A PARAMETRIC INVESTIGATION IN FEEDBACK CONTROL OF A HVAC SYSTEM FOR A SINGLE ZONE WITH SENSIBLE LOAD

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S1 – Sistemas de Ar Condicionado e Aquecimento

Abstract. *In this work was investigated the transient response of a single zone conditioned by a VAV(variable volume flow) under a local loop proportional HVAC controller. For the sake of simplicity only the sensible load will be considered in the zone. The model will be based on the lumped capacitance approach, with a single capacitance value for the zone air and interior furnishings. The system will be assumed to be linear. The different types of response were compared under the influence of different physical and controller parameters. The following physical parameters were considered : barometric pressure, air humidity, zone capacitance, sensible load and thermostat inertia. The following controler parameters were accounted for : Time delay, zone setpoint, deadband , gain, bias*

Keywords : *Feedback control systems, single zone load, VAV*

1. INTRODUCTION

Control strategies for heating, ventilation and air-conditioning (HVAC) systems are complex due to the fact that there are a great number of variables to be controlled. Mitchell and Braun (2013) instanced this issue showing a scheme of a central chilled water facility used to provide air conditioning in a large building. These authors argued that the control system in this building involve the control of many proprieties such as the air, the water and the refrigerant loops and the airflow rate and supply temperature to each zone of the facility to meet the necessary sensible and latent loads of each zone. In addition, they pointed that the control system algorithm (on-off, proportional, PI and PID control) affect the control efficiency related to how the system will distance itself from the setpoint and how fast the system will approximate this setpoint, related to the time for the control system to act over disturbances.

In this HVAC control complexity context, there have been a lot of recent studies which aims to reduce costs by saving energy with optimization studies. Aswani *et al.* (2012) emphasize that HVAC systems contribute to a significant fraction of building energy usage and these systems have seen an increasing amount of research towards their modeling and efficient control. These authors presented quantitative metrics for comparing the energy usage and comfort of different HVAC controllers and they also argued how difficult it is to be done as variations in weather and occupancy conditions preclude the possibility of establishing equivalent experimental conditions.

Another HVAC system aspect is that the room temperature usually varies over time. Riederer *et al.* (2002) explained the importance of dynamic models which does not consider the room air as perfectly mixed. These authors observed that the air temperature at the position of a controller sensor and at the center of a room can differ in its static as well as in its transient behavior. In addition to that, according to Canbay *et al.* (2004) HVAC processes are non-linear and characteristics change on a seasonal basis so the effect of changing the control strategy is usually difficult to predict.

As a result of all these issues that are needed to be controlled in HVAC control systems, there are significant number of models and simulations about this topic in literature. These models are usually strategies to enhance comfort and minimize energy usage as described by Mathews *et al.* (2001) who investigated different control strategies by studying air-bypass, reset control, setback control, improved start-stop times, economiser control and CO_2 control. In this study, the simulation models were verified against measurements and it was possible to ensure comfort and to predict savings of 60% in HVAC power consumption.

Although this large HVAC control modeling and simulation studies, there are few experimental tests of advanced HVAC controllers on physical systems according to Anderson *et al.* (2007) who also added that the PI (proportional-integral) controllers continue to dominate commercial HVAC systems. It is important this experimental procedure because experimental data can validate models and indicate their application.

2. BASIC EQUATIONS- METHODOLOGY

A simplified model for the transient response of a zone will be used in comparing the different control strategies for a local feedback control loop for a VAV box. Fig. (1) presents the block diagram of the control loop.

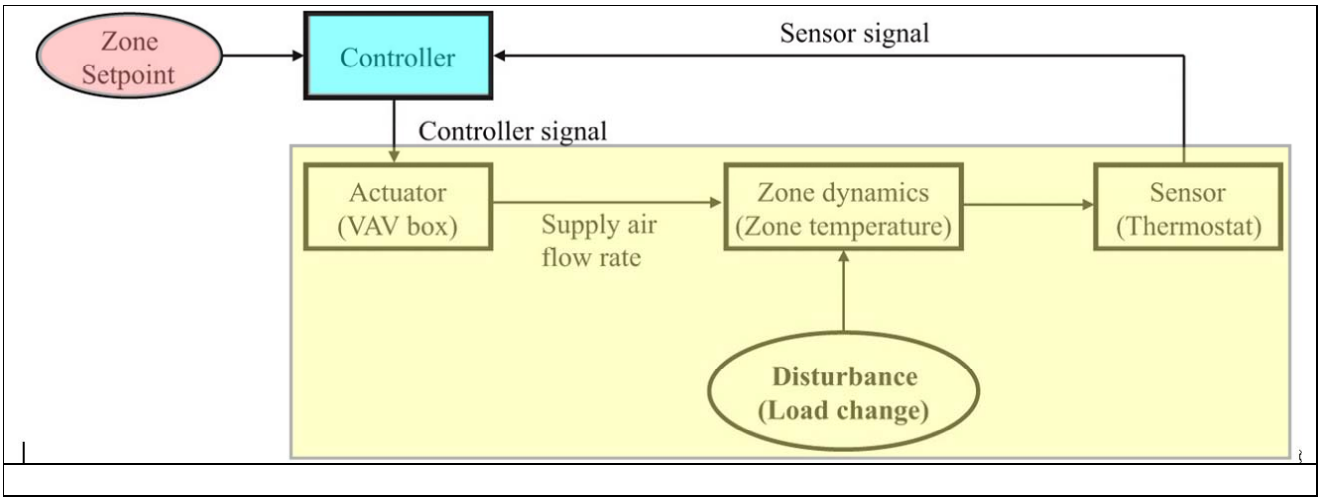

Figure 1 - Block diagram of the control loop (adapted from Mitchell and Braun, 2013)

The disturbance is related to the amount of energy that enters the room causing temperature variations over time. Based on this behavior it is necessary a dynamic model to describe the zone temperature variation in a specific position where the sensor is located. This sensor is responsible to send a signal that contains the temperature measure to the controller. The setpoint is the value of the sensor temperature desired. Depending on the difference between the desired temperature and the real temperature sent by the sensor signal, the controller will transmit a signal to the actuator that will manipulate the VAV box. The supply air flow rate is controlled by the VAV box and this flow rate affects directly the zone temperature. So if the disturbance modifies significantly the room temperature, the actuator will provide the necessary air flow rate for the room to achieve the setpoint temperature by opening or closing the VAV box. As a result of this, the control strategy will affect the way the VAV box will be opened or closed and will also affect the time for the system to response over disturbance and to achieve the setpoint temperature equilibrium.

The zone dynamics is mathematically represented by Eq. (1) which is the differential energy balance equation. C_z is the zone thermal capacitance in J/C ; T_z is the zone temperature in C, t is the time in s, $\dot{L}_S$ is the sensible thermal load in W, $\dot{m}_s$ is the supply air flow rate in kg/s, c_p is the air specific heat in J/kg C, and T_s is the supply air temperature. The zone air temperature is assumed to be at a single temperature.

$$C_z \frac{dT_z}{dt} = \dot{L}_S - \dot{m}_s c_p \left(T_z - T_s\right) \tag{1}$$

The supply air flow rate from the VAV box responds linearly and directly proportional to the control signal γ :

$$\dot{m}_s = \rho_s \dot{\forall}_{s,max}\ \gamma \tag{2}$$

Where

ρ_s is the supply air density and $\dot{\forall}_{s,max}$ is the maximum volumetric flow rate in m^3/s, and γ is defined by :

$$\gamma = K_p E + M + \frac{1}{\tau_I}\int_0^t E\ dt \tag{3}$$

In the previous equation K_p is the gain in 1/C, E is the error, i.e., the difference between the zone temperature and the setpoint temperature T_{set} . M is the bias, i.e., the value of the signal when the error is zero. τ_I in last term in Eq. (3) is the integral time constant for the controller.

When the sensor dynamics is considered, the error in Eq. (3) is the difference between the thermostat temperature and T_{set}. . The energy balance for the thermostat must be solved in order to have the thermostat temperature :

$$\tau_{th} \frac{dT_{th}}{dt} = -\left[T_{th} - T_z\right] \tag{4}$$

When the two first terms of eq. (3) is used the control is termed proportional and the inclusion of the last term represents the proportional integral control. When γ in eq. (3) assumes two discrete values 1 or 0, the control is termed On-Off.

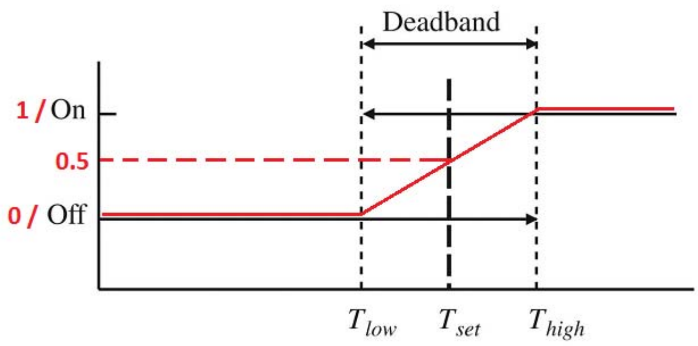

Figure 2. On-Off and Proportional control schemes (adapted from Mitchell and Braun, 2013)

Figure 2 presents schemes for On-Off and proportional control strategies that are related to how the system will react over disturbances. As a load change occurs in the system, the temperature measured by the thermostat will vary over time. The control strategy defines the way the actuator will manipulate the damper position for the zone temperature return on its setpoint value. The on-off technique is based on opening or closing completely the damper when the temperature is too hot (T_{high}) or too cold (T_{low}), respectively. The deadband is the difference between these two temperatures and this range is necessary as to avoid rapid cycling of the VAV box damper. The γ variable for the proportional control is plotted in red in Fig. (2). The proportional control strategy consists on opening or closing the damper proportionally to the error (the difference between the zone and the setpoint temperatures), namely the damper is not opened or closed completely at once. The farther the system is from the desired temperature, the greater the response will be. The 0 and 1 signals represent that the damper is completely closed and opened, respectively. For the situation depicted in Fig. 2, the bias is 0.5.

3. RESULTS AND DISCUSSION

The mathematical model for the control system (Eqs. (1), (2), (3)) is composed of a diferential equation with variable coeficients. The treatment of the system by use of traditional methods, in the Laplace domain is not the most adequate. In this way, the calculation procedure described in the previous section was implemented in the time domain for a system such as the one depicted in Fig 1, using the thermodynamics software EES (Klein, 1993). Table 1 depicts the parameters value in the baseline simulation.

Table 1. Baseline parameters

Parameter	Value
$\dot{L}_S$	3.0×10^4 W
C_z	1.5×10^7 J/C
T_{set}	21 C
T_s	13 C
$\dot{\forall}_{s,max}$	9 m^3/s

The atmospheric pressure value used for the baseline simulations were 1 atm and the relative humidity 50 %.

For the On-Off control the dead band was 2 C. The time simulation was 60 min with the zone initially at the set point and the control in the Off position, i.e., no air supplied to the zone.

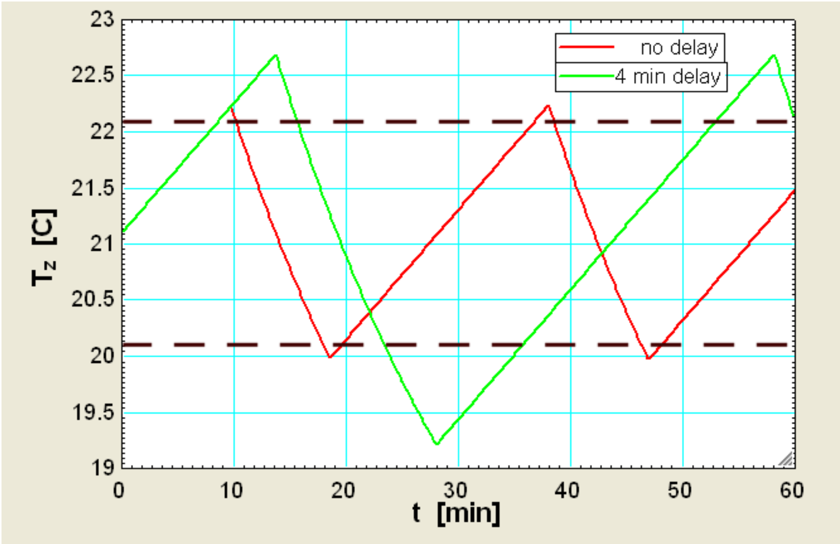

Figure 3. Effect of control action delay on the zone temperature response for an On-Off control

In Figure (3) is presented the effect of action delay in the zone temperature response. According to the figure, the time delay increases the overshooting. It also can be noticed the difference between the overshooting superior and the overshooting inferior. This difference can be attributed to the difference at the mass flow rate. Due to time delay and the high density of the air, the zone is cooled with a high-mass flow rate and the temperature drop more in the lower overshoot. The time delay used for the next simulations was 2 min.

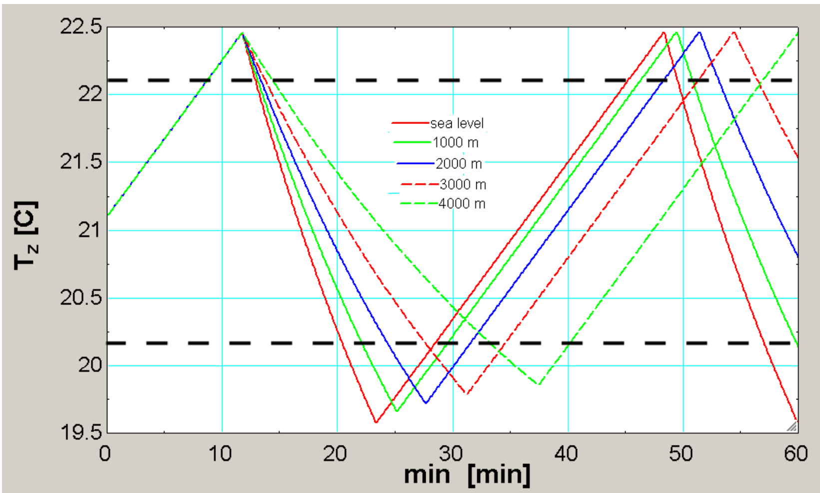

Figure 4 - Effect of altitude on the zone temperature response for an On-Off control

In Figure (4) is presented the altitude effect on the zone temperature. Figure analysis shows that during the cooling cycle, the rate of temperature drop with time is more intense for the lower temperature. This can be attributed to the greater mass flow rate. From equation (2), the greater flow rate is result of higher air density at low altitudes that corresponds to lower barometric pressures. The greater inferior overshoot for low altitude can be also attributed to the high mass flow rate due to higher density. It was also verified that the results for different air humidities were not affected. The reason can be attributed to c_p and ρ being weak functions of humidity for dry air –water vapor mixtures.

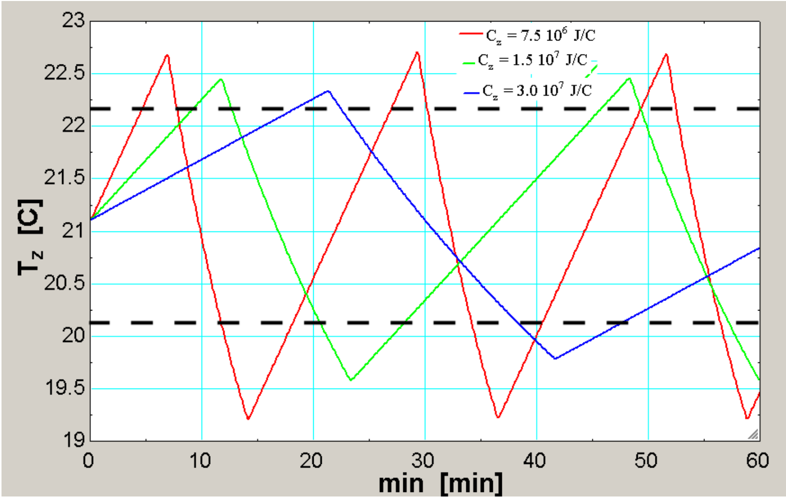

Figure 5 – Effect of zone capacitance on the zone temperature response for an On-Off control

In Figure (5) is presented the effect of the zone capacitance. Figure analysis shows that higher capacitance results in lower rates of temperature variation during the heating and cooling periods. Besides, lower number of oscillations with time, the overshooting is diminished for higher capacitances.

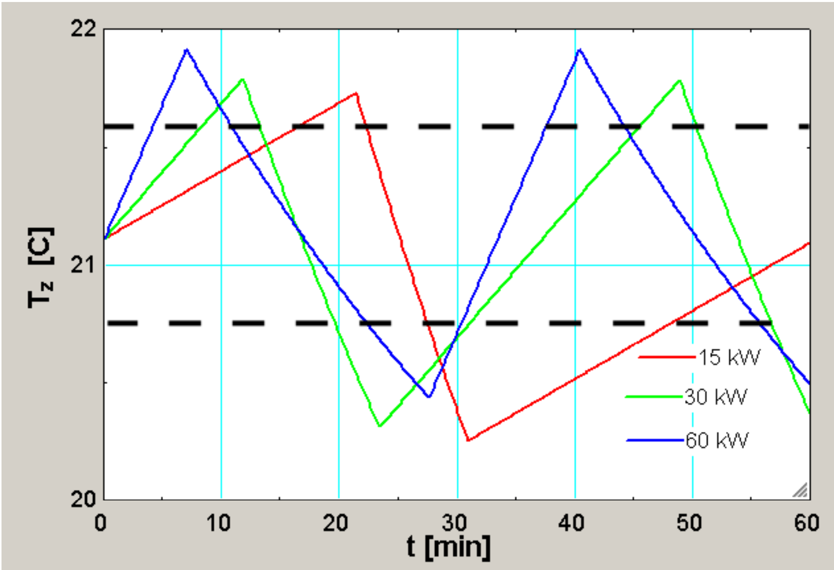

Figure 6 – Effect of zone sensible load on the zone temperature response for an On-Off control

In Figure (6) is presented the effect of increasing sensible load. From the figure, it can be seem the zone temperature increases more rapidly for greater loads since the zone capacitance is the same for all loads. Figure analysis also shows the number of cycles of variation is lower for the lower load. The differences in the superior and inferior overshootings are due to time delay.

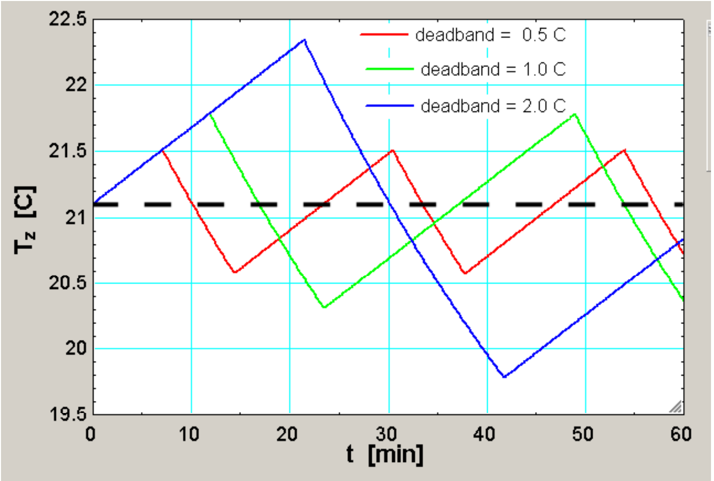

Figure 7 - Effect of deadband on the zone temperature response for an On-Off control

In Figure. (7) is presented the effect of the deadband choice for the On-Off control. For the same load the temperature variations are greater for the larger deadbands. However, the degree of overshooting stays equal to all deadband choices.

For the next figures for the proportional controller, the simulations are started considering a steady-state situation. The zone initial temperature in this steady-state situation is obtained from eq. (1) , dropping the transient term and using eqs. (2) and (3). The bias used was 0.5. All the applicable conditions are the same as the On-Off control according to Table 1. The zone response when the load is doubled is studied for different values of gain K_p.

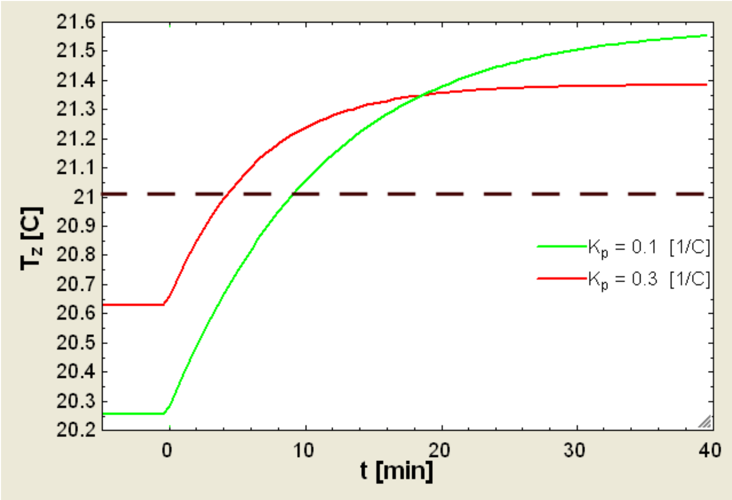

Figure 8 - Effect of gain on the zone temperature response for a proportional control

Figure 8 shows the system response using a proportional control for different gain values. Figure analysis shows that with proportional control, the zone temperature increases rapidly and then asymptotically aproaches the steady state value. The system response is faster for greater gains K_p . The offset, i.e., the difference between the setpoint and the steady temperature, is lower for greater gains. By his turn, the time to reach a new steady temperature depends upon the flow rate and zone thermal capacitance.

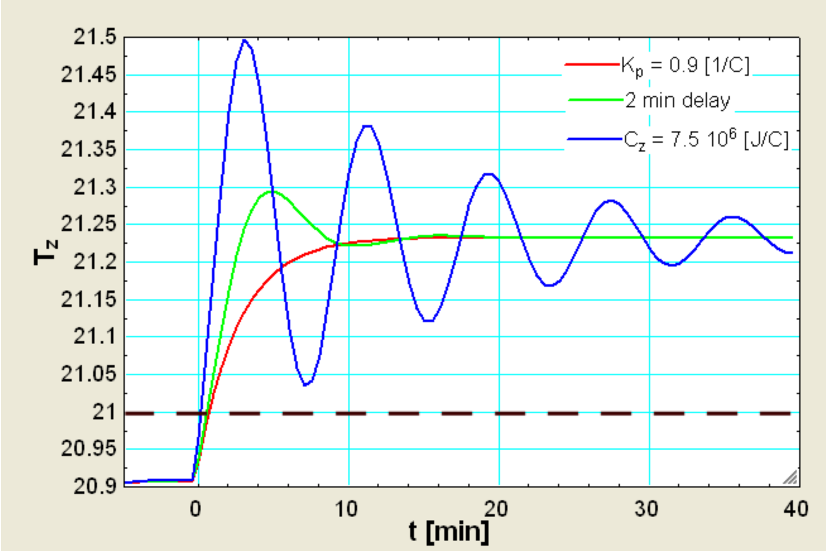

Figure 9 – Cumulative effect of high gain, time delay and low capacitance on the zone temperature response for a proportional control

In Fig. 9 is presented the cumulative effect of gain, time delay and capacitance. All the other parameters are kept the same as the previous cases for the proportional control. Figure analysis shows that increase of the gain K_p results in a lower offset. By his turn, the introduction of time delay and lower capacitance results in overshooting and oscillations.

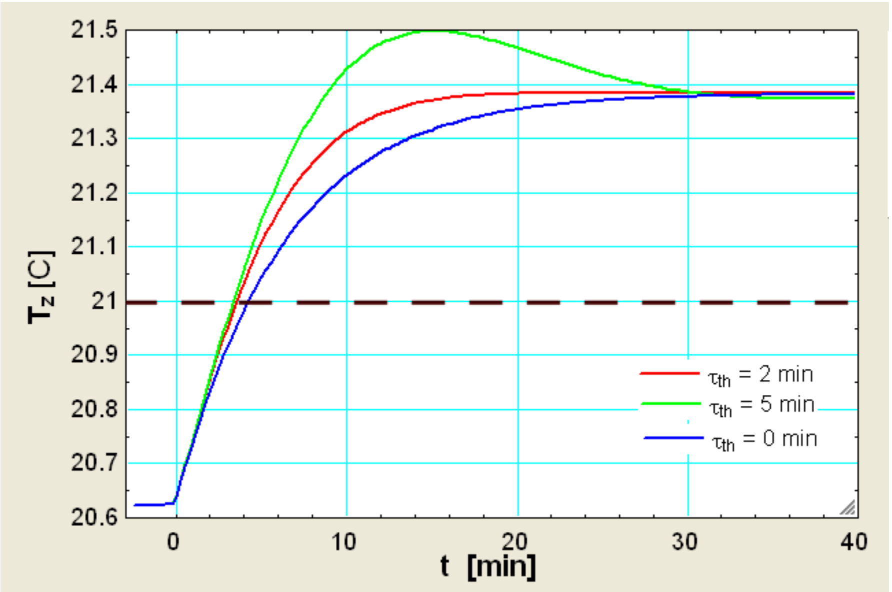

Figure 10 – Thermostat inertia effect on the zone temperature response for a proportional control

In Fig 10 is presented the thermostat effect for a proportional control. Analysis shows that the offset is not modified, but the overshoot increases with the increase of the time constant of the thermostat.

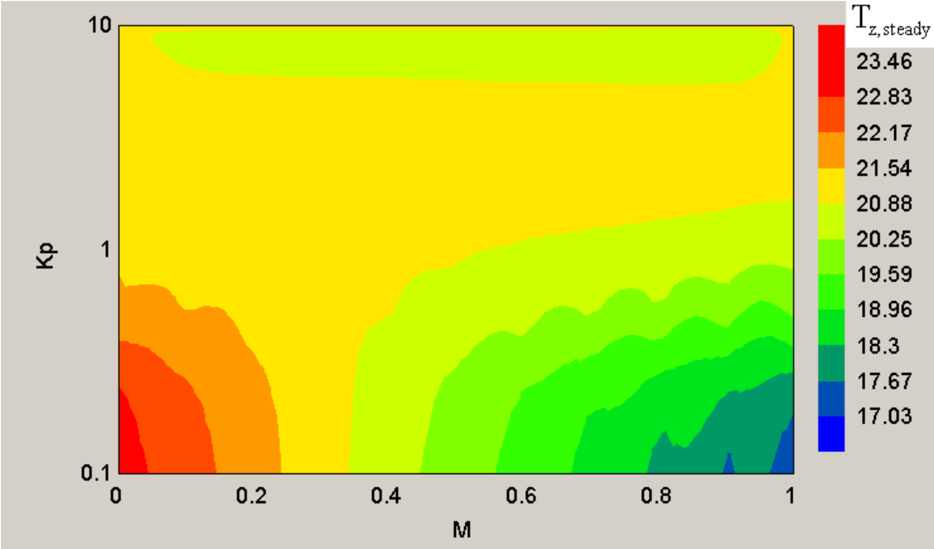

Figure 11 – Effect of gain and bias on the zone steady temperature for the proportional controller case considered.

In Figure 11 is presented the effect of bias and gain on the steady-state temperature. All the other parameters are kept the same as defined previously and in Table 1. Figure analysis shows that in the lower range of K_p (near 0.1) the effect of increasing bias is to decrease the steady-state temperature. Also in the lower range of K_p , as K_p increases, the range of temperature offsets with M is diminished. The bias value for zero offset can be seen located between 0.2 and 0.4. For the higher range of K_p (1-10), the steady temperature can be seen relatively insensitive to bias variation. From Mitchell and Braun (2003) a gain value of 0.5 can be used as the superior limit for K_p in proportional controllers for HVAC applications.

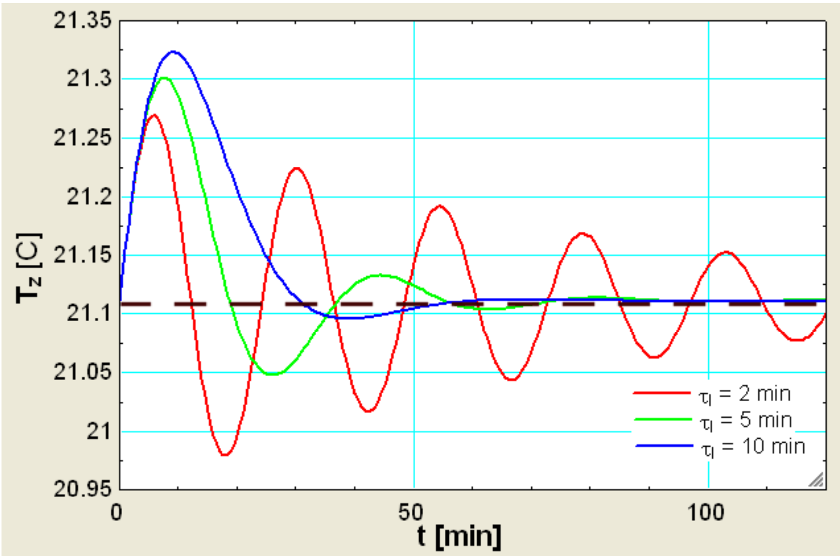

Figure 12 – Effect of integral time constant on the zone steady temperature for a proportional integral controller.

In Fig. 12 is presented the system response for different integral time constants for the proportional integral controller when the sensible load is doubled. Figure analysis shows that use of the proportional integral control eliminates the offset. The increase in the integral time constant results in faster stability with less oscilations. On the other hand the overshooting is greater. The inclusion of a derivative action in the Eq. (3) would be expected to speed the stability with reduced overshoot. However, in the practice of HVAC this is not usual as the system would be sensible to noise and a more elaborate tuning will be required (Mitchell and Braun, 2013).

4. CONCLUSION

The purpose of this work was to evidence some properties of the typical controllers in HVAC feedback control loop for a VAV box. Typical operating conditions of HVAC were used . The feedback control system with a integral proportional controler was the more adequate, which proper reduction of system instability to change in process parameters, and allowing to adjust the transient response and suppress the offset. The advantages in implementing the feedback system control in a VAV box surpasses the costs associated with more components and complexity.

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