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Modelling and control of a free cooling system for Data Centers

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Abstract

Data centers are facilities hosting a large number of servers dedicated to data storage and management. In recent years, their power consumption has increased significantly due to the power density of the IT equipment. In particular, cooling represents approximately one third of the total electricity consumption, therefore efficiently cooling data centers has become a challenging problem and it represents an opportunity to reduce both IT energy costs and emissions environmental impact. The efficiency of computers room air conditioning (CRAC) systems can be increased using both advanced control techniques and new free cooling technologies, such as the indirect adiabatic cooling (IAC), that is the humidification of air under adiabatic conditions. Water sprinkled by spray nozzles humidifies and cools down the air taken from the outside, which then cools down the computers room air by means of a crossflow heat exchanger. In this way, the process air temperature is economically reduced and the cooling process is effective even when the outside temperature is warmer than that desired in the computers room. Beside the traditional approach, that improves energy efficiency of CRAC systems through advanced hardware design, nowadays advanced control systems offer the opportunity to improve both efficiency and performance by mostly acting on software components. In particular, a model-based paradigm can result very useful in the design of the controller. This approach involves three main steps: plant modelling, controller design, and simulations. In this paper, First-Principle Data-Driven (FPDD) techniques have been considered in the modelling phase, in order to obtain a model as simple as possible but accurate enough. All the main components of the plant, such as fans, spray nozzles, heat exchanger, and the computers room have been taken into account and they have been calibrated exploiting real data. The dynamics of the computers room variables (e.g. temperature) are slower than those of the components of the cooling system, due to higher thermal inertias of the computers room. Therefore, fans, heat exchanger, and spray nozzles are described by static models, whereas the computers room is described by a LTI dynamic model. Once obtained a model of the plant, a simulation environment based on Matlab/Simulink is designed accordingly. The developed control system is hierarchical: a supervisor determines the best combination of CRAC water and process air flows which minimizes the total power consumption, while satisfying the cooling demand. This system energy management problem is formulated as a non-linear optimization problem,

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subject to internal air condition requirements and system operating constraints. The optimization problem is repeatedly solved at each supervision period by using a population based stochastic optimization technique (Particle Swarm Optimization). Results of simulations show that the proposed control system is effective and minimizes the input electric power while satisfying both the data center thermal load and system operating constraints.

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Keywords: Cooling Device; Energy Efficiency; HVAC

1. Introduction

Data centers are facilities hosting a large number of servers dedicated to massive computation and storage. They can be seen as a composition of information technology (IT) systems, which provides services to the end users, and a support infrastructure, which supplies power and cooling. Power consumption in data centers has increased significantly in the past few years. In particular, costs for air conditioning systems represent more than one third of the total energy consumption, therefore efficiently cooling data centers has become a challenging problem. New free cooling technologies, coupled with advance control systems, allow the increase of the efficiency of computers room air conditioning (CRAC) systems, [1]. Among the available free cooling technologies, the indirect adiabatic cooling (IAC) grants low energy consumption while meeting air conditioning needs, [2]. Nowadays, advanced control systems give the chance to improve both system efficiency and performance by mostly acting on software components. Modelling and control of data centers is discussed in various works in the latest literature, for example in [3] a control-oriented data centers model is depicted, including the coupling in the dynamics between computational and cooling resources. In [4], a control strategy that provides the best trade-off between energy consumption and cooling needs satisfaction is presented. In [5], a MPC approach is used to obtain a cooling control system for data centers based on the use of indirect fresh air.

In this paper, a model-based approach is developed for the design of an efficient control strategy for CRAC systems. This approach involves three main steps: system modelling, analysing and developing of the controller, and simulating both the plant and the controller. In particular, a model for the IAC system has been obtained by resorting to First-Principle-Data-Driven technique, [6]. Moreover, a simulation environment based on Matlab/Simulink is designed accordingly. Then, a hierarchical control system for CRAC optimal operation has been developed. Specifically, the problem is formulated as a non-linear constrained optimization problem and it is solved by a population based stochastic optimization technique, the PSO (Particle Swarm Optimization). Simulations results show that the proposed control strategy minimizes the input total power while satisfying the operational constraints.

2. Indirect Adiabatic Cooling

In this paper, an IAC system is considered. It exploits the outside air to cool an internal environment, avoiding external and internal airstreams to directly mix. Cool air taken from the outside is forced through a heat exchanger (HX) and then immediately exhausted, whereas internal air is drawn from the room and circulated through the other side of the HX before being re-inserted into the room. In order to economically reduce even more the temperature of the air used for cooling, the outside air is humidified before it enters the heat recovery unit. This reduces the use of traditional expansion air conditioning systems, reducing the cooling costs. This also makes the cooling system effective even during periods when the outside air temperature is warmer than the desired internal room condition. In particular, a system of spray humidifiers delivers moisture to the air using water. The fact that the internal air never mixes with the outside air reduces the possibility of internal air contamination due to external pollutants. The IAC considered system is depicted in Fig. 1. Two distinct air passages can be distinguished, called respectively primary (or supply) and secondary (or process). Main components of the cooling system are fans, spray nozzles, HX, and computers room.

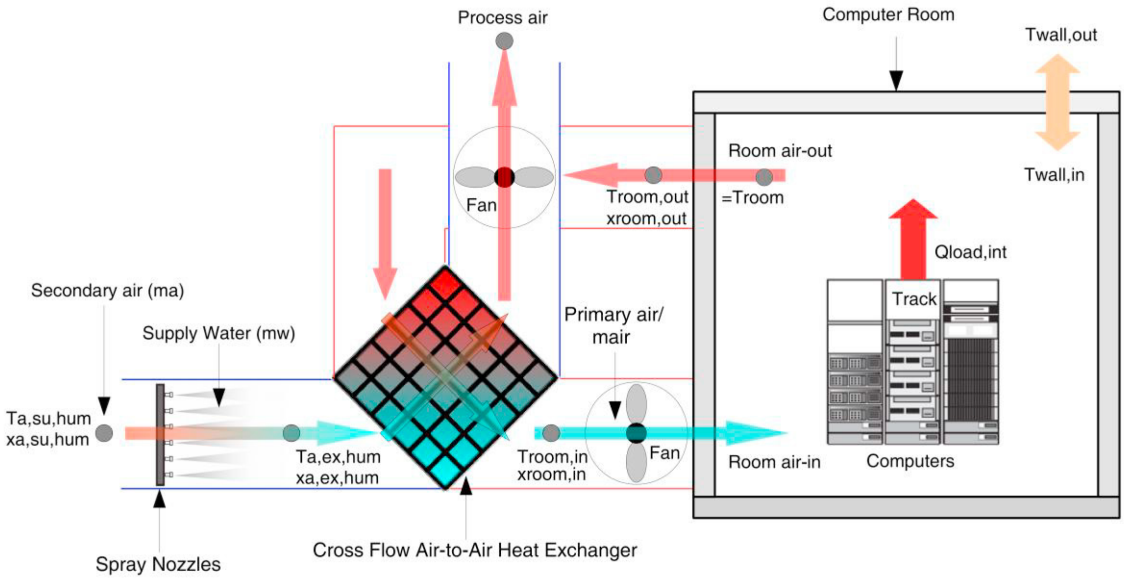

Fig. 1 - Computers room indirect adiabatic cooling system.

3. Modelling of the CRAC system

The first step in the modelling of a real system is collecting and systematic treatment of available knowledge. Depending on the availability of the a priori knowledge, a first-principle or a data-driven model, or a combination of them, can be applied. First-principle (FP) models describe in depth the physics of the system, allowing high capacity of performance prediction, but usually it is difficult to obtain the specific relationships and values of certain parameters, even if the physical principia are known. Instead, data-driven (DD) models exploit experimental data acquired from the plant to obtain a suitable description of the system. Lowering time in the modelling phase is guaranteed by this kind of models, but meaningful information is required in order to cover all the system operation range. To obtain a CRAC model as simple as possible but accurate enough, First-Principle Data-Driven (FPDD) models, which are a combination of FP and DD ones, are here considered. They take advantages of FP and DD approaches and they are suitable for simulation and model-based control design purposes. In the next subsections, as an example, models of spray nozzles, HX, and computers room are described. Further details of those models and a description of that of the fans can be found in [7]. It is worth highlighting that the dynamics of the variables of interest (e.g. temperatures) of the computers room are slower than those of spray nozzles, HX, and fans. Therefore, the computers room is described by a dynamic LTI model, whereas spray nozzles, HX and fans are described by static models.

3.1. Spray Nozzles

In the following, the lumped static model obtained for the spray nozzles is described. Adiabatic humidification process is characterized by negligible variations of wet bulb temperature and enthalpy between inlet and outlet conditions. A 'wet' effectiveness of the process is defined as function of the specific humidity *x*:

$$\varepsilon_{wet} = \frac{x_b - x_a}{x_c - x_a}. \tag{1}$$

The specific humidity x_c is referred to saturation condition, whereas x_a and x_b are the specific humidity of the air at the inlet and outlet sections respectively, Fig. 2.
In fair approximation, the humidifier 'thermal' effectiveness is calculated as function of inlet and outlet dry air temperatures as follows:

$$\varepsilon_{hum} = \frac{T_b - T_a}{T_c - T_a}. \tag{2}$$

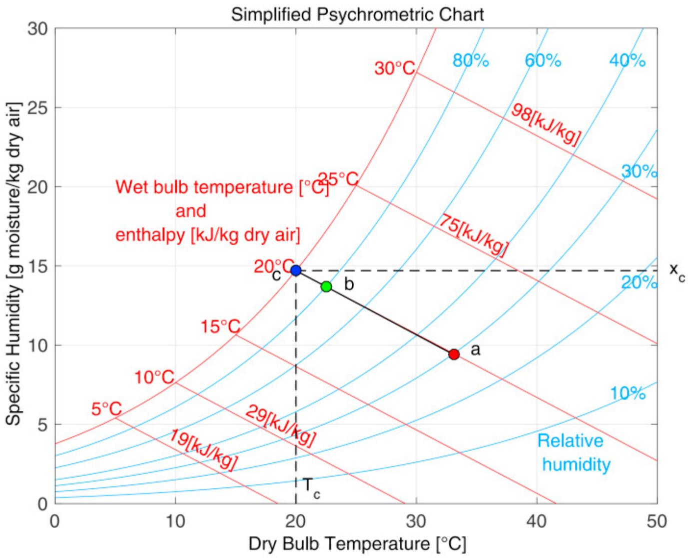

Fig. 2 - Adiabatic humidification process.

The humidifier can be considered as a heat exchanger working with a fictitious moisture fluid, which is characterized by a fictitious specific heat, [8, 9]. Furthermore, the water can be considered as a quasi-isothermal fluid. The humidifier outlet air temperature $T_{a,ex,hum}$ is calculated as function of the inlet air temperature $T_{a,su,hum}$ and the wet bulb temperature $T_{wb,su,hum}$ by using the ε-NTU method (Effectiveness-Number of Transfer Unit), [10]:

$$T_{a,ex,hum} = T_{a,su,hum} + (T_{wb,su,hum} - T_{a,su,hum})\varepsilon_{hum}, \tag{3}$$

with

$$\varepsilon_{hum} = 1 - e^{-NTU}, \tag{4}$$

$$NTU = \frac{AU}{\dot{C}_{\min}}. \tag{5}$$

The evaporative source is assumed as having an infinite capacity flow rate in the calculation of the 'thermal' effectiveness in Eq. (4). In Eq. (5) the term $\dot{C}_{\min}$ is the minimal thermal capacity flow rate.
The fictitious overall heat transfer coefficient AU is calculated as function of both air and water flow rates $\dot{m}_a$ and $\dot{m}_w$ as follows:

$$AU = AU_n \left[\frac{\dot{m}_a}{\dot{m}_{a,n}} \right]^{p_a} \left[\frac{\dot{m}_w}{\dot{m}_{w,n}} \right]^{p_w} . \tag{6}$$

In Eq. (6), AU_n, $\dot{m}_{a,n}$, and $\dot{m}_{w,n}$ are respectively the overall heat transfer coefficient and the air and water flow rates at nominal conditions. Moreover, the parameters p_a and p_w are calibrated on experimental data.
Finally, the humidifier outlet air specific humidify can be calculated as follows:

$$x_{a,ex,hum} = x_{a,su,hum} + \left(x_{wb,ex,hum} - x_{a,su,hum}\right) \frac{T_{a,ex,hum} - T_{a,su,hum}}{T_{wb,ex,hum} - T_{a,su,hum}} . \tag{7}$$

3.2. Heat exchanger

The crossflow heat exchanger HX receives the air taken from the computers room (primary or supply air) and cools it with the external air (secondary or exhaust air), which is treated, if need be, by the adiabatic humidifier. The heat exchanged between primary and secondary airflows can be only sensible, when the specific humidity of both airflows remains constant, or both sensible and latent, when condensation occurs. In fact, when the temperature of the secondary air is lower than the dew point temperature of the supply air, this one condenses, creating a film of water on the supply channels surface of the HX. For these reasons, the overall model of the HX is divided in two sub-models. The 'Dry sub-model' considers only sensible heat exchange phenomena and it is a grey box model calibrated on experimental data. On the other hand, the 'Wet sub-model' describes both sensible and latent heat exchanges.

3.2.1. Dry sub-model

The Fanning friction factor *f* and the Nusselt number *Nu* can be described using the correlations suggested by Abu-Khader and Polley in [11].
The Fanning friction factor *f* is given by:

$$f = \left[\left(\frac{16}{\mathrm{Re}} \right)^3 + \left(n_1 \mathrm{Re}^{n_2} \right)^3 \right]^{\frac{1}{3}} , \tag{8}$$

and the Nusselt number *Nu* is calculated as follows:

$$Nu = \left(Nu_{turb}^{n_{10}} + Nu_{lam}^{n_{10}} \right)^{\frac{1}{n_{10}}} . \tag{9}$$

Laminar and turbulent components of the Nusselt number Nu_{turb} and Nu_{lam} are given by:

$$Nu_{turb} = n_3 \sqrt{f}\, \mathrm{Re}^{n_4}\, \mathrm{Pr}^{n_5} , \tag{10}$$

$$Nu_{lam} = \left[n_6^3 + n_7^3 + \left(n_8 Gz^{n_9} - n_7 \right)^3 \right]^{\frac{1}{3}}, \tag{11}$$

where *Re* is the Reynolds number, *Pr* is the Prandtl number, and *Gz* is the Graetz number.
Once calibrated parameters $n_1, n_2, \ldots, n_{10}$ on experimental data, convective heat transfer coefficients in dry conditions for both supply and exhaust air $\alpha_{dry,S}$ and $\alpha_{dry,E}$ can be calculated as follows:

$$\alpha_{dry} = \frac{\lambda_i Nu}{d_h}. \tag{12}$$

Then, the overall heat transfer coefficient is:

$$K_{dry} = \left(\frac{A_{plates}}{A_S \alpha_{dry,S} \Omega_{dry,S}} + \frac{A_{plates}}{A_E \alpha_{dry,E} \Omega_{dry,E}} + \frac{s_{plates}}{\lambda_m} \right)^{-1}, \tag{13}$$

where A_{plates} and s_{plates} are respectively the total area and the thickness of the plates in the HX, while A_S and A_E are respectively the supply and exhaust side exchange surface areas.
Once obtained the overall heat transfer coefficient, *P-NTU* method is used to calculate outputs for dry channels outflow, that are temperatures, humidities and heat flows.

3.2.2. Wet sub-model

Condensation process occurs in supply air channels when the wall surfaces temperature is lower than the dew point temperature of the supply air. The process is described, for example, by Threlkeld et al. correlation, [12].
The convective heat transfer coefficient of supply air in wet-fin condition can be calculated as function of that in dry-fin condition as follows:

$$\alpha_{wet,S} = \alpha_{dry,S} \frac{b_w}{c_{p,S,i}}, \tag{14}$$

where b_w is the slope of the temperature-specific enthalpy curve of the saturated air at the wall mean temperature t_w:

$$b_w = \left. \frac{\partial h_{sat}}{\partial t} \right|_{t_w}. \tag{15}$$

The Wet sub-model outputs are obtained using the P-NTU method.

3.2.3. Condensation process evaluation

In order to evaluate the condensation process in supply channels, we exploit a fictitious dimensionless coordinate ξ that represents the portion of the channels where the condensation occurs and which is defined as function of the inlet and outlet sections wall temperature $t_{w,A}$ and $t_{w,B}$.

Depending on the value of ξ, weights ω_{dry} and ω_{wet} are accordingly introduced. If $\xi \geqslant 1$ then there is no condensation and $\omega_{dry} = 1$, $\omega_{wet} = 0$, whereas, if $\xi \leqslant 1$ then the supply air channels are completely wet and $\omega_{dry} = 0$, $\omega_{wet} = 1$. If $0 \leqslant \xi \leqslant 1$ then there is a mixed condition and $\omega_{dry} = 1 - \omega_{wet}$, $\omega_{wet} = 1 - \xi$.

Once obtained the weights, sensible, latent and total heat flow rates of the overall HX model are obtained by combining those of the two sub-models as follows:

$$q_{sens} = \omega_{dry} q_{dry} + \omega_{wet} q_{wet,sens}, \tag{16}$$

$$q_{lat} = \omega_{wet} q_{wet,lat}, \tag{17}$$

$$q_{tot} = q_{sens} + q_{lat}. \tag{18}$$

3.3. Computers room

The lumped-parameter thermal dynamic model of the computers room is derived by means of the electrical-thermal analogy. In particular, a RC (Resistance-Capacity) model has been employed, where electrical resistances represent thermal resistances between adjacent nodes, whereas electrical capacities represent nodes thermal capacities, Fig. 3. Air temperatures and humidities of the room and working temperatures of electrical devices are the most important

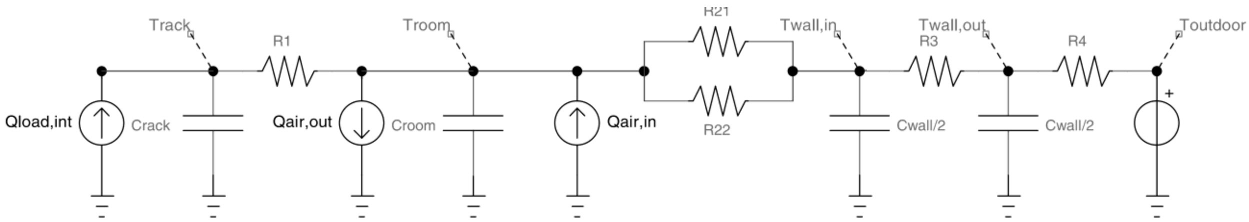

Fig. 3 - The computers room RC model.

variables. Internal and external thermal loads (e.g. $Q_{load,int}$, $T_{outdoor}$) are treated as disturbances. Temperatures and humidities of the primary air are inputs and outputs of the model.
The dynamic of the most important variables (e.g. rack temperature T_{rack}, room temperature T_{room}, wall temperatures $T_{wall,in}$ and $T_{wall,out}$, and the computers room humidity $x_{room,out}$) are described by the following equations:

$$\frac{\partial T_{rack}}{\partial \tau} = \frac{1}{c_{rack}} \left[Q_{load,\text{int}} - \frac{T_{rack} - T_{room}}{R_1} \right], \tag{19}$$

$$\frac{\partial T_{room}}{\partial \tau} = \frac{1}{c_{room}} \left[\dot{m}_{air} \left(h_{room,in} - h_{room,out} \right) - \frac{T_{room} - T_{rack}}{R_1} - \frac{T_{room} - T_{wall,in}}{R_2} \right], \tag{20}$$

$$\frac{\partial T_{wall,in}}{\partial \tau} = \frac{2}{c_{wall}} \left[-\frac{T_{wall,in} - T_{room}}{R_2} - \frac{T_{wall,in} - T_{wall,out}}{R_3} \right], \tag{21}$$

$$\frac{\partial T_{wall,out}}{\partial \tau} = \frac{2}{c_{wall}} \left[-\frac{T_{wall,out} - T_{wall,in}}{R_3} - \frac{T_{wall,out} - T_{outdoor}}{R_4} \right], \tag{22}$$

$$\frac{\partial x_{room,out}}{\partial \tau} = \frac{\dot{m}_{air}}{M_{air,room}} \left(x_{room,in} - x_{room,out} \right), \tag{23}$$

where $R1$, $R2$, $R3$, $R4$ are the thermal resistances, whereas C_{wall}, C_{rack}, C_{room} are the thermal capacities. $M_{air,room}$ represents the total mass of air in the computers room and $\dot{m}_{air}$ is the room air flow rate.

4. Control of CRAC systems

The optimization of the CRAC system operation guarantees significant energy savings. The optimal operation is obtained by choosing the best combination of the modes of operation that minimises the input power consumption while satisfying cooling load demand and constraints (e.g. computers room air temperature restrictions). It is worth noticing that, the cooling needs can be achieved with various combinations of CRAC modes of operation. In Fig. 4, the iso-temperature computers room curves are depicted as function of air process flow rate and supply water flow rate.

In this paper, we employ a hierarchical control system. Local control loops (e.g. standard regulators) are used to regulate the water-flow sprinkled by spray nozzles and the air-flow rate by process fan, which characterize the mode of operation of the CRAC system. A supervisory control loop has to determine the set-points for the local loops minimizing the power consumption, while satisfying operational constraints.

For this purpose, at each supervision period (e.g. 15 min), the local set-points are obtained by solving a non-linear constrained optimization problem, which can be formulated on the given supervision time interval as follows:

$$\arg\min_{\mathbf{u}} J(\mathbf{x}, \mathbf{u}, \mathbf{d}), \textit{ subject to:} \tag{24a}$$

$$\Sigma, \tag{24b}$$

$$h(\mathbf{x}, \mathbf{u}, \mathbf{d}) \le 0, \tag{24c}$$

where J is the objective function to minimize, e.g. the total input electric power. In Eq. (24b), Σ represents the overall CRAC model, which includes fans, spray nozzles, HX, and the computers room. States (i.e. air temperatures and humidities), inputs (i.e. air and water flow rates) and disturbances (i.e. cooling load) of the system are the vectors **x**, **u**, **d**, while **y** are the output variables (i.e. input power consumption, air temperatures and humidities). Finally, Eq. (24c) represents the constraints on the CRAC modes of operation.

To solve the optimization problem (24) we have employed an ad-hoc PSO (Particle Swarm Optimization) algorithm, which intensifies its search ability in specific promising regions, [13]. PSO is a derivative-free, stochastic and population-based algorithm, inspired by the motion of bird flocks or fish schooling. It is often used to optimize functions in rather unfriendly non-convex, non-continuous search spaces, [14]. Each particle of the swarm represents a candidate solution and it moves with its own velocity in the multidimensional search space, determines its own position and calculates its fitness using an objective function. The interaction between particles grants global and collective search capabilities, allowing the particles to move towards the global extremum.

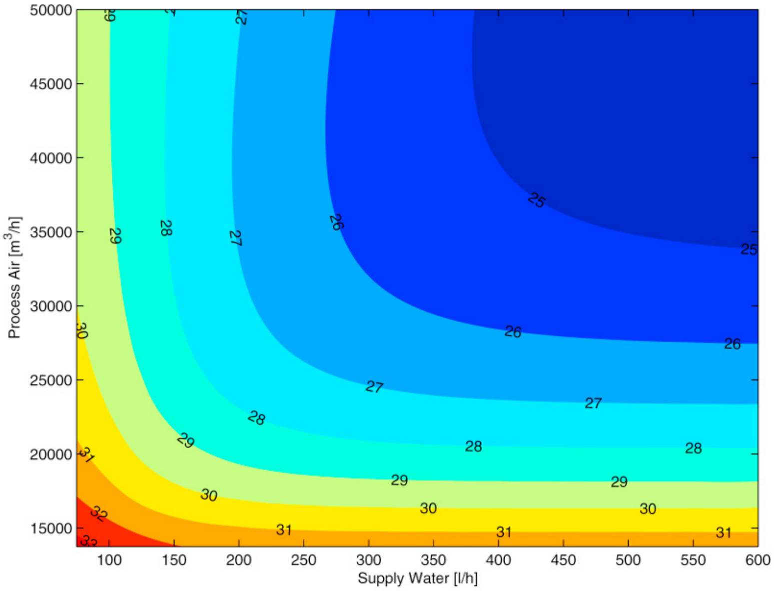

Fig. 4 - The iso-temperature computers room curves

In the following, an example of application of the control strategy is reported. The aim is to determine the best combination of CRAC water and air flow that minimizes the total electric power consumption while ensuring fixed computers room air temperature conditions. The air-flow of the computers room is kept constant at its nominal value and initial CRAC nominal condition are depicted in Table 1.

Table 1. CRAC nominal conditions.

Cooling load	$Q_{load,int}$	75	kW
External temperature	$T_{outdoor}$	25	$°C$
Process inlet temperature	$T_{process,in}$	25	$°C$
Process inlet humidity	$x_{process,in}$	$8 \cdot 10^{-3}$	$kg_{wv}kg_{da}^{-1}$
Process outlet temperature	$T_{process,out}$	25	$°C$
Process outlet humidity	$x_{process,out}$	$8 \cdot 10^{-3}$	$kg_{wv}kg_{da}^{-1}$
Process air flow-rate	$\dot{m}_a$	$15 \cdot 10^{3}$	m^3h^{-1}
Room inlet temperature	$T_{room,in}$	25	$°C$
Room inlet humidity	$x_{room,in}$	$8 \cdot 10^{-3}$	$kg_{wv}kg_{da}^{-1}$
Room outlet temperature	$T_{room,out}$	25	$°C$
Room outlet humidity	$x_{room,out}$	$8 \cdot 10^{-3}$	$kg_{wv}kg_{da}^{-1}$
Room air flow-rate	$\dot{m}_{air}$	$39 \cdot 10^{3}$	m^3h^{-1}
Supply water	$\dot{m}_w$	100	s^{-1}

The solution of the optimization problem (24) is obtained for various values of the constraint on the computers room air temperature: in Fig. 5 the green stars represent the optimal combinations of water and air flows. It is worth noticing that, for a fixed thermal load, as the desired computers room temperature decreases, then it is more convenient, in terms of energy savings, increase the water-flow rather than increase the secondary air-flow.

In Fig. 6 the green stars represent the energy consumption that corresponds to the best combination of the parameters for various values of the constraint on the computers room temperature.

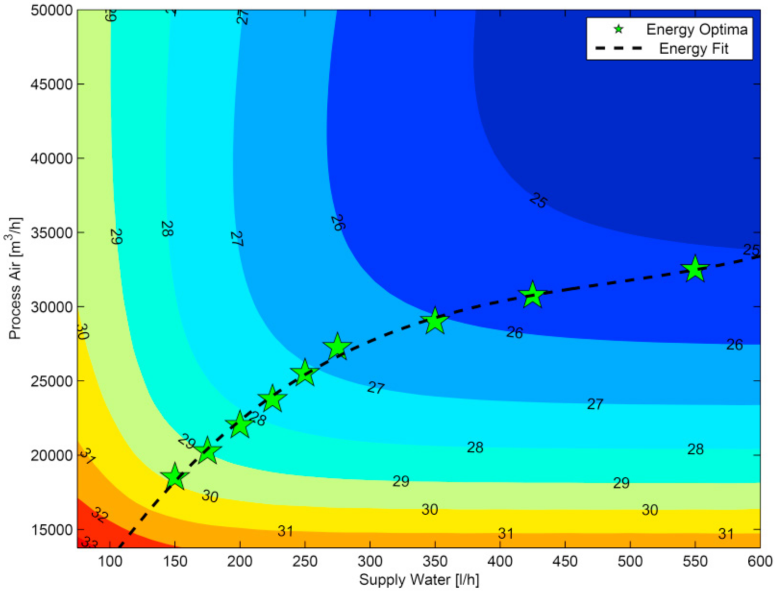

Fig. 5 - Optimal combinations of modes of operation.

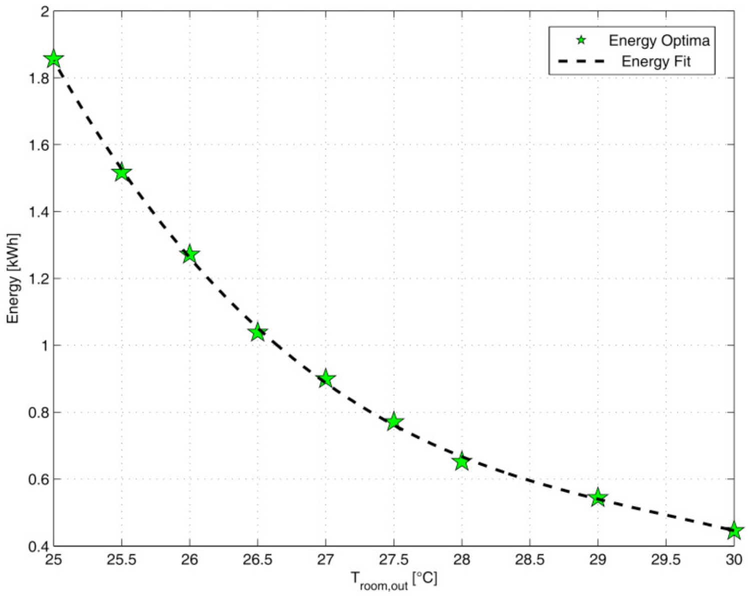

Fig. 6 - Energy consumption corresponding to the best combination of the modes of operation.

Conclusions

In this paper, an efficient control strategy for IAC systems for data centers has been designed using a model-based optimization approach. In particular, the main components of the CRAC system, such as spray nozzles, heat exchanger and computers room, have been modelled through FPDD techniques, granting simplicity and reliability. A Matlab/Simulink simulation environment has been developed accordingly. The operation of the cooling system is optimized by minimizing the energy consumption while satisfying the constraint on the cooling demand. The corresponding nonlinear and constrained optimization problem has been successfully solved by using a PSO algorithm. The effectiveness of the proposed control architecture is confirmed by simulation results.

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