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A novel solar-assisted air-conditioner system for energy savings with performance enhancement

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Abstract

This paper presents an effective technique to enhance the performance of a newly-developed direct expansion air conditioner when combining with a vacuum solar collector that is installed after the compressor. In this approach, a novel configuration including a by-pass line together with a three way proportional control valve is proposed in discharge line after the compressor in order to control the refrigerant flow rate. In this design, the refrigerant flow rate is controlled as a function of the refrigerant temperature leaving the compressor, the refrigerant temperature leaving the solar storage tank and the ambient dry-bulb temperature. A generalized optimization algorithm is developed using sequential quadratic programming (SQP) along with a proposed empirical model for the objective function. The key challenge is to estimate the optimum refrigerant temperature entering the condenser in the new design. The optimization algorithm is simulated in a transient simulation tool to predict the optimum set-points of refrigerant temperature entering the condenser, and then implemented as a reference for an on-line closed-loop controller. The system under investigation is extensively equipped with a number of instrumentation devices for data logging. The benefits of the new design lie in the fact that the new designed system operates at a higher subcool temperature after the air-cooled condenser which significantly result in increasing the overall system coefficient of performance.

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Keywords: by-pass discharge line, optimal control, performance improvement, solar-assisted air-conditioner

Nomenclature

| Symbol | Description                                      |
|--------|--------------------------------------------------|
| COP    | Coefficient of performance                      |
| \( h \) | Refrigerant enthalpy (kJ/kg)                     |
| \( T_{\text{com}} \) | Refrigerant temperature leaving the compressor (°C) |
| \( T_{\text{db}} \) | Ambient dry-bulb temperature (°C)                |
| \( T_{\text{rec}} \) | Refrigerant temperature entering the condenser (°C) |
| \( T_{\text{rlc}} \) | Refrigerant temperature leaving the condenser (°C) |
| \( T_{\text{wst}} \) | Refrigerant temperature leaving the water storage tank (°C) |

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1. Introduction

Continuing increase in the global electricity demand and depletion of fossil fuel resources have necessitated a paradigm shift in the development of eco-friendly and energy-efficient technologies. Summer air-conditioning is significantly contributing to the peak electricity demand, affecting the cost of maintaining a reliable electricity supply, and hence remains a challenging problem. Among possible solutions is the use of renewable energy sources, of which solar energy is very promising as it is abundantly present and environmentally sustainable. Therefore, solar-assisted cooling systems using solar thermal panel as heat source are of increasing interest to academia and industry [1-6]. The direct expansion (DX) air conditioning plant is one of the main cooling systems for different types of buildings which is simpler in configuration and more cost-effective to maintain than central cooling plants using chillers and cooling towers [7,8]. Therefore, an ongoing research challenge is to improve the performance of the solar-assisted DX air conditioners. Different control strategies were used and compared in a solar cooling system installed in an office building by Bujedo et al. [9]. The objective of the study was to emphasise the importance of regulation strategies for correcting partial load with sorption chillers. Results showed that a better coefficient of performance (COP) can be achieved when using closed-loop control in comparison with a conventional system. A sliding mode predictive control was applied to an air conditioning solar plant by Garcia-Gabin et al. [10], where considerable improvement is achieved in terms of robustness and the enhanced ability to handle set-point changes. Huang et al. [11] developed an optimal control approach for an ejector solar-assisted cooling system which is connected in parallel with an inverter-type air-conditioner. To obtain a better performance, an electronic expansion valve was installed in the suction line of the ejector to regulate the opening of the expansion valve for controlling the evaporator temperature. Results showed a total energy saving of 52% over the entire test period. Lygouras et al. [12] designed and implemented a two-input/two-output variable structure fuzzy-logic controller for a solar-powered air conditioning system, where heuristic rules are utilized to effectively improve the control performance. The application of a model predictive controller to the temperature control in a solar air conditioning plant was presented by Nunez-Reyes et al. [13], using a conventional Smith predictor and a feed forward action. Their experimental tests showed that the approach can be feasible for a solar air conditioning plant. So far, no work has been reported in the literature on performance improvement of a new hybrid solar-assisted DX air-conditioner in which the vacuum solar collector is installed after its compressor.

The objective of this study is to improve the inherent operational characteristics of a new direct expansion (DX) air-conditioner when combining with a vacuum solar collector. In the solar-assisted unit under investigation, the solar collector is installed after the compressor in order to reduce the compressor work to produce the required essential temperature for the refrigerant entering the condenser. Therefore, the compressor can be sized solely with additional heat input for condenser heat rejection by the solar collector, which allows the system to save around 40% of electrical energy. However, it increases the refrigerant temperature leaving the condenser because of the additional heat added before the condenser especially in the partial load condition. This results in some reduction of the system coefficient of performance (COP). On the other hand, when the ambient temperature and hence the essential condensing temperature are less than the designed condensing temperature in a partial load, the refrigerant temperature entering the condenser can be lower by some degrees than the temperature required at full load for perfect heat rejection in the condenser. This is because in partial loads the condensing temperatures are normally less than the designed condensing temperature. Therefore, the optimum set-points for the refrigerant temperature entering the condenser should lower the refrigerant temperature leaving the condenser and, in turn, increase system overall COP. In principle, a proper control strategy for the refrigerant temperature entering the condenser can lead to more energy-efficient operations [14]. For this purpose, a synergetic framework of system identification, empirical modeling, optimization and control has been proposed [15]. To achieve this, an appropriate modification should be applied to adapt the plant with changing operational conditions caused by a transient cooling load and varying ambient conditions. Here, a novel configuration including a by-pass line together with a three-way proportional valve is proposed in the discharge line after the compressor in order to control the refrigerant temperature. Several field tests were conducted for system identification and control purposes. Explicit relationships of the air conditioning system process characteristics with respect to the controlled and uncontrolled variables under operation conditions are established by using experimentally-collected data and empirically-based regression. The temperature of the refrigerant leaving the condenser can be then expressed as a function of these variables in a proposed quadratic equation to facilitate the derivation of minimum temperature leaving the condenser. Experimental data and the sequential quadratic programming based optimization algorithm are implemented on, TRNSYS (version 16), a transient simulation tool [16], to solve the minimization problem and predict the system set-points under transient loads. A simple closed-loop control strategy is designed and implemented. Results show that the new implemented mechanical design together with its proposed control system can yield to higher subcool temperature after the condenser can increase further the overall coefficient of performance to around 6.7%.
2. New Design Description

A single-stage vapor compression solar air-conditioner consists of six major components, namely a compressor, a condenser, an expansion device, an evaporator, a solar vacuum collector and a solar storage tank. Figure 1 shows a schematic block diagram of the new hybrid solar air conditioning system (HSAC). The cycle starts with a mixture of liquid and vapor refrigerant entering the evaporator (point 1). The heat from warm air is absorbed by an evaporator DX coil. During this process, the state of the refrigerant is changed from a liquid to a gas and becomes superheated at the evaporator exit. The superheat vapor then enters the compressor (point 2) where a rising pressure will in turn increase the temperature. A vacuum solar panel installed after the compressor, uses solar radiations as a heat source to warm up the water. An insulated water storage tank is connected to the vacuum solar collector to maintain the water temperature. Therefore, the vacuum solar collector reheats the refrigerant to reach the necessary superheat temperature in order to reduce the required electrical energy to run the compressor. However, a disadvantage of this design is that the storage tank tends to increase the refrigerant temperature entering the condenser even in the partial load condition. This is accounted for by the fact that the condensing temperature is not constant due to changes in the ambient temperature. Therefore, the necessary refrigerant temperature entering the condenser at partial loads is less than the nominal one at full load.

In this study a novel design is implemented to the system to increase the refrigerant sub-cool temperature. A three-way proportional control valve is installed after the compressor to regulate the refrigerant mass flow rate. Therefore, the control valve can adjust the temperature of the refrigerant entering the condenser. During partial loads, when the condensing temperature is low, the refrigerant superheat temperature leaving the compressor is normally sufficient for condenser heat rejection and thus the control valve shall lead the refrigerant to directly go through the new by-pass discharge line. During a high cooling demand when the condensing temperature is high, the refrigerant from the compressor goes through the copper coil inside the tank where a heat exchange is undertaken (point 3). Since the nature of building cooling loads and ambient condition is highly transient, the control valve should act continuously to optimally keep the desired temperature entering the condenser. After the control valve, the high-pressure superheated gas travels to the condenser for heat rejection of the ambient air (point 4). A further reduction in the refrigerant temperature takes place in the condenser, causing it to de-superheat and thus the refrigerant liquid is sub-cooled as it enters the expansion device. The high-pressure sub-cooled refrigerant flows through the expansion device (point 5) which serves to reduce its pressure and temperature as well as to control the flow rate of the refrigerant entering the evaporator.

Fig. 1. Schematic diagram of a hybrid solar assisted air conditioner
3. Experimental Set-Up

The experimental work has been carried out using an experimental solar DX hybrid air-conditioner together with its programmable logic controller (PLC) device as shown respectively in Fig. 2(a) and 2(b). The experimental set-up is mainly composed of two parts, i.e., an air-conditioned room served by the DX evaporator unit and the condensing unit which is combined with a solar vacuum collector. The floor area of the air-conditioned room is 38 square meters and its high is 3 meters. The nominal output cooling capacity of the air-conditioner is 6 kW. The plant has one scroll hermetic compressor and uses refrigerant R410A as a working fluid. The amount of charged refrigerant into the system is 1.8 Kg. The design air flow rate of the evaporator fan is 850 $\text{m}^3/\text{h}$. The condenser in the plant is an air-cooled shell and of a coated fin tube type. The design air flow rate of the condenser axial fan is 2500 $\text{m}^3/\text{h}$ and its rated power input is 50 W. The collector is integrated at a tilted angle of 5° and oriented towards North (for Southern Hemisphere). The solar collector is equipped with a 35-litre horizontal cylinder as a storage tank. The existing HSAC system has been fully-instrumented to facilitate a number of tests under different operation conditions. High precision sensors/transducers are used for measuring all operating variables. The following parameters are measured: meteorological parameters (global solar radiation, ambient temperature and ambient relative humidity), solar storage tank water temperature, indoor temperature and relative humidity, temperature of refrigerant before and after solar storage tank and after condenser and evaporator and total power consumption of the plant. The global radiation is measured by means of a solarimeter of the type CR100-A mounted on a surface parallel to the plane of the collector. The solarimeter is integrated to a computer based-KT250 datalogger. The outdoor temperature and relative humidity are measured by using a combined sensor-datalogger device type with a calibrated accuracy of $\pm 0.4^\circ \text{C}$ for temperature and $\pm 2\%$ for relative humidity. The measuring range for temperature is between -20°C to +70°C, and from 5% to 95% for humidity. The sensor is integrated with a computer based-KH250-AO datalogger. A vertical array of semiconductor temperature sensor of model LM335A with accuracy $\pm 2^\circ \text{C}$, mounted in a plastic tube with a current limiting resistor is located inside the solar storage tank to measure the average temperature of the water inside the tank. The temperature sensors for refrigerant are of resistive transducer type with a calibrated accuracy of $\pm 0.5^\circ \text{C}$ for a temperature range from -40°C to +85°C. The sensors are in the range from -40°C to +100°C, connected to the KH-250 type datalogger, used to measure the refrigerant temperature leaving the water storage tank, compressor, condenser, capillary tube and evaporator.

Fig. 2. (a) Experimental hybrid solar-assisted air conditioner; (b) Programmable logic controller implementation

The outputs from the sensors were recorded by means of a computer based-KT20L datalogger. For this purpose, the surface of discharge and liquid tubes were polished to remove any dust and rust and then the temperature sensors were fixed onto the surface. In order to reduce the thermal contact resistance between sensors and the tube surface, a thermal grease tape was wrapped around the tube to push the sensors against the tube surface and also prevent any convection effects of the ambient air on the temperature readings. Powers of the HSAC components are measured by a digital ac/dc power clamp multimeter of precision $\pm 3.5\%$. Field tests were run continuously to monitor the system performance at various weather conditions. Day-long tests were carried out for 24 hours and all measured data are monitored with an interval of 10 min. All measurements were then computerized so that all the measured data can be recorded for subsequent analysis.
4. Optimization and control development

A control strategy is developed to control the proportional valve. In this system the energy efficiency is closely dependant on the thermodynamic properties of the refrigerant. This section details the generation of the desired temperature of the refrigerant entering the condenser to minimize the temperature when leaving the condenser by closed-loop control of its flow rate.

4.1 Reference generation

In principle, the proposed design for the HSAC can be characterised by the refrigerant temperatures leaving the condenser and leaving the storage tank. A proper regulation of these state variables and their dynamic behaviour can lead to more energy-efficient operations. The minimization problem for refrigerant temperature leaving the condenser is then formulated through the determination of controlled variables, subject to constraints. The basic idea is to formulate and solve a quadratic programming sub-problem at each iteration by linearizing the constraints and approximating the Lagrangian function. For this, we consider in general the nonlinear constrained optimization problem to minimize the objective function \( f(x) \) under \( m \) nonlinear equality constraints as:

\[
\begin{align*}
\text{Minimize} & \quad f(x) \\
\text{subject to} & \quad g(x) = 0, \quad x \in \mathbb{R}^n
\end{align*}
\]  

where \( f : \mathbb{R}^n \rightarrow \mathbb{R} \) is the objective function and \( g : \mathbb{R}^n \rightarrow \mathbb{R}^m \) describes the equality constraints. The sequential quadratic programming (SQP) is an iterative procedure to deal with the nonlinear optimization problem for a given \( x^k (k \in N_0) \) by a linearized quadratic programming subproblem and then uses its numerical solution to construct a new iterative \( x^{k+1} \).

Therefore, the SQP optimization problem can be formulated as:

\[
\begin{align*}
\text{Minimize} & \quad \sum_{i=1}^{m} \lambda_i \nabla^2 g_i(x^k) \\
\text{subject to} & \quad g(x^k) + \nabla g(x^k)^T d_k = 0, \quad d_k \in \mathbb{R}^n
\end{align*}
\]

where \( \lambda \) is the Lagrange multiplier, \( d_k \) is a stationary point to minimize the object function if satisfying sufficient conditions, and the Hessian matrix of the Lagrangian function is introduced as:

\[
\nabla^2 \ell(x^k, \lambda^k) = \nabla^2 f(x^k) + \sum_{i=1}^{m} \lambda_i \nabla^2 g_i(x^k),
\]

in which the Lagrangian function is:

\[
\ell(x^k, \lambda^k) = \nabla f(x^k)d_k + \frac{1}{2} d_k^T \nabla^2 \ell(x^k, \lambda^k)d_k + \lambda^k \left( g(x^k) + \nabla g(x^k)^T d_k \right).
\]

From the following gradients:

\[
\nabla x \ell(x^k, \lambda^k) = \nabla^2 \ell(x^k, \lambda^k)d_k + \nabla f(x^k) + \nabla g(x^k) \lambda^k,
\]

\[
\nabla \lambda \ell(x^k, \lambda^k) = g(x^k) + \nabla g(x^k)^T d_k,
\]

the necessary conditions for optimization are formulated as a Karush-Kuhn-Tucker (KKT) algorithm:

\[
\begin{bmatrix} 
\nabla^2 \ell(x^k, \lambda^k) & \nabla g(x^k)^T \\
\nabla g(x^k)^T & 0 \\
\end{bmatrix} \begin{bmatrix}
\begin{array}{c}
d_k \\
\lambda^k \end{array}
\end{bmatrix} = \begin{bmatrix} 
\nabla f(x^k) \\
g(x^k) \\
\end{bmatrix}.
\]

Here, the aim is to minimize the refrigerant temperature leaving the condenser in order to increase the system COP. The controlled variables include refrigerant temperature leaving the water storage tank and entering the condenser while uncontrolled variables are the temperatures of the refrigerant leaving the compressor and the ambient dry-bulb. For this, the objective function can be explicitly established as follows:

\[
\begin{align*}
\text{min } T_{rc} &= \min_{a_0 + a_1 T_{rec} + a_2 T_{rec}^2 + a_3 T_{rec} T_{wst} + a_4 T_{rec} T_{com} + a_5 T_{rec} T_{db} + a_6 T_{wst} + a_7 T_{wst}^2 + a_8 T_{wst} T_{db} + a_9 T_{com} + a_1 T_{com}^2 + a_2 T_{com} T_{db} + a_3 T_{db} + a_4 T_{db}^2},
\text{subject to constraints}
\end{align*}
\]
where coefficients from $a_0$ to $a_{14}$ are constant values determined by curve-fitting the experimental data. The constraint for optimization problem represents a relationship between various independent variables that must be satisfied and can be obtained using the collected data as follows:

$$b_0 T_{\text{rec}} + b_1 T_{\text{wst}} + b_2 T_{\text{rec}} T_{\text{wst}} + b_3 T_{\text{rec}}^2 T_{\text{wst}} + b_4 T_{\text{rec}}^2 T_{\text{wst}}^2 = 0,$$

where coefficients from $b_0$ to $b_4$ are constant and determined again by curve-fitting the experimental data. According to the data obtained by field tests, the corresponding coefficients of the optimization and constraint are obtained by a regression technique by using the MINITAB statistical tool [17]. The R-square value of the model indicates a good fit. These coefficients obtained for the aforementioned system are listed below:

$$a_0 = -511.817 \quad a_1 = 11.904 \quad a_2 = 0.110 \quad a_3 = -0.104 \quad a_4 = -0.349$$

$$a_5 = 0.085 \quad a_6 = 9.964 \quad a_7 = -0.001 \quad a_8 = -0.105 \quad a_9 = 0.123$$

$$a_{10} = -7.654 \quad a_{11} = 0.329 \quad a_{12} = -0.098 \quad a_{13} = 3.024 \quad a_{14} = -0.185$$

The KKT optimization algorithm is coded into the TRNSYS simulation studio using FORTRAN to compute the optimum desired refrigerant temperature entering the condenser and leaving the storage tank. The generated optimal set-points are used as the reference to the control system and each new point is updated to the objective function for the desired system response.

### 4.2 Modeling and control

Based on the heat transfer processes in buildings, ordinary differential or difference equations (ODEs) with unknown parameters can be developed. In many intricate cases, the problem may be too complicated to explicitly solve, recursive identification algorithms are used [18], based on experimental data collected. Here, by assuming isentropic operations of the compressor and assigning the system state variables as the temperatures of the refrigerant when leaving the compressor, leaving the water storage tank and entering the condenser, a linearized model of the designed HSAC is obtained as:

$$\dot{x} = Ax + Bu, \quad y = Cx$$

where $x = [T_{\text{com}} \ T_{\text{wst}} \ T_{\text{rec}}]^T$, $u$ is the refrigerant mass flow rate (in g/s), and from data collected:

$$A = \begin{bmatrix} -0.009 & 0.021 & -0.017 \\
7.61 & -0.091 & -6.982 \\
-0.008 & 0.09 & -0.014 \\
\end{bmatrix}, \quad B = \begin{bmatrix} 0 \\
0 \\
-0.139 \\
\end{bmatrix}, \quad C = \begin{bmatrix} 0 & 0 & 1 \end{bmatrix}.$$

To obtain zero tracking error for system (11) of type 0 with a transfer function

$$\frac{Y(s)}{U(s)} = C(sI - A)^{-1}B = \frac{-0.1392s^2 - 0.01392s + 0.02213}{s^3 + 0.114s^2 + 0.4707s + 0.01389},$$

a simple PI controller $K_p(1 + \frac{1}{T_i s})$, where $K_p = 0.3865$ and $T_i = 500$ sec, is implemented on the PLC. Furthermore, the overall coefficient of performance for the system can be estimated as:

$$\text{COP} = \frac{\text{refrigeration effect}}{\text{compressor work}} = \frac{h_2 - h_i}{h_3 - h_2},$$

where $h_i$ is the enthalpy at point $i$ shown in Fig. 1. From (13), it is clear that there are two possible modifications to improve the system performance, namely by increasing the refrigeration effect or decreasing the compressor power [19]. Thus, the new design can increase the refrigeration effect and in turn increase the COP via minimization of the refrigerant temperature leaving the condenser.
5. Results and Discussion

This section describes the effectiveness of the proposed approach on the system performance. The monitored data for the refrigerant temperatures leaving the DX evaporator and compressor, the storage tank water temperature and the ambient dry-bulb temperature are respectively shown in Fig. 3. The optimal set-points for the temperatures of the refrigerant entering the condenser and leaving the water storage tank then are continuously determined and compared with those from a conventional system. These set-points aim at minimizing the refrigerant temperature leaving the condenser. The temperature variations of both the refrigerant entering and leaving the condenser with and without the new design are shown in Fig. 4. The results show that both temperatures will drop by using the new design. The reason is that in a conventional system, the temperature of the refrigerant leaving the condenser increases with any increase in the refrigerant temperature after the water storage tank while with the proposed design, this temperature is controlled according to the required refrigerant temperature entering the condenser. Furthermore, the lower refrigerant temperature leaving the condenser decreases the refrigerant temperature entering evaporator and in turn increases the refrigeration effects. On the other hand, as the refrigerant temperature in the condenser outlet drops, the refrigerant temperature in the evaporator inlet also drops and the enthalpy of the refrigerant entering the evaporator is lowered. This reduction allows the refrigeration effect to increase significantly and thus increasing the evaporator capacity. Furthermore, sub-cooling the refrigerant after the condenser prevents flash gas forming at the inlet of the expansion device. Therefore as emphasized earlier, this approach can increase the system coefficient of performance, as clearly shown in Fig. 5. It can be seen that the average COP by using the optimal set-point values is around 6.7% more than under the commonly-used design. Moreover, the lower refrigerant enthalpy entering the evaporator tends to decrease the compressor work and in turn avoids the unnecessary heat rejection in the condenser.

Fig. 3. Experimental monitored data

Fig. 4. Temperatures of refrigerant entering and leaving the condenser

Fig. 5. System coefficient of performance
6. Conclusion

In this paper we have presented a new design together with its control approach for a hybrid solar-assisted air-conditioner to demonstrate the possibility of improving the system coefficient of performance. In this design a by-pass line is implemented in the discharge line after the compressor to control the refrigerant mass flow rate by using a three-way proportional valve. The sequential quadratic programming based optimization-simulation method is developed to generate the optimal set-points for the refrigerant temperature leaving the water storage tank and entering the condenser as a reference for a PI controller. The objective function and constraint were determined by using experimentally-collected data and empirically-based regression from the existing system. The optimization algorithm, cast in a KKT algorithm, is then implemented within TRNSYS environment. The obtained results for the closed-loop system under optimal input have shown that the new design delivers higher system efficiency than a conventional system. This is due to the higher refrigeration effect in the direct expansion evaporator which allows for a performance gain. Thus, the new design is promising for an improvement of the system performance while fulfilling the cooling demand with guaranteed energy efficiency.

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References

[1] Henning, H. Solar assisted air conditioning of buildings—an overview. Applied Thermal Eng., vol.27, 2007, pp. 1734-1749.
[2] Wang Q, Liu YQ, Liang GF, Li JR, Sun SF, Chen GM. Development and experimental validation of a novel indirect-expansion solar-assisted multifunctional heat pump. Energy and Buildings; vol. 43, 2011, pp. 300-304.
[3] Al-Alli A, Hwang Y, Radermacher R, Kubo I. A high efficiency solar air conditioner using concentrating photovoltaic/thermal collectors. Applied Energy; vol. 93, 2012, pp. 138-147.
[4] Zambrano D, Bordons C, Garcia-Gabin W, Camacho EF. Model development and validation of a solar cooling plant. International Journal of Refrigeration; vol. 31, 2008, pp. 315-328.
[5] Kim DS. Feasibility of a compact heat recovery ventilation module with an integrated air-cooled solar absorption air-conditioner. International Journal of Thermal Sciences; vol. 50, 2011, pp. 1604-1614.
[6] Audah N, Ghaddar N, Ghali K. Optimized solar-powered liquid desiccant system to supply building fresh water and cooling needs. Applied Energy; vol. 88, 2011, pp. 3726-3736.
[7] Li Z, Deng S. An experimental study on the inherent operational characteristics of a direct expansion (DX) air conditioning(A/C) unit. Energy and Buildings; vol. 40, 2008, pp. 394-398.
[8] Vakiloroaya V, Zhu JG, Ha QP. Modelling and Optimisation of Direct Expansion Air Conditioning Systems for Commercial Building Energy Savings. Proc. Int. Symp. Automation and Robotics in Construction, Seoul, Korea, 2011, pp. 232-237.
[9] Bujedo LA, Rodriguez J, Martinez PJ. Experimental results of different control strategies in a solar air-conditioning system at part load. Solar Energy; vol. 86, 2011, pp. 1302-1315.
[10] Garcia-Gabin W, Zambrano D, Camacho EF. Sliding mode predictive control of a solar air conditioning plant. Control Engineering Practice; vol. 17, 2009, pp. 652-663.
[11] Huang BJ, Yen CW, Wu JH, Liu JH, Hsu HY, Petrenko VO, Chang JM, Lu CW. Optimal control and performance test of solar-assisted cooling system. Applied Thermal Engineering; vol. 30, 2010, pp. 2243-2252.
[12] Lygouras JN, Kodogiannis VS, Pachidis TH, Tarchanidis KN, Koukourlis C. Variable structure TITO fuzzy-logic controller implementation for a solar air-conditioning system. Applied Energy; vol. 85, 2008, pp. 190-203.
[13] Nunez-Reyes A, Normey-Rico JN, Bordons C, Camacho EF. A Smith predictive based MPC in a solar air conditioning plant. Journal of Process Control; vol. 15, 2005, pp. 1-10.
[14] Vakiloroaya V, Samali B, Madadnia J, Ha QP. Component-wise optimization for commercial central cooling plant. International Conference of the IEEE Industrial Electronics (IECON); Melbourne, Australia, 2011, pp. 2686-2691.
[15] Ha, Q. Data acquisition, monitoring and control for hybrid solar air-conditioners. Gerontotechnology; vol. 11, 2012, p. 314.
[16] TRNSYS software. A transient system simulation program., version 16. <http://sell.me.wisc.edu/trnsys/>.
[17] Minitab user’s guide release 16, Minitab Inc., 2010.
[18] Wigren, T. Recursive prediction error identification and scaling of nonlinear state space models using a restricted black box parameterization. Automatica, vol. 42, no. 1, pp. 159–168, 2006.
[19] Afonso CFA. Recent advance in building air conditioning systems. Applied Thermal Eng.; vol. 26, 2006, pp. 1961-1971.