Fast prototyping and indirect adaptive GPC temperature control of a class of passive HVAC

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Abstract: - This study focuses firstly on the numerical investigation of an air conditioning unit prototype developed to guarantee a microclimate with controlled temperature and relative humidity set points for crop growth chambers. Numerical techniques based on Computational Fluid Dynamics (CFD) were implemented to analyze the flow characteristics of the mixed air produced by the device. Simulations disclose a non-linear relationship between the air flow rate and the aperture opening, and also show that the mixing zone was not perfect but could be improved by the addition of baffles. The promising results obtained from CFD were used to improve the automation of the device for potential application in closed greenhouses. Secondly, the indirect adaptive generalized predictive control strategy (IAGPC) with a decentralized architecture was used to control the temperature of the air conditioning unit. As the process involves time-varying, the use of a recursive estimation approach with a fixed forget factor was adopted to estimate in real time the system parameters, and to adapt simultaneously the GPC controller parameters. In order to achieve a local and global efficient performance, the proposed decentralized IAGPC architecture was applied to two principal subsystems, which compose the global unit. A significant real time experimental improvement in the system performance is observed on temperature control for a wide range of operating points.

Key-Words: numerical simulation, air flow distribution, time varying parameters, decentralized control, adaptive control.

1 Introduction

The energy consumption by the heating, by ventilation and by air conditioning equipment in the industrial buildings, constitutes 50% of the world energy consumption [1]. Among these buildings, one distinguishes horticultural greenhouses, which present an important branch of the agriculture sector [2], [3]. The optimal management of the greenhouse microclimate (temperature, moisture) is a dominating factor, on the one hand to deal with the market quantitative and qualitative requirements [4], [5] and on the other hand to ensure a possible better economic profitability for the farmers.

In general, air conditioning units used in crop growth chambers are often made up of elements of heating and cooling systems with a compression cycle [6], [7]. In addition to the energy cost and the high expenses of maintenance of this type of systems, they present also an ecological problem because of the pollutant emissions due to the used refrigerating gases. For those reasons, we have investigated an alternative system, which is passive and does not use the more typical compression device or absorption-refrigeration cycle. The unit can also be helpful in investigation of relative humidity changes and the morphological responses of the plant (growth stoppage, floral transformation, delayed blossom-time, dormancy, etc...).

This paper proceeds as follows. The first part presents the system requirements. The second part focuses the numerical studies conducted based on partial differential equations models with the aim to better understand the mixing process and to forecast the device behaviour for a set of operating conditions. The last part proposes the synthesis of indirect adaptive generalized predictive control law based on differential equations models, with a decentralized architecture in order to manage the air conditioning temperature intended to produce a specific climate in growth chambers.

2 Problem Formulation

The micro-climate needed must be produced by a

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passive air-conditioning system that is without a freezing unit and compressor, or refrigeration cycle, and without pollutant emissions [8]. The specificity of this system is to produce a variable microclimate with variable temperatures and relative humidity set points. Since temperature and relative humidity are highly coupled, one way to achieve these objectives is to delink the control of the temperature from the relative humidity control.
The air-conditioning cycle is presented graphically below

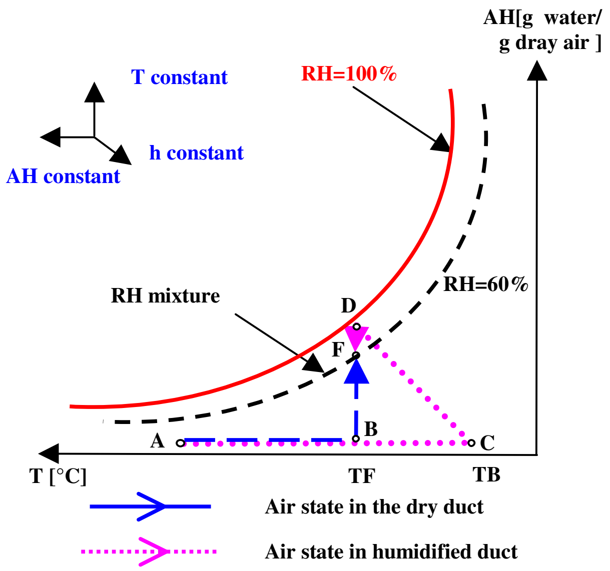

Fig. 1 : Thermodynamic cycle of the unit

Figure 1 shows the different thermodynamic phases of the air-conditioning cycle and the region corresponding to our zone of interest. The system depends on the mixing two air flows, each with a different humidity level. The air intake can be from inside greenhouse (point B) or outside greenhouse (point A). Regardless of the source of the air supply, the characteristics of the air are clearly defined.
The characteristics of the air at point F are also known because the final temperature T_F has to track the temperature of the second growth chamber, and the moisture RH_F is prescribed by the user. As the air heating operates at a constant absolute humidity, point B can be easily found by knowing the value T_F.
The computation of the characteristics of the air at point C is more complex. These characteristics can be deduced from point D, at which the temperature equals T_F. In D, air must be practically saturated. As cooling humidification (from C to D) operates at constant enthalpy, point C can be calculated by knowing the characteristics of points D and A.

Based on the enthalpy values of points A, B and C, the energy needed for heating can be computed. The air flow rate required to obtain the relative humidity set point is computed using the relative evolution of the line 'D-B'. Considering the values of q_i , the final expressions of the absolute humidity (absolute moisture content) and of the temperature are given by:

$$AH_F = \frac{q_1 . AH_B + q_2 AH_D}{q_1 + q_2} \qquad (1)$$

$$T_F = \frac{q_1(\alpha + \beta AH_B)T_B + q_2(\alpha + \beta AH_D)T_D}{q_1(\alpha + \beta AH_B) + q_2(\alpha + \beta AH_D)} \qquad (2)$$

with $\alpha = 0.24$, $\beta = 0.46$ and q_i is the air-flow mass proportional to the aperture position. Knowing both AH_F and T_F gives an unique value of RH_F [9].

3 Numerical simulation and air-conditioning design

3.1 Conditioning unit distributed model

3.1.1 Turbulence model

CFD simulations were carried out with the commercially available package Fluent 6.1. The method is based on the solution of the 2D-3D convection-diffusion equation which, for incompressible fluids and under steady state conditions, may be written as:

$$\frac{\partial(U\phi)}{\partial x} + \frac{\partial(V\phi)}{\partial y} + \frac{\partial(W\phi)}{\partial z} = \Gamma\Delta\phi + S_\phi \qquad (3)$$

where ϕ represents the concentration of the non-dimensional transported quantity, namely momentum, mass and energy; U, V and W are the components of the velocity vector; Γ is the diffusion coefficient and S_ϕ is the source term.
The equations are Reynolds-averaged Navier-Stokes and the solution variables in the instantaneous Navier-Stokes equation are decomposed into mean and fluctuating components. The classical turbulence k-ε closure model is chosen because of its good ability to describe fully developed turbulent flows. The above mentioned equations are discretized on a triangular-cell grid. The grid is refined close to orifices where strong velocity gradients may occur.

3.1.2 Porous media.

The corrugated pads create a sink of momentum due to friction forces (drag forces) of airflow through the pads. The pads are considered as macro-porous media in which the Darcy-

Forchheimer model is assumed valid [10], [11]:

$$-\frac{\Delta p}{L}=\left(\frac{\mu}{\alpha}+\frac{C_2}{2}\|v\|\right)v_i \qquad (4)$$

where Δp is the pressure drop, L is the thickness of the pads, α is a coefficient independent of the nature of the fluid but depending on the geometry of the medium. It has the dimension of a squared length and is called the specific permeability of the medium. C_2 is a dimensionless form-drag constant dependent on the nature of the porous medium and is called inertial factor, μ is the air dynamic viscosity, v_i are the velocity components,

3.1.3 Virtual tracer gas technique.

The species transport model without reaction was activated to solve the mixing problem. The governing equation was written as:

$$\frac{\partial}{\partial t}(\rho C)+\nabla(\rho\vec{v}C)=-\nabla\vec{J}+S \qquad (5)$$

with C, the concentration of the transported quantity, S, the mass source term of the considered species, $\vec{J}$, the diffusion flux. S was set to zero in the air mixing computation, and equal to the water source mass in the pads during humidification investigation.

3.1.4 Boundary Conditions

The boundary conditions considered at the inlet circular section, of diameter 0,02m², were homogeneous velocities of 1,5 m s^{-1}, 2 m s^{-1}, 4 m s^{-1} and 8 m s^{-1} respectively, which were assumed to be normal to the section. The outlet section varied from 0,003 to 0,015 m², and the boundary conditions were outflow with zero output gradient. A classical wall function was imposed at the solid boundaries (wall). These were later considered as adiabatic. The turbulence parameters were determined specifying the turbulence length scale and the hydraulic diameter.

3.2 Simulation result and process design

Fig. 2 shows the contours of the velocity magnitude when pads are not included. One ca observes the preference way of the air flow and its heterogeneous distribution. Next simulation demonstrates that the use of pads contributes to a reduction of the vortex and an improvement in homogeneity of the air distribution in the duct. The influence of the pads is significant, the main advantage being the accurate saturation of air leaving the pads.

Figure 3 shows the behaviour of the mixing zone in the original device. There is no migration of species in the mixing zone. The molecular diffusion seems to be much smaller than the turbulent diffusion. When baffles were added downstream the unit, the mixing was much better as shown in Figure 4. More results can be found in [12].

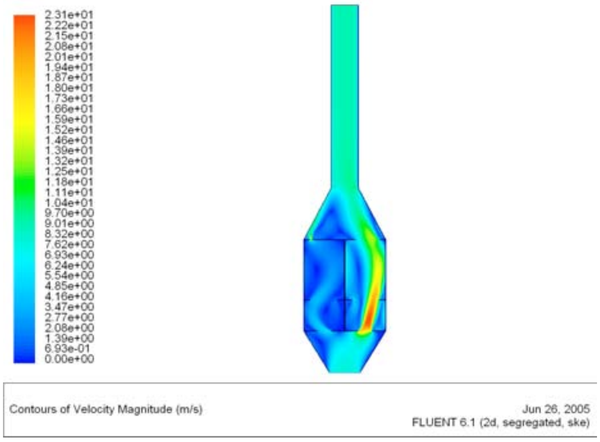

Fig. 2. Contours of velocity magnitude m s^{-1} (inlet velocity 8 m s^{-1}, opening 10%-90%)

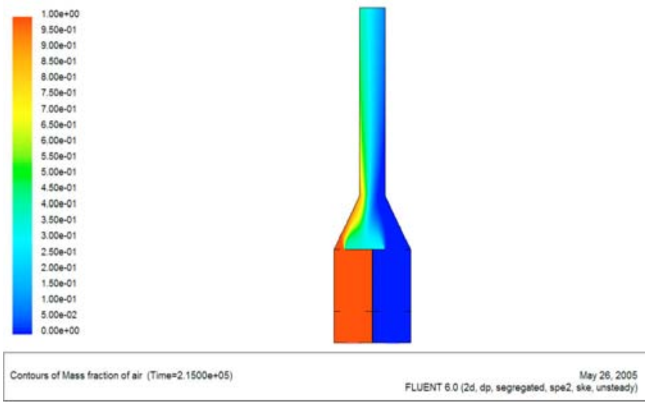

Fig. 3. Contours of mass fraction of air, inlet velocity 4 m s^{-1}, opening 20%-80%.

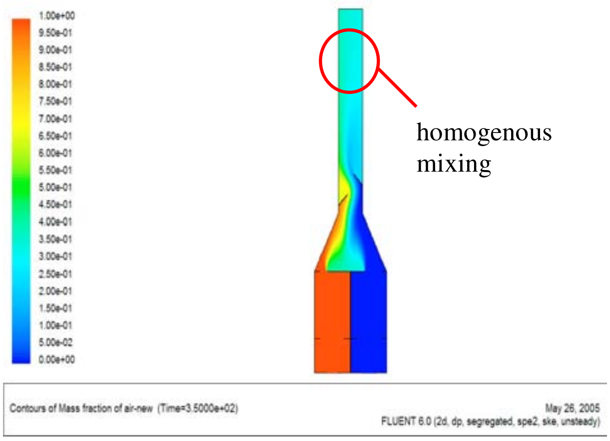

Fig. 4. Device with baffles : Contours of mass fraction of air, inlet velocity 4 m s^{-1}, opening 20%-80%.

The final form of the air conditioning device is presented on figure 5.

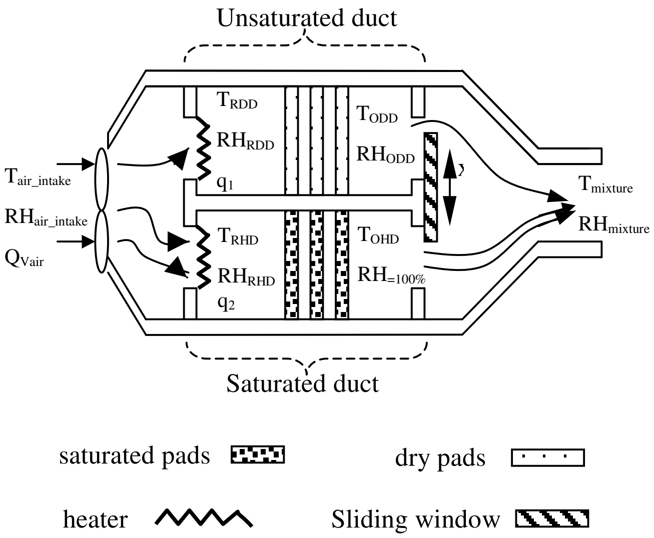

Fig. 5 Cross section view of the air-conditioning unit

The unit is composed of two flows: a non-saturated flow (or dry duct) and a saturated flow (or humidified duct) [12]. As shown in "Figure 5", in the saturated air flow, fresh air is saturated in humidity after being heated by a coil resistor. Saturation operates at constant [13]. The saturation unit consists of a closed system, including a pump, a water tank and cross-corrugated cellulosic pads of the type used in cooling. The suction pump carries water from the tank to the top of the pads. Once a steady state of saturation is reached, the pads contain a constant mass of water with a given water output and a given temperature. In the unsaturated air flow, fresh air is only heated by another resistor coil. Dry pads are included to provide pressure drop balance. The low speed of the air and of the water through the pads reduces the difference of pressure drop between the two streams.

The proportional mixing of the two air flows is carried out by an aperture operated by a DC motor. Assuming that the two air flows are well mixed, a local climate can be easily produced in the growth chamber. A complete physical model of the device is presented in [14]. As the system is nonlinear and time varying parameters, an adaptive predictive approach was used to control the temperature.

4 Indirect Adaptive GPC temperature control

4.1- Indirect Adaptive GPC theory

Considering the complexity of the air conditioning model, the decentralized control architecture was selected to carry out the thermodynamic objective described by the psychometric diagram (Figure 1).

The aim of the thermodynamic strategy is to guarantee the same temperature set point for each duct. Since the steady state temperatures in the dry and humidified ducts are obtained, the set point temperature will be guaranteed for an air mixture as shown in fig. 5. In order to validate the feasibility of this thermodynamic strategy by the air conditioning unit, we are first focused on the control of different temperatures of each sub-system in spite of air intake variation and variable operating conditions Figure 6.

The predictive control algorithms based on generalized predictive control or even long range predictive control strategies have proved to be efficient, flexible and successful for industrial applications [15], [16], [17], [18], [19], [20], [21]. This strategy is associated with the recursive estimation algorithm in order to get better performance in both tracking and regulation problems.

The dynamic of conditioning unit can be described around an operating point by the following CARIMA equation :

$$A\left(q^{-1}\right)\Delta y(k) = B\left(q^{-1}\right)\Delta u(k-d) + C\left(q^{-1}\right)\varepsilon(k) \quad (6)$$

$y(k)$ is the system output, $u(k)$ the system input, $\varepsilon(k)$ the uncorrelated random sequence, $\Delta\left(q^{-1}\right)=1-q^{-1}$ the difference operator, $A\left(q^{-1}\right), B\left(q^{-1}\right), C\left(q^{-1}\right)$ are polynomials with n_a, n_b and n_c degree respectively.

The predictive output vector can be formulated as

$$\hat{y}(k+j) = G_j\left(q^{-1}\right)\Delta u(k+j-1) + \ell_j \quad (7)$$

where: $\ell_j = F_j\left(q^{-1}\right)y(k) + H_j\left(q^{-1}\right)\Delta u(k-1)$

Where: F_j, E_j, G_j, H_j are obtained by a recursive calculation of Diophantine equations proposed by [22], [23].

The controller design is based on the minimization of the following cost function :

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$$J(k)=\sum_{j=N_1}^{N_2}\left(w(t+j)-\hat{y}(k+j)\right)^2+\lambda\sum_{j=1}^{N_u}(\Delta u(k+j-1))^2 \quad (8)$$

where: $\Delta u_j(k+j)=0$, for $j \geq N_u$.

$w(k+j)$ are the set points values at time $k+j$, $\hat{y}(k+j)$ the output prediction at time $k+j$, N_1 the minimum prediction horizon, N_2 the maximum prediction horizon, N_u the control horizon, λ the control-weighting factor.

The control law is therefore given by

$$\Delta U_{opt}=\left[G^T G+\lambda\, I\right]^{-1} G^T (W-L) \qquad (9)$$

With G is a $(N_2-N_1+1)\times N_u$ matrix. Only the first control value is finally applied to the system according to the receding horizon strategy:

$$u_{opt}(k)=u_{opt}(k-1)+\bar{G}(W-L) \qquad (10)$$

where, $\bar{G}$ is the first line of matrix $\left[G^T G+\lambda\, I\right]^{-1} G^T$.

The adaptive controller is obtained by simply invoking the certainty equivalence principle, which consists of replacing the process model parameters by their estimates when deriving the control law. To estimate the parameters of the model given by (6), and RLS parameter adaptation algorithm with fixed forgetting factors chosen to provide a better adaptation alterness was used [24].

4.2 Application to the process

The dry duct temperature discrete model:

$$\frac{T_{ODD}(k)}{U_{DD}(k)}=\frac{q^{-1}(b_{11}(k)+b_{12}(k)q^{-1})}{1+a_{11}(k)q^{-1}+a_{12}(k)q^{-2}+a_{13}(k)q^{-3}} \qquad (11)$$

The humid duct temperature discrete model:

$$\frac{T_{OHD}(k)}{U_{HD}(k)}=\frac{q^{-1}(b_{21}(k)+b_{22}(k)q^{-1})}{1+a_{21}(k)q^{-1}+a_{22}(k)q^{-2}+a_{23}(k)q^{-3}} \qquad (12)$$

The recursive identification and GPC code developed with Matlab® software were connected to the industrial automation via a local area network managed by interface developed with Delphi® software. A set of electronic units was used to apply heating voltage on the resistors or to control the DC motor and thus, the window opening rate. Measurements were carried out using Pt100 sensors for temperature and encoder sensors for window position. A sampling interval of Te=30 seconds was chosen to satisfy the predominant time constant, and data acquisition time about twelve hours. The operating point (aperture opening) values interval is $x \in [0\%,100\%]$. More results can be found in [25].

In general, good control performance is shown, by the IAGPC for different set points values. The temperature ducts are closed to the set points Figure 6, Figure 8. The figures show in general, an efficient disturbance rejection. These disturbances caused by the intake air temperature are eliminated by the integral action existing in the CARIMA basic model.

The control strategy robustness is also observed, through some temperature overshoots rejection. This type of disturbance is caused by the aperture commutation (operating point system variations) which affects in reality the air rate flow variation. At 700th sampling time in Figure 8, the overshoots presented by the air temperature response of the humid duct, result from the abruptly aperture opening commutation, which introduces a parametric error estimation and by consequence instantaneous closed loop instability between the 800th and the 900th sampling time. These can be explained by the non-persistence of the control signal in steady state, causing the cross-correlation of the covariance matrix vectors, which leads the estimator divergence.

In the Figure 10, the air temperature fluctuations do not appear between the 800 th and the 900 th sample time, such as in Figure 8, because during this time window, the humid duct was almost closed ($10\% < x\% < 18\%$), and thus, its contribution to the air mixing was reduced.

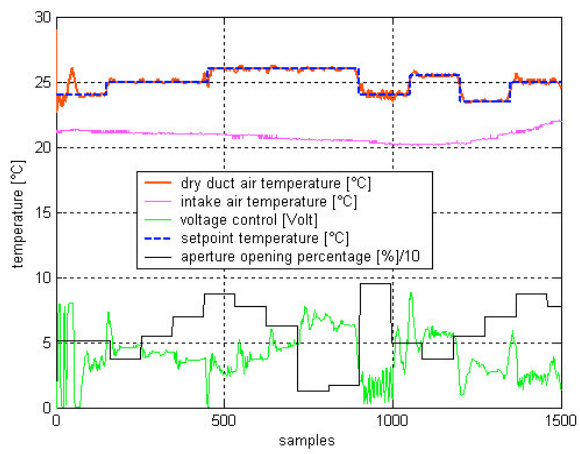

Fig. 6. IAGPC of the air dry duct temperature

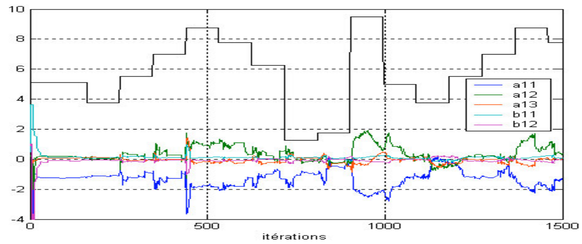

Fig.7. Dry duct model estimated parameters

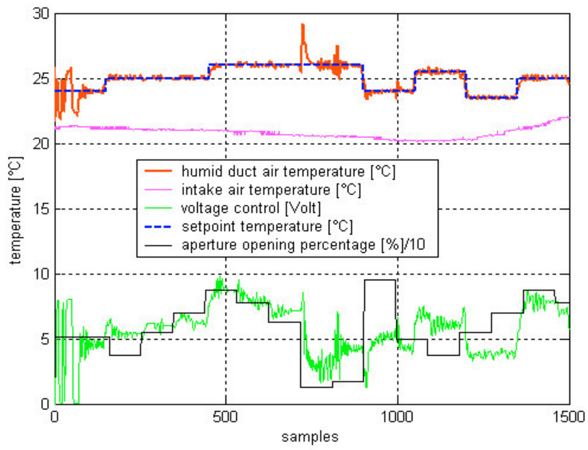

Fig.8. IAGPC of the air humid duct temperature

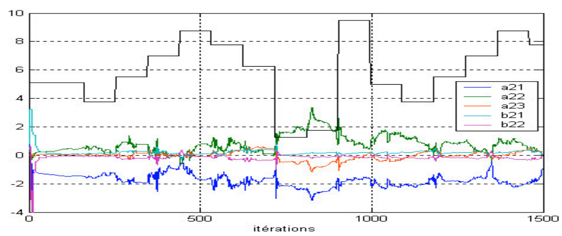

Fig.9. Humid duct model estimated parameters

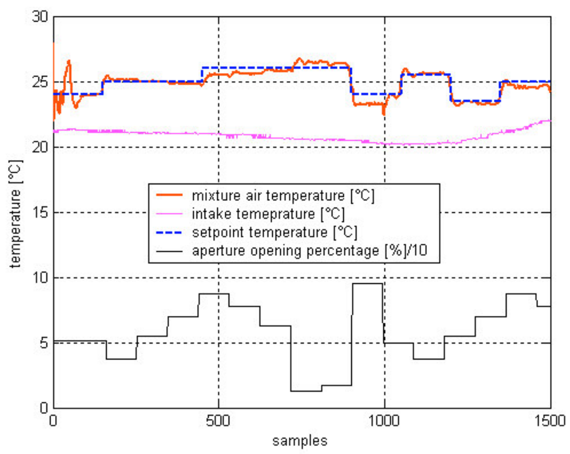

Fig.10. IAGPC of the air mixture duct temperature

To sum up, the air mixture temperature set points are guaranteed indirectly as consequence to the accuracy of the two temperatures at the upstream ducts "Figure 9", with an accepted accuracy, showing the feasibility of the proposed humid air thermodynamic strategy.

5 Conclusion

A CFD software was developed to investigate airflow velocity, pressure and temperature fields, through an air-conditioning unit. Different configurations were tested and compared to experimental values. The results obtained describe and validate the nonlinear relationship between the airflow rate and the aperture opening, and the independence of this relationship to air inlet velocity values. Simulations also show that adding baffles downstream the unit enhanced the air mixing process. These results provide significant information on sensor locations in order to assess the real behaviour of the mixing. Ongoing studies are being conducted in order to understand the transient behaviour during set point modification and to improve the shape of the mixing zone using the shape optimization theory.

Concerning the control aspect, the proposed IAGPC provides, an accepted robustness in spite of the parametric variation and the operating point changes. The relative humidity and the temperature control improvement will be brought by imposing the constraints on both outputs and input control. Taking into account a supervision stage on the local controllers is in study, to enhance the control performance.

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