and an AHU model, is implemented in Dymola [8], using Modelica modeling language [9, 10, 11]. Simulations were carried out and analyzed over both summer and winter periods, using the operating conditions for each season. Models for the three different faults (drifted sensor, water valve with leakage, air leakages in the air duct) were developed and implemented in Dymola.

## 2 Reference case

### 2.1 Office building and Air Handling Unit

The studied building, located in Trappes, close to Paris (France), is a single zone office of $10.25 \times 8.20 = 84.05$ m$^2$ and constructed out of four walls which are well isolated by mineral wool, a window (double glazing) in each wall and a door on the eastern wall. The southern window has a relatively larger surface area (10.1 m$^2$), this will enhance the effect of solar radiation. The building is considered as well isolated, in accordance with the French thermal regulation, [12].

The AHU considered in this study is a constant-air volume, balanced flows with a plate heat exhanger. Figure 1 shows the components of the AHU[1]. The air coming from outside is mixed with return air from the zone in the mixing chamber. In order to increase the energy efficiency of the air handling unit and as there is no need to introduce fresh air when the building is not occupied; the introduction of fresh air has been limited by the working hours of the building. In other words the economizer provides 180 m$^3$ h$^{-1}$ of fresh air from 9:00 a.m. till 6:00 p.m. during working days, and no fresh air otherwise[2]. Then a heat exchanger enables to enhance the heat transfer between this mixture and the return air, to recover the heat existing through the return air. The indoor air temperature is then controlled when air passes through cooling or heating coil depending on the season. The control is made through the water flow rate of the cooling or heating coil, using a PI controller. Finally, the air is circulated to and from the zone using supply fan

[1]The air flow rates are given here in terms of volume flow rate. However, in Dymola mass balances are used.

[2]Air flow rate per person has been taken as 30 m$^3$ h$^{-1}$, and as there is six occupants in the office, the total flow rate of fresh air equals to 180 m$^3$ h$^{-1}$. The return air flow rate is fixed to 300 m$^3$ h$^{-1}$.

and return fan.

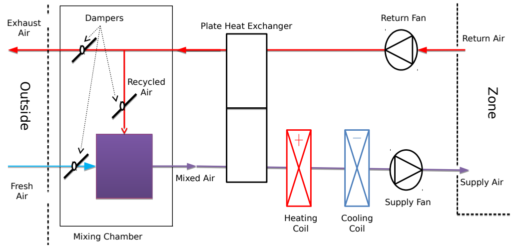



Figure 1: Components of the AHU.

### 2.2 Occupation scenarios and building operation

The indoor temperature set points are 26 °C in summer and 19 °C in winter. There are six occupants in the office and the parameters for each of occupation are determined according to the French thermal regulation, [13]:

- Occupation is from Monday to Friday, from 9:00 a.m. till 6:00 p.m.
- The heat gain per person is 90 W.
- The $CO_2$ emission per person is $1.10^{-5}$ kg s$^{-1}$.

In order to mitigate the effect of the solar radiation inside the space, shutters have been used in the following way;

- North, East and West windows are open from 9:00 a.m. till 6:00 p.m.
- South window from 7:00 a.m. to 9:00 a.m and from 5:00 p.m. to 7:00 p.m.

There are less opening hours for the southern window because it has a relatively large surface area, which enhances the effect of the solar radiation and so rises the indoor temperature significantly. The fresh air flow rate of ventilation is determined for one occupant, according to the health regulation in France to be 30 m$^3$ h$^{-1}$.

### 2.3 Preliminaries results

The AHU and building models have been coupled and implemented in Dymola, using the free open-source `Buildings` library [14]. Simulations have been conducted for the reference case model, during winter from the 1st of October till the 19th of May and during summer from 20th of May till 30th of September, recalling that the set point temperature in winter is 19 °C and 26 °C in summer. The time step of registering the simulation outputs was fifteen minutes.

The air flow rates in the AHU during occupation period (from 9:00 a.m. till 6:00 p.m. during working days) are given in Figure 2. They correspond to the established scenarios and this result enables to validate the AHU model implemented in Dymola. The $CO_2$ concentration in the building is 658 ppm, during the occupation period, which corresponds to an acceptable value according to the French thermal regulation, [12]. The mean indoor temperature and energy consumption depending on the season are given in Table 1. These results show that the heating energy consumption is nine times higher than the cooling energy consumption. This is in accordance with the climatic conditions of Trappes: the outside temperature is below the temperature set point during winter for 89% of the period, while during only 1% of the period, it is over the set point temperature during summer.

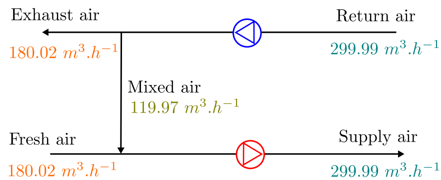



Figure 2: Air flow rates during occupation period.

| | **Summer** | **Winter** |
|---|---|---|
| **Mean indoor temperature** | 21.51°C | 19.62°C |
| **Energy consumption** | 246.18 kW h | 2273.38 kW h |

Table 1: Simulation results of the reference case model.

## 3 Sensor drift

The effect of a sensor drift on the overall system performance is crucial and can not be neglected. Indeed, the sensor measure is the input of the PI controller and if this value is drifted, the control will be wrong and the zone temperature will be different than desired.

Salsbury [15] proposed a model for sensor drift by adding an offset parameter to the actual temperature $T_{\mathrm{a}}$ of the space. This model has been adopted in this study using a constant offset parameter $D$, as explained by:

$$T_s = T_a + D \tag{1}$$

where $T_s$ is the sensor reading. The implementation of the model in Equation (1) in Dymola was done by adding a constant offset parameter to the feedback sent to the PI controller for the heating and cooling coil.

As in the reference case, the simulations have been carried out during two periods. Winter from the 1st of October till the 20th of May, and summer from 20th of May till the 1st of October. The temperature set point in winter was 19 °C and 26 °C in summer. Twelve simulations (six in winter and six in summer) for the offset parameters ($D = -1.5; -1; -0.5; +0.5; +1; +1.5$ °C) have been conducted. The data, issued from simulations, have been studied and analyzed with a time step of fifteen minutes.

The results are listed in Table 2. A positive drift decreases the set point temperature and vice versa. It seems that the offset parameter changes directly the set point temperature. This can be easily explained. From the feedback of the indoor temperature sensor and the set point temperature and considering the Equation (1), the error $e$ of the PI controller is calculated as;

$$e = T_{sp} - T_s = \underbrace{(T_{sp} - D)}_{T_{sp}^{new}} - T_a \tag{2}$$

where $T_{sp}$ is the set point temperature and $T_s$ is the sensor reading. As a result, the energy consumption increases significantly when decreasing the value of the offset parameter in winter and when increasing the value of the offset parameter in summer. The results in terms of energy consumption in Table 2 confirm this.

| $D$ (°C) | **summer** | | **winter** | |
|---|---|---|---|---|
| | $T_{mean}$ (°C) | $C$ (kW h) | $T_{mean}$ (°C) | $C$ (kW h) |
| +1.5 | 20.99 | 363.13 | 18.35 | 1876.70 |
| +1.0 | 21.16 | 328.33 | 18.76 | 2005.44 |
| +0.5 | 21.34 | 295.19 | 19.19 | 2137.65 |
| 0 | 21.51 | 264.18 | 19.62 | 2273.38 |
| -0.5 | 21.67 | 235.11 | 20.06 | 2410.86 |
| -1.0 | 21.82 | 207.55 | 20.50 | 2551.04 |
| -1.5 | 21.96 | 181.49 | 20.95 | 2692.90 |

Table 2: Mean indoor temperature ($T_{mean}$) and energy consumption ($C$) as a function of the offset parameter.

## 4 Valve leakage

One of the degradation faults of the valves is the valve leakage at the full closed position. This happens when we attain the wished indoor temperature, in other words the zone temperature matches the set point temperature. In this case the error is zero and the output control signal from the PI controller also has zero value. As a result the hot or chiller water valve must be closed. Although the valve is fully closed, there is still water flow rate running through the heating or cooling coil.

The used valve model has been implemented in the two way linear valve model of `Buildings` Library. The leakage coefficient $l$ of the two way valve model is the ratio of the $K_v$ factor of the valve in closed position ($y = 0$) and the $K_v$ at the fully open position ($y = 1$):

$$l = \frac{K_v(y = 0)}{K_v(y = 1)} \tag{3}$$

The performance of the AHU was analyzed for four values of hot/chilled water valve leakage: 1%, 2%, 5% and 10%. In order to easily interpret the simulation results after, another leakage parameter has been chosen. The selected parameter is the ratio of the leak mass flow rate over the maximum flow rate of the valve $l_m$.

The same method for sensor drift has been applied for the valve leakage in summer and winter. The results are given in Table 3. The mean indoor temperature is inversely proportional with the valve leakage during the summer period and proportional in winter. The effect of valve leakage is also quite significant, since the energy consumption can be multiplied by three if the valve leakage reaches 10 % during summer. Although the heating coil works more time in comparison with the cooling coil, the increase of energy consumption during the winter period is less significant.

| $l_m$ | **summer** | | **winter** | |
|---|---|---|---|---|
| | $T_{\text{mean}}$ (°C) | $C$ (kW h) | $T_{\text{mean}}$ (°C) | $C$ (kW h) |
| 0 % | 21.51 | 264.18 | 19.62 | 2273.38 |
| 1 % | 21.21 | 316.11 | 19.74 | 2314.63 |
| 2 % | 20.67 | 409.43 | 20.02 | 2403.12 |
| 5 % | 19.83 | 558.78 | 20.53 | 2560.79 |
| 10 % | 18.90 | 724.37 | 21.39 | 2833.64 |

Table 3: Mean indoor temperature ($T_{\text{mean}}$) and energy consumption ($C$) as a function of the leakage parameter $l_m$.

## 5 Duct leakage model

Due to the important effect of the duct faults in the HVAC system performance the HVAC engineer has to take it into account since the design process. ASHRAE (*American Society of Heating, Refrigerating and Air-Conditioning Engineers*) [16] classified the ducts into leakage classes, in order to study the effect of duct leakage on energy consumption and indoor air environment. Equation (4) gives the relation between the pressure and the leakage flow rate for a specific duct class:

$$F = 0.0014 C_L \Delta P^{0.65} \tag{4}$$

with $F$ the duct leakage rate ($\mathrm{l\,s^{-1}\,m^{-2}}$), $C_L$ the duct class (-) and $\Delta P$ the static pressure differential from duct interior to exterior (Pa). Considering the air density $\rho = 1.2$ $\mathrm{kg\,m^{-3}}$ and a duct surface area of 7.85 $\mathrm{m^2}$, the Equation (4) can be written as a function of the mass flow rate such as:

$$\dot{m} = 1.32.10^{-5} C_L \Delta P^{0.65} \tag{5}$$

where $\dot{m}$ represents the mass flow rate ($\mathrm{kg\,s^{-1}}$). Ducts of class 3 and class 12, according to ASHRAE [16] classification, have been used and implemented.

The air flow rate entering the inlet duct is divided into two flow rates; leakage flow and the outlet flow. The quantity of these air flow rates depends on flow resistance on the leakage opening and the duct. Therefore, the duct leakage model has been implemented as a constant resistance model, using the `FixedResistances` model in `Buildings` Library.

The simulations have been conducted for the two duct classes. Each duct class has been implemented in two places upstream and downstream the fan, see Figure 3. Two types of leaks were simulated: one upstream of the fan where the pressure is lower than the atmospheric pressure and so air enters into the duct and another, downstream of the fan where the pressure is higher than the atmospheric pressure and the air leaks outside of the duct.

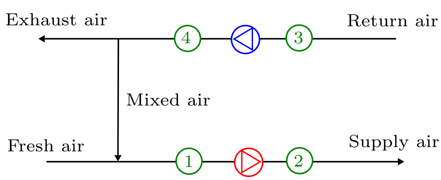
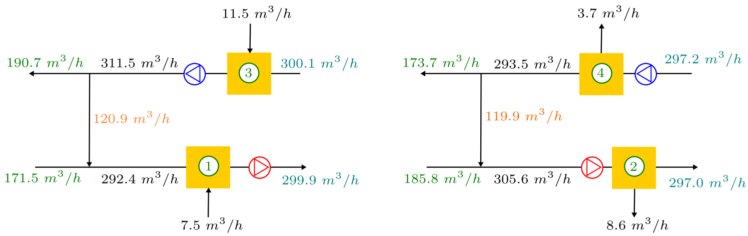


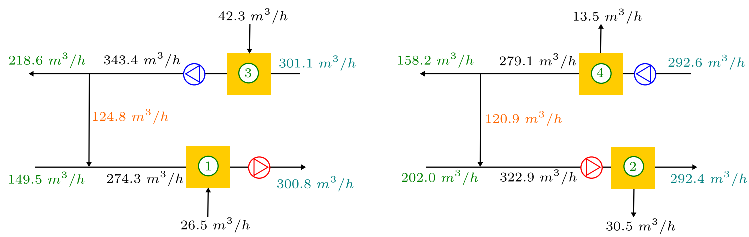

Figure 3: Positions of the duct leakage model implemented in the AHU.

In the duct leakage model the air pressure and flow rate will vary across all the component of the AHU. In order to well understand and analyze the results for thermal comfort and energy consumption, it is helpful to look at the air flow rates in the AHU. Figures 4(a), 4(b), 5(a) and 5(b) illustrate the air flow rates through the AHU. Comparing Figure 4(a) and Figure 5(a) with the reference case (Figure 2) shows that the air flow runs inside the duct, when the duct leakage model is placed upstream to the fan. This is because the pressure difference in this case is negative, i.e. before the fan the atmospheric pressure is higher than the pressure inside the duct. In addition, it is found that the air flow entering from the return air duct is more that the one entering the supply duct in both cases. On the contrary, when placing the same duct leakage model downstream the fan, the air flow runs outside the duct. The same conclusion about a higher flow rate for the duct of class 12 and duct of class 3 is noticed. Meanwhile the leakages in the return duct have lower air flow rates.




(a) upstream (b) downstream

Figure 4: Air flow rates in the AHU with a duct leakage model of class 3.




(a) upstream (b) downstream

Figure 5: Air flow rates in the AHU with a duct leakage model of class 12.

The results in $CO_2$ concentration, indoor temperature and energy consumption are given in Table 4 for each studied duct class and position. Whatever the duct leakage, the $CO_2$ concentration is slightly lower than that obtained with the reference model. This is easily explained; air leakage causes fresh air flow (incoming or outgoing) and thus the $CO_2$ concentration decreases. For an air leakage upstream of the fan, the decrease is more pronounced, in particular for $C_L = 12$. The mean indoor temperature decreases slightly with a duct leakage. For this AHU fault, the influence of the leakage thermal comfort is low and the energy consumption is less than the reference case. In summer, the energy consumption for the downstream models is less than the one of the upstream. This might be due to the large percentage of the fresh air mixed with the return air in the mixing chamber. This assumption would put in question recirculation of the return air during summer, as it appears that more fresh air is introduced to the system, less energy is consumed. Concerning the winter period, as the air outside temperature is mostly lower that the zone temperature, the energy consumption is more important than the reference case. Only a slight change is noticed between the models upstream and downstream.

| | | $C_L = 3$ | | $C_L = 12$ | | **reference** |
|---|---|---|---|---|---|---|
| | | **upstream** | **downstream** | **upstream** | **downstream** | |
| $CO_2$ **concentration (ppm)** | - | 645 | 656 | 618 | 647 | 658 |
| **Mean indoor temperature** | **summer** | 21.32 | 21.35 | 21.01 | 21.11 | 21.51 |
| (°C) | **winter** | 19.00 | 19.56 | 19.45 | 19.50 | 19.62 |
| **Energy consumption** | **summer** | 260.55 | 223.05 | 256.63 | 223.05 | 264.18 |
| (kW h) | **winter** | 2461.56 | 2469.53 | 2885.95 | 2954.67 | 2273.38 |

Table 4: Summary of the results for the duct leakage model.

## 6 Conclusion

In this paper, the impact of three faults of an AHU was studied on the indoor temperature and energy consumption of an office building using Dymola. The developed fault models and the obtained results show clearly that the effect of the faults can be significant in the operation of the AHU. In the case of the sensor drift, a positive drift decreases the set point temperature and vice versa. Obviously, the energy consumption increases with the offset parameter in summer and decreases with the offset parameter in winter. The performance of the AHU was analyzed for four values of hot/chilled water valve leakage: 1%, 2%, 5% and 10%. The results show that the valve leakage is a key point of the valve efficiency. It causes serious consequences on the thermal comfort, until -2.6°C in summer and +1.77°C in winter. Furthermore, it increases the energy consumption of the building (2.75 times in summer and 1.25 times in winter). The last model is the duct leakage model, as a function of duct leakage class and position in AHU. Duct leakage has a favorable effect on the indoor air quality, the $CO_2$ concentration decreases in all cases. During both summer and winter, the indoor temperature decreases with duct leakage. This causes a decrease of the energy consumption during summer, and an increase in heating energy consumption during winter. The effect of duct class is more important than the position of the duct on indoor air conditions and energy consumption.


## Acknowledgment

The authors gratefully acknowledge the financial support provided by the Agence Nationale de la Recherche (ANR-12-VBDU-0005).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