

## A MULTILOOP CONTROL SCHEME FOR A DESICCATING ROTOR BASED DEHUMIDIFICATION SYSTEM

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**Abstract.** The fast world-wide industrial development of the last decades and the new standards for product quality control have established the need for new and more efficient environment control in industrial processes. Well known applications of this kind of system are: storage of pharmaceutical products, food, sweets and grains; manipulation of humidity sensitive products such as soluble and effervescent products, laminated glass; finishing of metal surfaces; fungus and bacteria prevention in laboratories and hospitals; and much more. The independent control of temperature and humidity of the environment air is required in modern industrial plants; they affect the final product quality in almost all industrial processes and because of that, they are in the concern of the production engineers. Classical solutions such as vapor compression-based cooling systems are not able to deal with the need of humidity reduction during the production and storage of humidity-sensitivity goods. In the same manner, conventional air conditioning and ventilation systems fail in situations in which the air input flow is at extreme humidity conditions. Modern solutions include desiccating dehumidification systems (DDS) that allow the independent control of temperature and humidity. This paper presents a multiloop control scheme for a desiccating dehumidification plant. It presents a nonlinear model obtained through physical laws. Also presents the control design and controller tuning based on a linear model of the plant. Finally, model validation and control performance assessment are performed through computer simulation.

**Keywords:** Desiccating Dehumidification Systems, Environment Control, Air Quality Control.

### NOMENCLATURE

$P$	=	Resistors Bench Nominal Power	$A$	=	Area
$\theta_{pi}$	=	Input Process Air Temperature	$V$	=	Volume Inside the Resistor Chamber
$\theta_{po}$	=	Output Process Air Temperature	$R_{th}$	=	Thermal Resistance
$\theta_e$	=	Environment Air Temperature	$C_{th}$	=	Thermal Capacitance
$T_{rotor}$	=	Rotor Time Constant	$H_r$	=	Heat Produced by the Resistors Bench
$T_{sensor}$	=	Humidity Sensor Time Constant	$H_i$	=	Enthalpy at the Process Input
$T_{th}$	=	Heating Time Constant	$H_o$	=	Enthalpy at the Process Output
$\tau_{pipe}$	=	Pipe Length Time Delay	$H_e$	=	Heat Loss to the Environment
$R/P$	=	Reactivating Air / Process Air Relation	$L$	=	Pipes Length
$Q_p$	=	Process Airflow Rate	$W_r$	=	Reactivating Air Humidity
$Q_r$	=	Reactivating Airflow Rate	$W_p$	=	Process Air Humidity
			$RH_{po}$	=	Output Process Air Relative Humidity

### 1. INTRODUCTION

The worldwide technological boom of the last decades made necessary the development of new techniques for the industrial environment air control. Final product quality must be kept changeless despite the manufacturing conditions. In this perspective, the independent control of humidity and temperature of the process air is vital to the modern production processes. Conventional vapor-compression-based cooling systems are not able to cope with the current humidity standards required by the production and storage of humidity sensitive products. Currently, the interior air quality (IAQ) rigid standards made problematical the use of conventional air conditioning systems, especially in extreme humidity conditions. Presently, the best technical solution for the control of humidity, temperature and flow rate of the process air is given by desiccant dehumidification systems (DDS), which are largely spread among the industrial processes. The purpose of industrial desiccant dehumidification systems (DDS's) is to control the humidity, temperature and flow rate of the air inside a target environment. In general, PID type controllers are utilized as control elements; however, their tuning usually presents challenging difficulties.

The work presented here is concern with the modeling and control design of a DDS, its main objective is to produce a tool for analysis, simulation and controller tuning. The DDS considered in this work was designed to keep the air relative humidity in the range of 10-40 %, the air temperature between 10 and 25°C and the air flow rate in the range

1200 and 4200 m<sup>3</sup>/hr. It initially presents a brief description of a desiccant dehumidification system and its control strategy, next it discusses the modeling procedure with the objective of control design and tuning and finally it presents simulation results of the closed loop system.

## 2. THE DESICCANT DEHUMIDIFICATION SYSTEM

The desiccant dehumidification system (DDS's) considered in this work is constituted by a filtering system, a first stage of the cooling system (first coil), a desiccating rotor, a second stage of the cooling system (second coil) and an air blower. Besides those, it also includes airflow control valves, pipelines and an automation and control system. Figure 1 presents the DDS.

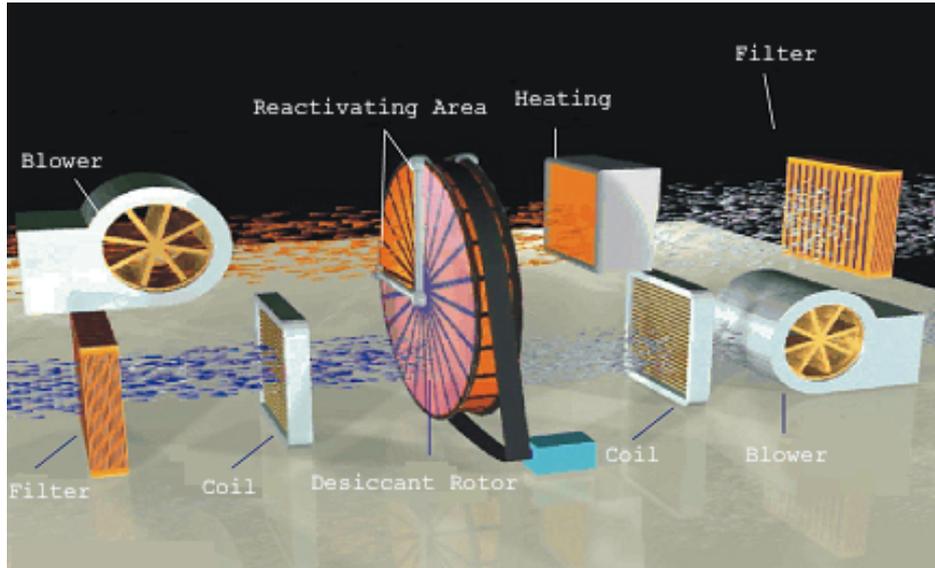


Figure 1. The Desiccant Dehumidification System.

The air passes through a filtering system that is constituted by three stages: a G0-type metal filter (washable), a G3-type flat filter and an F1-type pouch filter. Next, the filtered air goes through the first stage of the cooling system. Table 1 shows the thermal conditions at this stage.

Table 1. Thermal Conditions at the First Stage Coil (at full load).

	Inlet	Outlet
Water Temperature	8 °C	14 °C
Air Temperature	40 °C	10 °C
Air Humidity	50 %	100 %
Total Heat	73,230 kcal/h (30,482 + 42,748)	
Condensed Flow Rate	79.5 l/h	

From the cooling first stage the air is directed to the desiccating rotor that has the following characteristics: 965 mm diameter, 200 mm width, speed of 24 rph and surface relation (R/P) of 0.333. The desiccant rotor turns in a system with two airflows: the process airflow (previously cooled at 10°C) and the reactivating air flow (previously heated).

The humidity is transfer from the process air to the desiccant rotor. It is removed from there by the reactivating air in a region that is 25% of the rotor total area. The process air leaves the desiccating rotor at 20°C to 30°C (depending on the required humidity level). To restore its previous conditions the process air is directed through the second stage of the cooling system. Table 2 shows the thermal conditions at this stage.

Table 2. Thermal Conditions at the Second Stage Coil (at full load).

	Inlet	Outlet
Water Temperature	8 °C	14 °C
Air Temperature	30.7 °C	9.9 °C
Air Humidity	2.11 g/kg	2.11 g/kg
Total Heat	22209.2 kcal/h	

The treated air is finally leaded to the production area through a thermal-insulated rectangular-shape cross-section pipe (50 x 0.4 x 0.6 m). The controller is implemented in a Siemens 314 C 2DP PLC connected at a Profibus field bus. Figure 2 presents the desiccating cycle.

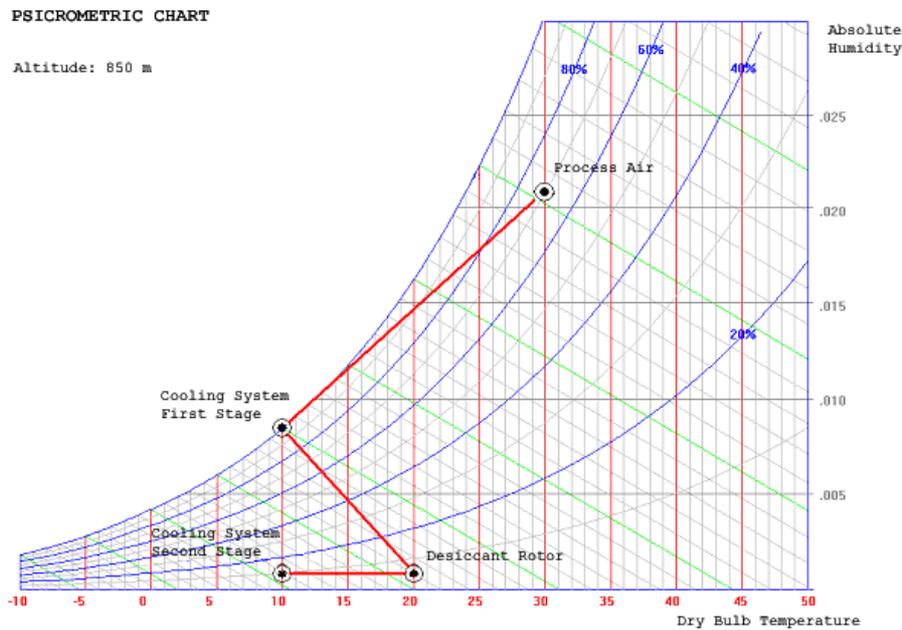


Figure 2. The Desiccating Cycle.

The desiccating rotor is made of silica gel, two opposite air flows pass through it. The process air goes trough the main part of the rotor and becomes dryer. The reactivating air passing by a smaller section of the rotor removes the water vapor. The humidity level of the process air can be controlled manipulating the temperature of the reactivating air. In general, the reactivation process starts at 60°C, the maximum temperature for safe reactivation is about 160°C. For non-controlled systems, the manufacture’s recommendation is 140oC. The reactivating air is heated up in the resistors bench that has a maximum power of 52 kW. The delivered power to the resistors bench is adjusted through a thyristor-based static converted.

The dehumidification processes is a function of the R/P relation, the temperature of the reactivating air, the temperature of the process air, the speed of the process through the rotor, the humidity of the process air, the reactivating air humidity, the speed of the reactivating air through the rotor, the mechanical efficiency of the desiccant rotor, etc. Figure 3 shows the R/P relation effects on the air temperature and on the air humidity at the dehumidifier outlet (Camargo, 2003).

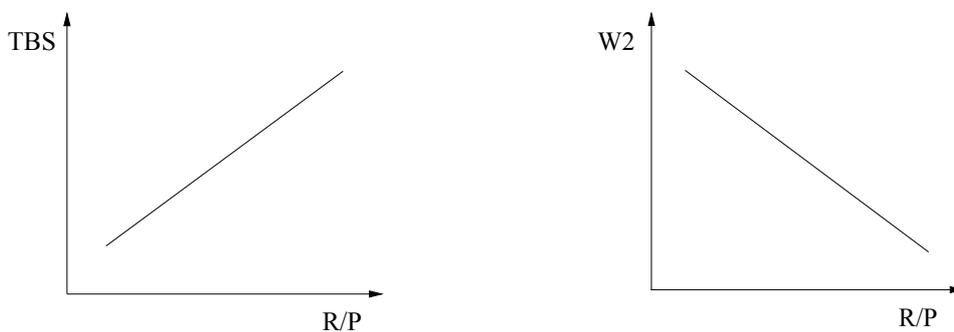


Figure 3. Reactivating Air Flow - Process Air Flow Relation Effects.

Figure 4 shows the effects of the reactivating air temperature on the air temperature and on the air humidity at the dehumidifier outlet (Camargo, 2003).

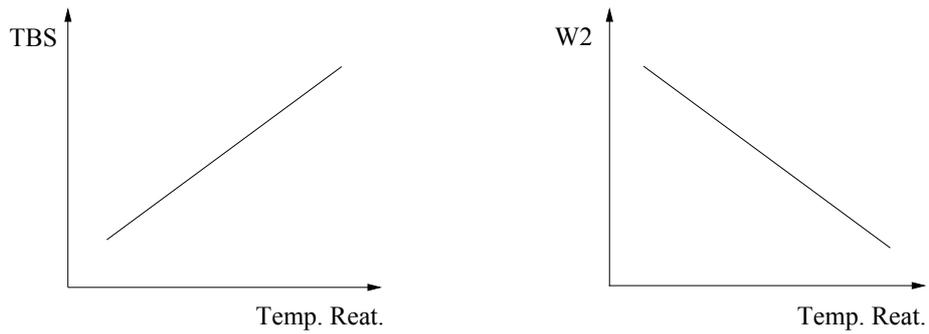


Figure 4. Reactivating Air Temperature Effects.

### 3. THE SYSTEM MODEL

Thermal systems are accurately described by distributed parameter models, unfortunately such models are not adequate for control design purposes. The heuristic modeling procedure adopted here had as its main objective to simplify the controller design and tuning. The modeling procedure was performed in two steps: in the first one, the model parameters, that define the system steady state, were found; in the second one, time constants and delays were added to the previous steady state model in order to include dynamics. A brief discussion of the subsystems models is presented next.

- **The Desiccating Rotor**

Desiccating rotors have a frontal area whose dimensions are designed based on the optimal flow rate and also a large adsorption surface whose size depends on the rotor width and on the porosity of its internal surface. Figure 5 presents a typical configuration of this type of dehumidifier based on rotating cylinders.

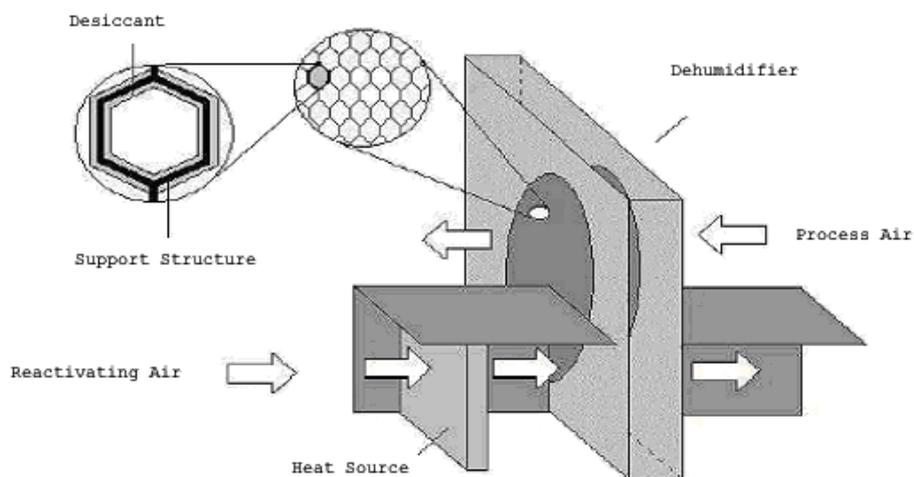


Figure 5. Dehumidifier Based on Rotating Cylinders (Camargo, 2003).

Figure 6 presents the air flows in a desiccant rotor. Inside the rotor, a thermodynamic interaction between the process air and the reactivating air takes place. Based on that, one can consider that a desiccating dehumidifier rotor works as a humid-heat pump, which means that the desiccating rotor is able to transfer latent heat (humidity) from the process air to the reactivating air (Camargo, 2003).

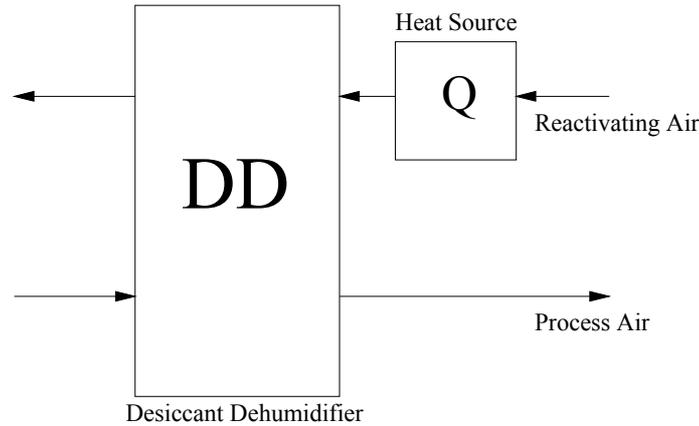


Figure 6. Process and Reactivating Air Flows.

Thus, a mathematical correlation can be established between the rotor input variables (reactivating air temperature and flow rate) and the output process variables (air humidity and flow rate) measured at the end of the pipe line distant 50 m away. Treating the desiccant rotor as a humidity exchanger, one has:

$$Q_r (W_{ro} - W_{ri}) = Q_p (W_{pi} - W_{po}) \quad (1)$$

The optimal operation point (manufacturer's data) gives a 1/3 relation between the reactivating air flow rate ( $Q_r$ ) and the process air flow rate ( $Q_p$ ). Based on manufacturer's technical data, the dehumidification rate can be approximated by

$$(W_{pi} - W_{po}) = 0.051 \theta_r + 1.8 \ln(Q_r) \Rightarrow W_{po} = W_{pi} - 0.051 \theta_r - 1.8 \ln(0.333 Q_p) \quad (2)$$

The desiccant rotor has a lag type time response. The associated time constant was experimentally determined and it can be approximated as

$$T_{rotor} = (60/5)RPM_{rotor} \quad (3)$$

- **The Relative Humidity Sensor**

The relative humidity is a function of the air temperature, so, the sensor model also includes the process air temperature. For a 10°C to 40°C temperature range, at nominal atmospheric pressure, the relative humidity sensor model can be written as

$$RH_{po} = \frac{W_{po}}{0.0004 \theta_p^2 - 0.004 \theta_p + 0.0905} \quad (4)$$

The humidity sensor is located at the pipeline end and feeds the controller with the relative humidity value ( $RH_{po}$ ). From manufacturer's technical data the associated time constant approximately is

$$T_{sensor} = 5 \quad (5)$$

- **The Pipes**

The distance between the desiccating rotor and the relative humidity sensor (50 m of pipes in this case) causes a time delay in the control loop that is a function of the pipe length and air velocity inside the pipe, such that

$$\tau_{pipe} = \frac{AL_{pipe}}{Q_p} \quad (6)$$

- **The Ventilation System**

The ventilation system, part of the dehumidification system, is not included in the control closed loop and contributes to it with a constant air flow with constant temperature and can be manually tuned.

- **The Resistor Bench**

The resistor bench model has a delivered power proportional to its input manipulated variable.

$$H_r = K_h [u(t)] \quad (7)$$

Its model includes a time constant that depends on the thermal resistance and capacitance of the volume of air inside the equipment:

$$T_{th} = R_{th}C_{th} \quad (8)$$

- **The System Full Model**

The system equations relating the relative humidity of the process air with the airflow rate and temperature of the reactivating air can be found from the heat and humidity exchanges at the rotor input, pipes and humidity sensor. The full model includes all subsystem models previously described. Figure 7 presents the system model in block-diagram form.

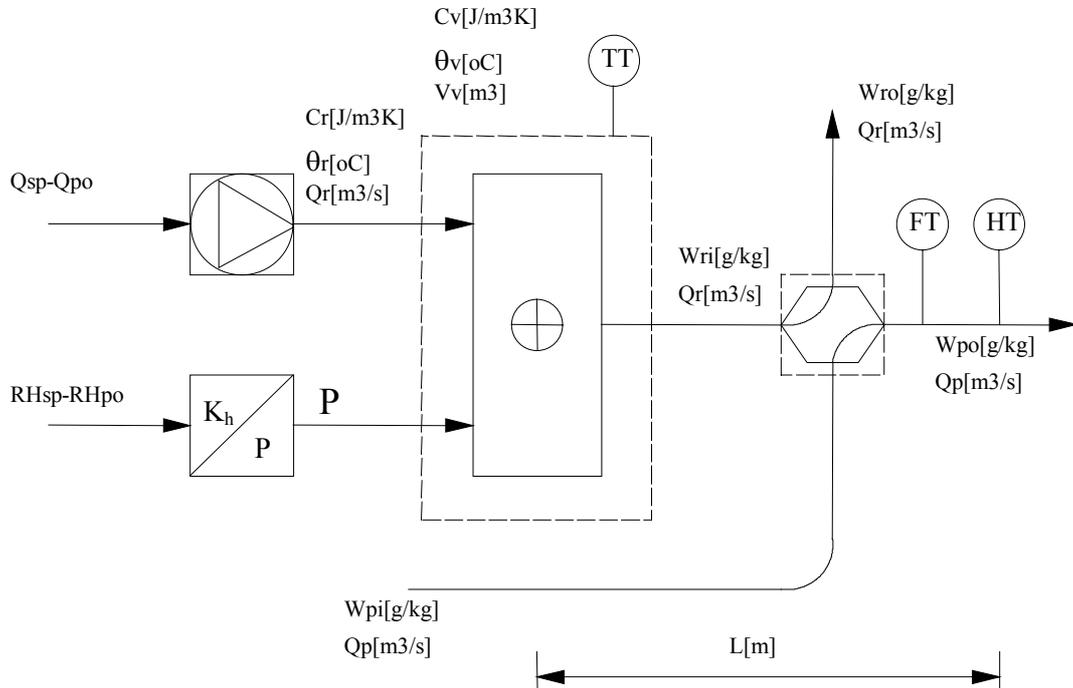


Figure 7. The Desiccant Dehumidification System.

In order to reach a dumped parameters model and for the sake of simplicity, some heuristic assumptions were made: Energy transfers to the air were considered linear and proportional to the manipulated variables. The heat and humidity storage at the pipes, metal structures, resistors and rotor were neglected as well as heat loss and humidity absorption through the walls and pipes. Heat flows were always considered as being at steady state.

The model was developed considering the following variables: Air temperatures at the process input ( $\theta_i$ ) and output ( $\theta_o$ ), air thermal capacitance ( $C_{th}$ ), air thermal resistance ( $R_{th}$ ), heat produced by the resistors ( $H_r$ ), heat loss to the environment ( $H_e$ ), enthalpy at the process input ( $H_i$ ) and output ( $H_o$ )

From the energy conservation principle

$$H_o = H_r + H_i - H_e \quad (9)$$

$H_e$  is relatively small compared with  $H_r$  and  $H_i$ , so one has

$$C_{th} \frac{d\theta_o}{dt} = H_i + H_r \quad (10)$$

in this case,  $H_i$  can be considered as constant at the operation point, then

$$H_i = \frac{C_{th} Q_r}{V} \theta_e \quad (11)$$

Considering the heating system power as being proportional to the humidity error, one has

$$H_r = K_h [u(t)] = K_h [K_p e(t)] ; \quad e(t) = [RH_{sp}(t) - RH_{po}(t)] \quad (12)$$

then,

$$\frac{d\theta_o}{dt} = \frac{1}{C_{th}} \left[ \frac{C_{th} Q_r}{V} \theta_e + K_h K_p e(t) \right] \quad (13)$$

On the other hand, by including a PID controller in the loop, the heating power becomes

$$\frac{d\theta_o}{dt} = \frac{1}{C_{th}} \left[ \frac{C_{th} Q_r}{V} \theta_e + K_h K_p \left( e(t) + K_i \int e(t) dt + K_d \frac{d e(t)}{dt} \right) \right] \quad (14)$$

From Eq. (2) and considering the process air flow rate ( $Q_p$ ) and process air humidity ( $W_{pi}$ ) as being constant parameters at the operating point, the absolute air humidity at the rotor output ( $W_{po}$ ) can be written as a linear function of the reactivating air temperature,

$$W_{po} = K_{rotor} \theta_r ; \quad K_{rotor} < 0 \quad (15)$$

Equations (3) and (15) lead to the transfer function for the desiccating rotor:

$$G_{rotor}(s) = \frac{W_{po}(s)}{\theta_r(s)} = \frac{K_{rotor}}{T_{rotor} s + 1} \quad (16)$$

in the neighborhood of a given operating point defined by  $\theta_p = 10^\circ\text{C}$ , Eq. (4) gives,

$$RH_{po} = K_{sensor} W_{po} \quad (17)$$

from Eqs. (5), (6) and (17), the transfer function for the humidity sensor can be written as

$$G_{sensor}(s) = \frac{RH_{po}(s)}{W_{po}(s)} = \frac{K_{sensor} e^{-\tau_{pipe} s}}{T_{sensor} s + 1} \quad (18)$$

From Eqs. (16) and (18) ones has

$$G(s) = G_{sensor}(s) G_{rotor}(s) = \left( \frac{RH_{po}(s)}{W_{po}(s)} \right) \left( \frac{W_{po}(s)}{\theta_r(s)} \right) = \left( \frac{K_{sensor} e^{-\tau_{pipe} s}}{T_{sensor} s + 1} \right) \left( \frac{K_{rotor}}{T_{rotor} s + 1} \right) \quad (19)$$

then

$$G(s) = \frac{RH_{po}(s)}{\theta_r(s)} = \left( \frac{K_{sensor}}{T_{sensor} s + 1} \right) \left( \frac{K_{rotor}}{T_{rotor} s + 1} \right) e^{-\tau_{pipe} s} \quad (20)$$

finally, the transfer function for the dehumidification system is given by:

$$G(s) = \frac{RH_{po}(s)}{\theta_r(s)} = \frac{K}{(T_{sensor} s + 1)(T_{rotor} s + 1)} e^{-\tau_{pipe} s} ; \quad K = K_{sensor} K_{rotor} \quad (21)$$

#### 4. THE CONTROL SYSTEM

The main objective of the desiccant dehumidifier system is to control the relative humidity of the air in a target environment. In the DDS case, the manipulated variables are: the flow rate of the reactivating air, the temperature of the reactivating air (defined by the heating power produced by the resistors bench) and/or the temperature of the process air. The controlled variables are: the temperature of the reactivating air, flow rate of the process air and/or the relative humidity of the process air. In this case, a heuristic tuning procedure was carried out based on the process time constants.

Figure 8 presents the block diagram of the control system with the inclusion of a block that transform temperature in absolute humidity (physically implemented by the desiccant rotor), a block that transform absolute humidity in relative humidity (physically implemented by the humidity sensor) and a block that incorporate all time delays of the system.

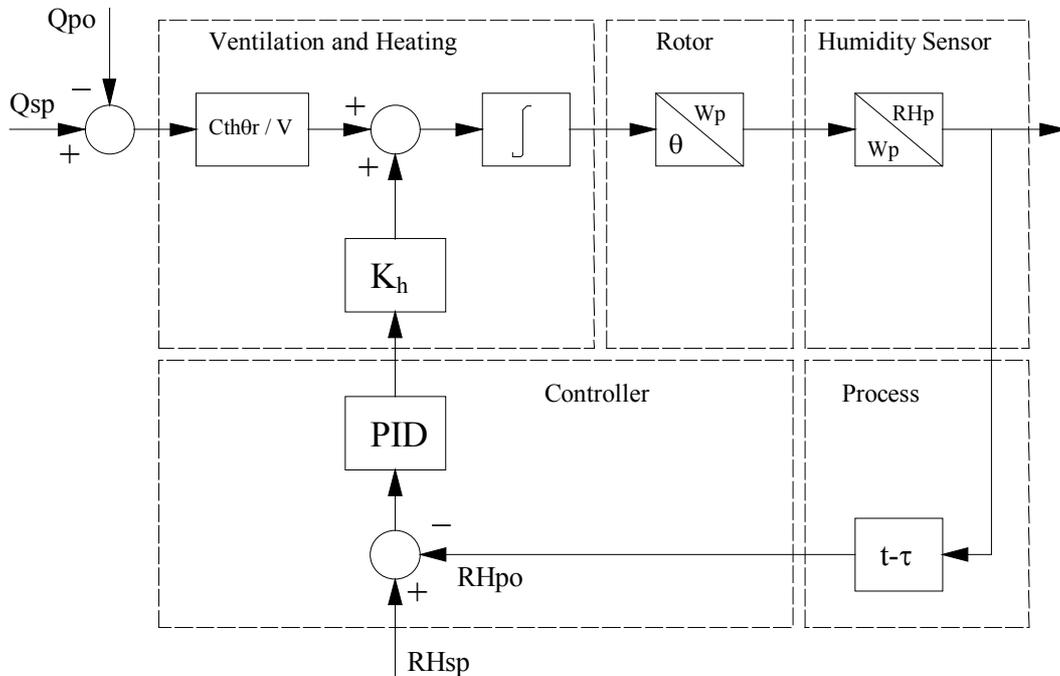


Figure 8. Humidity Control System Block Diagram.

The controllers were tuned starting with the process air flow rate, followed by the process air temperature at the rotor input, then by the process air temperature and finally by the relative humidity of the process air. Figure 9 presents the system devices block diagram.

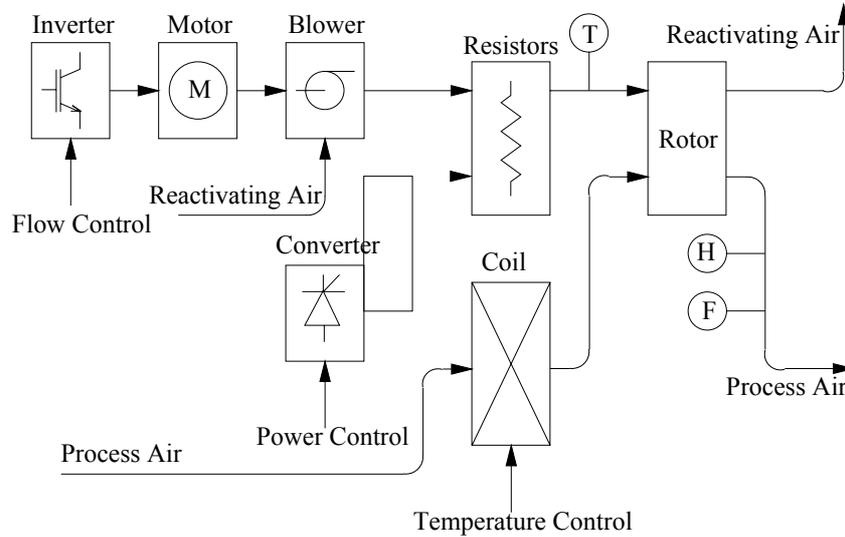


Figure 9. Control Devices for the Dehumidification System.

A frequency inverter acts over the motor of the air blower controlling the reactivating airflow rate. A flow rate meter is used to produce the feedback signal (4-20mA). The controlled flow rate range is 1200-4200 m<sup>3</sup>/h with steps of 50 m<sup>3</sup>/h. To keep the process air temperature at 10°C, a PID-type controller controls the temperature at the rotor inlet using a PT-100 as temperature sensor (that produces a feedback signal of 4-20mA). This controller acts over a 3-way valve on the cold-water circuit and controls the cold-water flow rate. The relative humidity control is achieved by a PID-type controller that acts, through a static converter, on the delivered power to the resistors bench.

## 5. EXPERIMENTAL RESULTS

The model was implemented in Simulink<sup>(R)</sup> - Matlab<sup>(R)</sup>. The model validation was performed using the values presented in Tab. 3 (model validation results have been omitted here to save printing space).

Table 3. Simulation Parameters.

Resistors Bench Nominal Power (Max)	$P = 52 \text{ kW}$
Input Process Air Temperature	$\theta_{pi} = 10 \text{ }^\circ\text{C}$
Output Process Air Temperature	$\theta_{po} = 10 \text{ to } 25 \text{ }^\circ\text{C}$
Output Process Air Humidity	$RH_{po} = 10 \text{ to } 40 \%$
Input Reactivating Air Absolute Humidity	$W_{ri} = 15 \text{ g}_{\text{agua}}/\text{kg}_{\text{ar}}$
Process Airflow Rate	$Q_p = 1200 \text{ to } 4200 \text{ m}^3/\text{h}$
Rotor Time Constant	$T_{rotor} = 30 \text{ s}$
Pipe Length Time Delay	$\tau_{pipe} = 20 \text{ s}$
Humidity Sensor Time Constant	$T_{sensor} = 5 \text{ s}$
Heating Time Constant	$T_{th} = 10 \text{ s}$
Reactivating Air / Process Air Relation	$R/P = 0,333$
Environment Air Temperature	$\theta_e = 10 \text{ to } 40 \text{ }^\circ\text{C}$

The proposed model was then used to tune up a PID controller for the process values presented in Tab. 4. The PID achieved values were  $P_b = 200$ ,  $I_r = 0.04$  e  $T_d = 5$ .

Table 4. PID Control Tuning Parameters.

Output Process Air Relative Humidity	$RH_{po} = 20\%$
Output Process Air Temperature	$\theta_{po} = 20 \text{ }^\circ\text{C}$
Process Airflow Rate	$Q_p = 3000 \text{ m}^3/\text{h}$
Environment Air Temperature	$\theta_{env} = 20 \text{ }^\circ\text{C}$

Figure 10 presents the simulation results for the closed loop systems. It can be observed that the system settling time is around 300 seconds and that the reactivating air temperature remains inside the safety range (less than 160 °C).

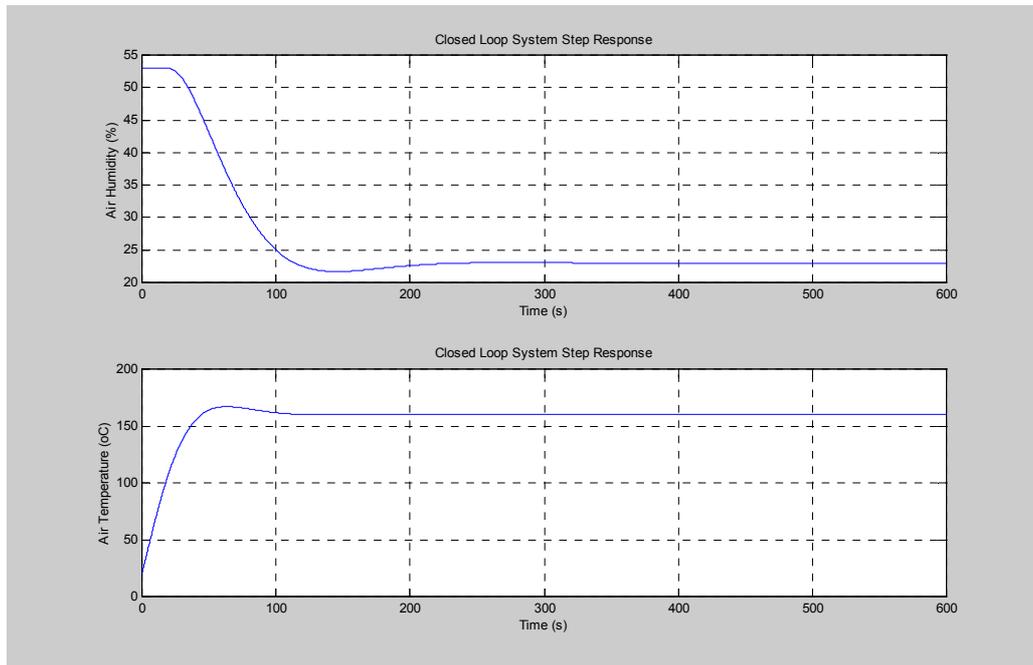


Figure 10. Closed Loop System Step Response.

## 6. FINAL COMMENTS

This work presented a dynamic model for analysis, simulation and design of PID controllers for dehumidification systems based on desiccating rotors. Two alternatives were considered, the first one uses the reactivating air temperature as the manipulated variable, and the second one uses the reactivating airflow rate. With the former delivering better results, a quasi-linear system response and wider humidity controlled range. The model was implemented in Simulink<sup>(R)</sup> validated and tested for several values of humidity, temperature and flow-rate inside the manufacturer specifications. Finally, the system PID control loops were tuned using a sequential tuned technique based on pole dominance criterion.

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