



KKU Engineering Journal

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Humidity and temperature control in an evaporative cooling system of a poultry house

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Received September 2011

Accepted January 2012

Abstract

This paper presents a mathematical model and a robust control technique for temperature and humidity control in an evaporative cooling system of a poultry house. The model was considered from mass and energy balance relations of air and water in the system. To validate the mathematical model, its responses were compared with a real system by using a set of feed forward experimental signals. Additionally, the well known sliding mode control with decoupling control law was also applied to the model. The simulation in case of summer conditions shows its behavior and demonstrates the ability of the proposed control technique in order to compensate for the changing ambient air conditions around the house.

Key words: Sliding mode control, Evaporative cooling control, Poultry house

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1. Introduction

Currently, the number of domestic poultry houses that have changed from open to closed system has increased for the sake of inside air condition control. Such systems use evaporative air conditioning systems for an appropriate adjustment of temperature and moisture content. This change results in higher production per unit of poultry house. With suitable design of temperature and moisture controller, increment of production and energy efficiency in terms of equipment used can be achieved. To reach that point, a mathematical model that elaborates relationship between temperature and humidity has to be derived first.

To validate the model, its response is compared with a real plant (Fig.1). The plant has a size of $14 \times 125 \times 4 \text{ m}^3$, the number of chicken is 60,000, and a chicken weighs 2 kg. For more details of the plant, see Table 1.

This work is organized as follows. Section 2 constructs a set of mathematical models. Section 3 introduces the response comparison between the model and the real plant, while in Section 4, simulation in the case of summer is presented. Finally, conclusions are given in Section 5. Figure 2 shows the Research methodology.

Table 1 Plant description

Nomenclature	Value
ρ_{air}	1.2 kg/m ³
C_p	1.005 kJ/(kg·K)
V_T, V_H	70% of poultry house size (14 x 125 x 4 m ³)
N_{ch}	60,000 unit
U_A	0.712 kW/K [6]
λ	2.257 kJ/g _{H₂O}

where

- ρ_{air} is air density
- C_p is specific heat of air
- V_T, V_H is inside air and heat
- N_{ch} is number of animals
- U_A is coefficient of heat convection
- λ is Latent heat



Figure 1 The poultry house in this study (a case study of a chicken farm in Khon Kaen, Thailand)

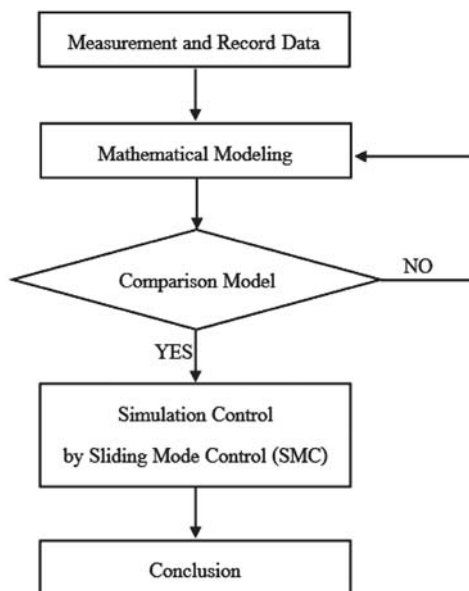


Figure 2 Research methodology

2. Mathematical modeling of an evaporative air-conditioning system

In this section, the theory of energy and mass balance in air and water is used to construct a set of equations. The equations can explain the relation of temperature and moisture content inside any control volume in terms of ordinary differential equations. In this form, further control law design can be conveniently made. However, before we get to those equations, the fundamental direct evaporative cooling process has to be considered first. Figure 3 shows the direct evaporative cooling process with constant enthalpy because of the adiabatic process assumption.

With the evaporative process, ambient air is cooled and moisture added from direct contact with water in the cooling pad simultaneously. The process is shown in the Psychometric chart from state 1 to 2 with the hypothesis that there is no loss of heat to the environment during the adiabatic process. In the other words, some sensible heat from the outside air is transformed to be latent heat for evaporation as show in Figure 4. The cooled supply air absorbs energy to increase its temperature from state 2 to 3.

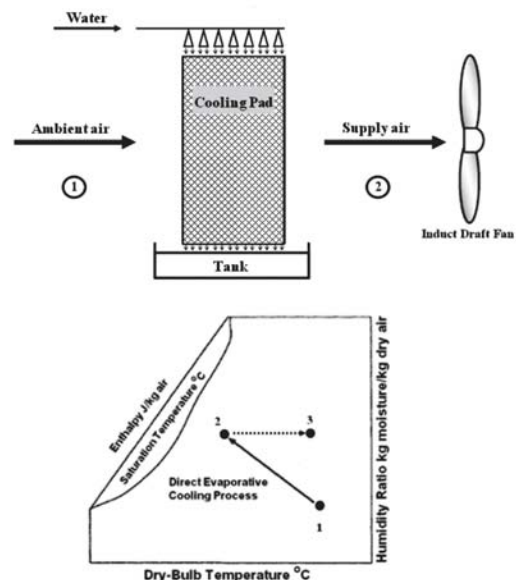


Figure 3 A direct evaporative cooling system and the Psychometric chart illustrating the constant enthalpy Process

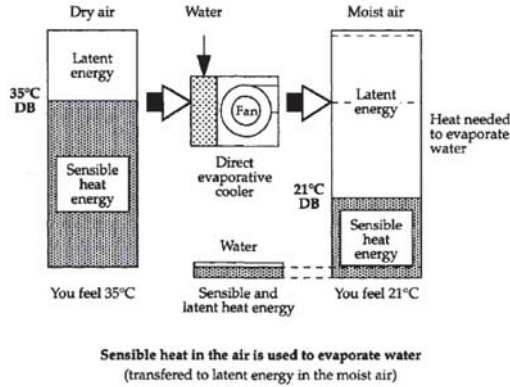


Figure 4 Simplified Evaporative Air-Conditioning Process [3]

In order to find the relationship between the inside air conditions and the evaporative cooling process, a block diagram of an air conditioning system is illustrated in Figure 6. Almost saturated air will occur when the air from state 1 passes the cooling pad into state 2. If the cooling pad effectiveness is 100%, state 2 can have perfect saturation with relative humidity of 100%. However it is not true in practice. Finally from State 2 to State 3, heat in the process will cause decrement of relative humidity. The cooled air state 2 (assume no moisture absorption in the poultry house) is mixed with the air in the poultry house. Then the mixture will reach the required temperature and be removed from the poultry house eventually.

2.1 Model for feed forward control

To meet the desired conditions the cooled supply air is sufficient against the amount of cooling load. To achieve this goal, the cooling load has to be calculated first. Assume that both dry air and water vapor are perfect gases.

Then the mass flow rate of the air (\dot{m}_a) can be determined by the cooling load as follows.[3]

Cooling load = Mass Flow rate x [Enthalpy of state(3)-Enthalpy of state(2)]

$$\begin{aligned}\dot{Q}_{latent} + \dot{Q}_{sensible} &= \dot{m}_a C_{pu} [T_3 - T_2] \\ \dot{Q}_{Total} &= \dot{m}_a C_{pu} [T_3 - T_2]\end{aligned}\quad (1)$$

where

$C_{pu} = C_{pa} + \omega C_{pv}$ is the specific heat of wet air.

As shown in Eq.(1), given the cooling load \dot{Q}_{total} and air mass flow rate, the final temperature inside the control volume T_3 can be evaluated by

$$T_3 = \frac{\dot{Q}_{Total}}{C_{pu} \dot{m}_a} + T_2 \quad (2)$$

From Eq. (2), we can consider the terms $1/\dot{m}_a$ and T_2 as input to the system. Consider the relation for Cooling pad efficiency (ϵ) as in Eqs (3) and (4). If $(T_1 - T_{wb1})$ is the wet bulb depression,

$$\epsilon = \frac{T_1 - T_2}{T_1 - T_{wb1}} \quad (3)$$

or

$$T_2 = T_1 - \epsilon [T_1 - T_{wb1}] \quad (4)$$

The pad efficiency or cooling effectiveness (ϵ) relates to speed of the air passing through the cooling pad in the form of an exponential decay as shown in Figure 5 and Eq (5).

From the work of Camargo *et al* [4]. the effectiveness can be considered from

$$\epsilon = 1 - e^{\left[\frac{-h_c A}{\dot{m}_a C_{pu}} \right]} \quad (5)$$

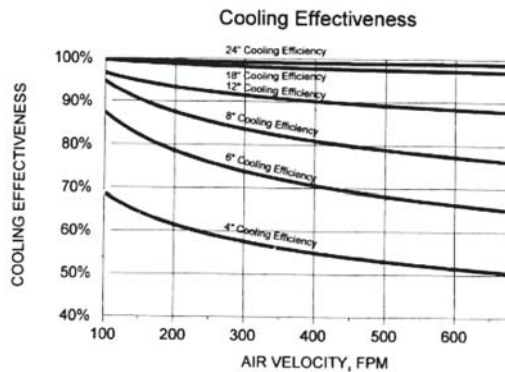


Figure 5 Characteristic curves for effectiveness [8]

where

h_c is coefficient of heat convection of Cooling pad ($\text{W/m}^2 \cdot ^\circ\text{C}$)

A is heat exchanger evaporation area of cooling pad (m^2)

C_p is specific heat of air ($\text{J/kg} \cdot \text{K}$)

\dot{m}_a is air mass flow rate (kg/s)

In the above equation, with constant air mass flow rate (\dot{m}_a) the effectiveness (\mathcal{E}) can be found. From considering Eqs. (2), (4) and (5), the equation that has the inside temperature (T_3) as response and air mass flow rate (\dot{m}_a) as control signal is formed to be Eq. (6).

$$T_3 = \frac{\dot{Q}_{Total}}{C_{pu}} \frac{1}{\dot{m}_a} + T_{wb1} + [T_1 - T_{wb1}] e^{\left[\frac{-h_c A}{\dot{m}_a C_{pu}} \right]} \quad (6)$$

2.2 Model for feedback control

From Eq. (6), the inside temperature (T_3) can be found from the heat load (\dot{Q}_{Total}). However, the heat load varies all the time. So, this equation can be used for feed forward control only. It cannot respond to the changing environment. However, the heat load can be updated according to the outside temperature. It is still

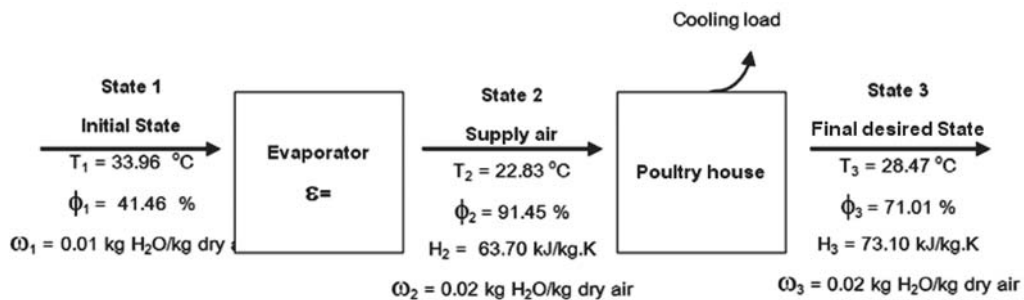


Figure 6 Block diagram of the ideal steady state evaporative cooling process.

(Where T is Dry-bulb Temperature, ϕ is Relative Humidity, H is *Enthalpy* and ω is humidity ratio)

difficult to enforce the system because there is no compensation in model errors of the heat load (4). In order to design a feedback control system, the balance of mass and energy has to be considered as follows.

Energy balance

Inside energy changes

$$= \text{Energy input} - \text{Energy output}$$

Mass balance

Inside humidity changes

$$= \text{Water input} - \text{Water output}$$

These relations can be expressed as

$$\rho_{air} C_p V_T \frac{dT_{in}}{dt} = Q_{ch} + Q_c - Q_v - Q_{ev} \quad (7)$$

$$\rho_{air} V_T \frac{d\omega_{in}}{dt} = w_{ch} + w_{air out} + w_{ev} - w_{air in} \quad (8)$$

where

ρ_{air} is air density

C_p is specific heat of air

V_T is inside air (m^3)

T_{in} is inside temperature ($^{\circ}C$)

t is time in second

Q_{ch} is sensible and latent heat from chicken in kJ/s

Q_c is heat load from the ceiling and wall into the poultry house in kW

Q_v is heat loss from ventilation in kW

Q_{ev} is heat used to evaporate water in kW

ω_{in} is humidity ratio in the poultry house in $\frac{g_{moisture}}{kg_{dry air}}$

w_{ch} is humidity ratio from chicken in $\frac{g_{moisture}}{s}$

$w_{air out}$ is humidity ratio from ambient air into the poultry house in $\frac{g_{moisture}}{s}$

w_{ev} is humidity ratio from evaporated water in $\frac{g_{moisture}}{s}$

$w_{air in}$ is humidity ratio out the poultry house in $\frac{g_{moisture}}{s}$

From Equations (7) and (8), each term on their right hand side can be supplied as in the following sections.

(1) Sensible heat (Q_{ch}) and Humidity (w_{ch}) from chicken

From data from the case study, one chicken generates heat and water of 10 Btu/hr and 105 cc./day, respectively, these data have been considered by an expert. In this work, both parameters follow the equation in the work of Daskalov(6) which used the case of 20 kg piglets. This is because there is no equation for chickens expressed as a function of sensible heat and ambient temperature. Rough estimation in the model can be done due to the ability of Sliding Mode Control (SMC) that can compensate for uncertainty in that model. Therefore, with 2kg per chicken, 60000 chickens is equivalent to 6000 piglets in Daskalov's equation. However, the equation was not designed for poultry but the objective of this work is to give guidelines for automatic control system design. However, if there is a relationship for the chicken, it can be substituted into the control model easily.

$$Q_{ch} = N_{ch} \times 0.096 \times [0.8 - 1.85 \times 10^{-7}(T_{in} + 10)^4] \quad (9)$$

$$w_{ch} = N_{ch} \times 0.001 \times [0.26T_{in}^2 - 6.465T_{in} + 81.6] \quad (10)$$

where

N_{ch} is number of animals

Q_{ch} is sensible heat from the animals (kW)

T_{in} is inside air temperature ($^{\circ}\text{C}$)

w_{ch} is humidity ratio production from the animals (kg/h)

(2) Heat load from the Ceiling and Wall into the poultry house (Q_c)

Normally, the heat conduction through the building can be expressed as a linear function of temperature difference between inside and outside.

$$Q_c = U_A [T_{out} - T_{in}] \quad (11)$$

where

U_A is coefficient of heat convection of ceiling and wall (kW/K)

T_{out} is outside air temperature ($^{\circ}\text{C}$)

(3) Heat loss from ventilation (Q_v)

This loss not only depends on the temperature difference, it also varies according to the ventilation rate as expressed in Eq. (12)

$$Q_v = \rho_{air} V_R C_p [T_{out} - T_{in}] \quad (12)$$

where

V_R is Volume of air flow rate (m^3/s)

(4) Heat used to evaporate water (Q_{ev})

The heat used to evaporate water can be obtained easily by multiplication of the latent heat of vaporization by the amount of evaporated water.

$$Q_{ev} = \lambda w_{ev} \quad (13)$$

where

λ is Latent heat

(5) Value of humidity ratio from ambient air coming to the poultry house ($w_{air out}$)

$$w_{air out} = \rho_{air} V_R \omega_{out} \quad (14)$$

(6) Value of humidity ratio flow out of the poultry house ($w_{air in}$)

$$w_{air in} = \rho_{air} V_R \omega_{in} \quad (15)$$

where

$w_{air out}$, $w_{air in}$ are external and inside absolute humidity in $\text{g}_{\text{H}_2\text{O}}/\text{kg}_{\text{dryair}}$ respectively.

Substituting equations (9) - (15) into equations (7) and (8) yields

$$\frac{dT_{in}}{dt} = \frac{N_{ch} \times 0.096 \times \left[0.8 - 1.85 \times 10^{-7} (T_{in} + 10)^4 \right] + U_A [T_{out} - T_{in}] - \lambda w_{ev}}{\rho_{air} C_p V_T} - \frac{V_R}{V_T} [T_{out} - T_{in}] \quad (16)$$

$$\frac{d\omega_{in}}{dt} = \frac{N_{ch} \times 0.001 \times \left[0.26 T_{in}^2 - 6.465 T_{in} + 81.6 \right] + w_{ev}}{\rho_{air} V_H} - \frac{V_R}{V_T} [w_{in} - w_{out}] \quad (17)$$

Then substituting the plant parameters from Table 1 into Eq. (16) and (17) leads to

$$\begin{aligned} \frac{dT_{in}}{dt} = & \left[77.76 - 1.801 \times 10^{-5} (T_{in} + 10)^4 \right] \times 10^{-3} + 0.12036 \times 10^{-3} [T_{out} - T_{in}] - \\ & 0.1059 \times 10^{(-3)} w_{ev} - \frac{[T_{out} - T_{in}] V_R}{4900} \end{aligned} \quad (18)$$

$$\frac{d\omega_{in}}{dt} = \frac{\left[0.265 T_{in}^2 - 6.5918 T_{in} + 83.26 \right] \times 10^{-3}}{3600} + 4.72 \times 10^{-8} w_{ev} - \frac{V_R}{4900} [w_{in} - w_{out}] \quad (19)$$

The above set of equations can be simplified to be multi input/output (M/M_o). That is

$$\begin{bmatrix} T_{in} \\ w_{in} \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 \end{bmatrix} + \begin{bmatrix} -b_1 & -b_2 \\ b_3 & -b_4 \end{bmatrix} \begin{bmatrix} w_{ev} \\ V_R \end{bmatrix} \quad (20)$$

Where $f_1 = \left[77.76 - 1.801 \times 10^{-5} (T_{in} + 10)^4 \right] \times 10^{-3} + 0.12036 \times 10^{-3} [T_{out} - T_{in}]$

$$f_2 = \frac{\left[0.265 T_{in}^2 - 6.5918 T_{in} + 83.26 \right] \times 10^{-3}}{3600}$$

$$b_1 = 0.1059 \times 10^{-3}$$

$$b_2 = \frac{[T_{out} - T_{in}]}{4900}$$

$$b_3 = 4.72 \times 10^{-8}$$

$$b_4 = \frac{[w_{in} - w_{out}]}{4900}$$

One way to control the inside temperature (T_{in}) and humidity (w_{in}) is to use some nonlinear control design to compensate the system for nonlinearity and uncertainty. For this work the well known sliding mode control technique with Lyapunov stability analysis has been used to design a control law in section 4. Before going to that section, validation of the mathematical model has to be done.

3. Model validation

To ensure correction of the model, response comparison of the real plant and the model was conducted by using the same actual input signal as illustrated in Figure 7. Collected data in the graph shows increment of air flow and switching operation of the water pump. These control input signals have been tested in the real plant and

fed forward to the mathematical model in terms of Volume of air flow rate (V_R) and water flow rate (w_{ev}).

By using the control input signal in Figure 7, the air flow rate is all the way from about 32 m³/s up to 104 m³/s while the under flow with 15 m³/s constant rate is on/off for 6/10 minutes respectively. A comparison of output signals is shown in Figure 8.

From the responses in Figure 8 the weather condition values from the derived mathematical model are quite close to the real measured values. Error between real conditions and model in terms of inside temperature and absolute humidity are 0.5487% and 2.9417 % respectively. These errors are acceptable in order to design a control law based on robust control technique. Such a technique will be presented in the next section.

4. Sliding mode control and simulation results

Due to limitations of the controller in the farm (using an open loop controller) and control stability concerns for production, a closed loop control experiment was not set up in this work. However, with the acceptable accuracy of the model, the simulation results of the SMC are presented here.

In the SMC technique, the sliding function has to be defined first. The errors between response and desired condition of the inside temperature and humidity content are given by:

Sliding function for temperature:

$$S_T = T_{in} - T_{ind} \quad (21a)$$

Sliding function for humidity:

$$S_w = w_{in} - w_{ind} \quad (21b)$$

where

T_{ind} is the designed temperature

w_{ind} is the designed humidity content

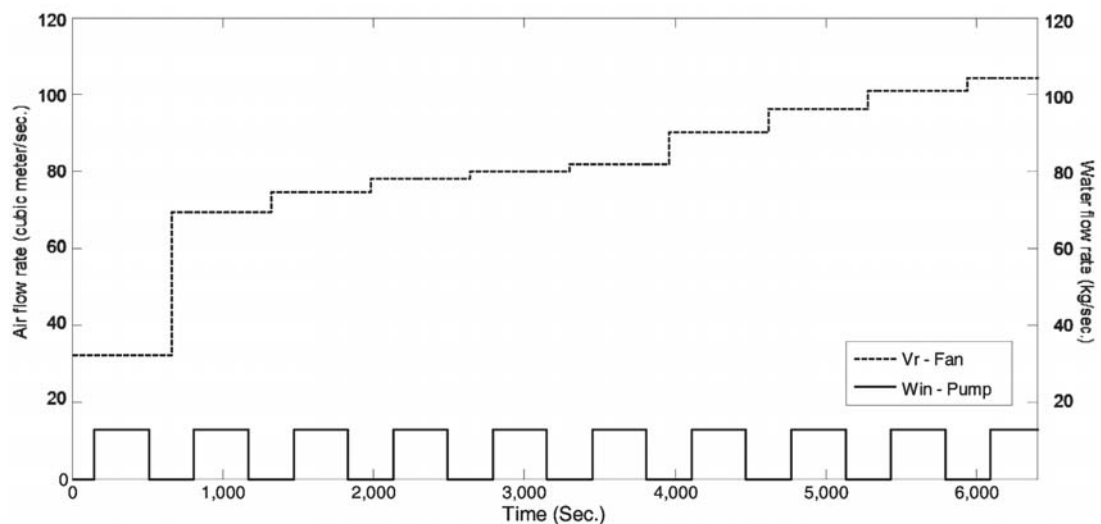


Figure 7 Air flow rate (m³/s) and water flow rate (kg/s)

Although the sliding function has been constructed, the control law cannot be derived yet because of the coupling condition of the control distribution matrix in equation (20) (2nd term of its RHS). Therefore, the decoupling process will be done here. The assumption is that all arguments in that matrix are known and its determinant is not zero ($\det B \neq 0$). Note that if the values are not known exactly, those imperfections still can be compensated for by the nature of the sliding mode control. Let

$$B = \begin{bmatrix} -b_1 & -b_2 \\ b_3 & -b_4 \end{bmatrix} \quad (22)$$

Then the control signal matrix of eq. (20) can be

$$\begin{bmatrix} w_{ev} \\ V_R \end{bmatrix} = B^{-1} \begin{bmatrix} U_T \\ U_w \end{bmatrix} \triangleq \begin{bmatrix} -b_1 & -b_2 \\ b_3 & -b_4 \end{bmatrix}^{-1} \begin{bmatrix} U_T \\ U_w \end{bmatrix} \quad (23)$$

By using the Lyapunov direct method, the Lyapunov function candidate can be written as

$$V = \frac{1}{2} [s_T^2 + s_w^2] + \frac{1}{2} [\tilde{R}_T^2 + \tilde{R}_w^2] \quad (24)$$

where

$\hat{\cdot}$ is any estimated value

\tilde{R}_T is an estimation error of the residual error that corresponding to a bound (M_T) of a nonlinear form $\tilde{f}_1 = f_1 - \hat{f}_1$ ($\tilde{R}_T = R_T - \hat{R}_T, R_T = |\tilde{f}_1| < M_T$)

\tilde{R}_w is an estimation error of the residual error that corresponding to a bound (M_w) of a nonlinear form $\tilde{f}_2 = f_2 - \hat{f}_2$ ($\tilde{R}_w = R_w - \hat{R}_w, R_w = |\tilde{f}_2| < M_w$)

and its 1st derivative is

$$\dot{V} = s_T \dot{s}_T + s_w \dot{s}_w + \tilde{R}_T [\dot{R}_T - \dot{\hat{R}}_T] + \tilde{R}_w [\dot{R}_w - \dot{\hat{R}}_w] \quad (25)$$

It has been noted here that uncertainties in the system are assumed to be bound. So there must exist bound M_T & M_w in the system. And because set point control of T_{ind} and w_{ind} will be applied, the residual uncertainty of R_T, R_w can be a slow variation value. Therefore, Equation (25) is simplified to be

$$\dot{V} = s_T [f_1 + U_T] + s_w [f_2 + U_w] - \tilde{R}_T \dot{\hat{R}}_T - \tilde{R}_w \dot{\hat{R}}_w \quad (26)$$

In order to satisfy the Lyapunov stability condition ($\dot{V} < 0$), the control law in this work is chosen to be

$$\left. \begin{aligned} U_T &= -\hat{f}_1 - c_T s_T - \hat{R}_T \\ U_w &= -\hat{f}_2 - c_w s_w - \hat{R}_w \end{aligned} \right\} \quad (27)$$

where

c_T & c_w are any positive definite values that can be used to adjust concentration of convergence. Their magnitude can be chosen as M_T and M_w , respectively. Then the residual error estimation is

$$\begin{aligned} \hat{R}_T &= \int s_T dt \\ \hat{R}_w &= \int s_w dt \end{aligned}$$

Substitute the control law in Eq. (27) into the system Eq. (26), its error dynamics can be expressed by

$$\begin{aligned} \dot{V} &\leq -c_T s_T^2 - c_w s_w^2 + \tilde{R}_T s_T + \tilde{R}_w s_w - \tilde{R}_T \dot{\hat{R}}_T - \tilde{R}_w \dot{\hat{R}}_w \\ \dot{V} &\leq -c_T s_T^2 - c_w s_w^2 \end{aligned} \quad (28)$$

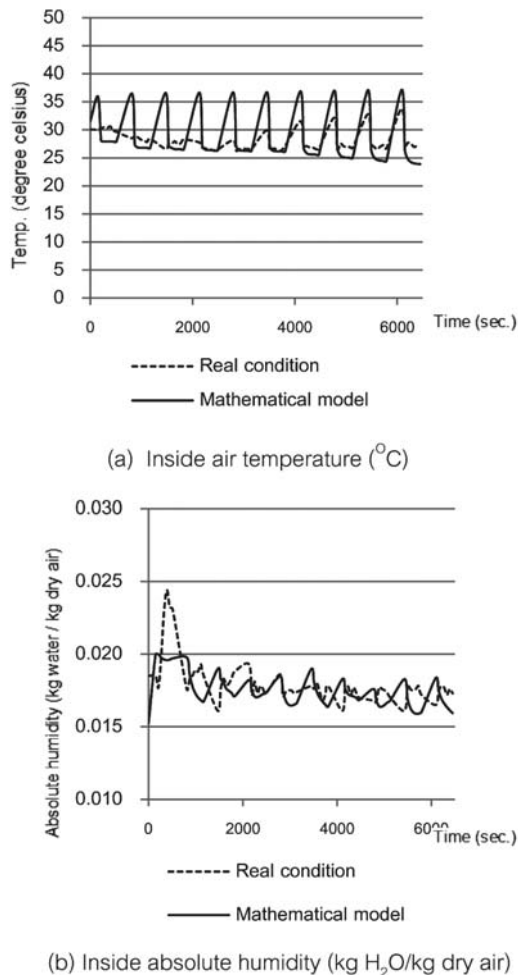


Figure 8 Comparison of the climate in the house between mathematical models and real conditions.

It has been seen that the control system is stable in the Lyapunov sense with the control law in Eq. (27). The control system components can be seen in Figure 9.

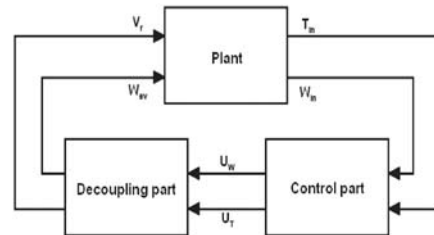


Figure 9 Block diagram of the closed Loop Control System in the Matlab simulation.

Performance of the control law in Eq. (27) will be shown in this section by simulation. Parameters for the model in Eq. (20) are addressed as follows. Ambient conditions are temperature of 33 °C, 40% relative humidity and moisture content of 12.5 g_{H₂O}/kg_{dry air}. The model has two saturation portions caused by real plant conditions. That is, ranges of air exchange and water flow rate are between 35–175 m³/s and 0–13 kg/s, respectively. Uncertainties are also added in the forms of outside temperature and moisture by using $\Delta T_{\text{out-in}}$ is $1 \sin \frac{2\pi}{900} t$ and $\Delta W_{\text{in-out}}$ is $1 \times 10^{-3} \sin \frac{2\pi}{900} t$. It has been noticed

that arbitrary uncertainty value can be added into the system as long as the value still bounds and does not make actuators in the model become saturated. The inside desired conditions in this work are 27 °C, 70% relative humidity and 16.8 g_{H₂O}/kg_{dry air} of moisture.

The simulation results are shown in three cases as follows.

Case 1 feed forward control,

Case 2 no variation of outside conditions,

Case 3 sinusoidal variation of outside conditions.

4.1 Case 1 feed forward control.

Results of the feed forward control have already been illustrated in Figure 8 of section 3. Normally, to meet the design conditions with this technique, an operator must have enough experience to set the actuator as in Figure 7. Although the set of control signals may be recorded and then used by a predefined controller, it is still unusable for ambient changing. As you can see in Figure 10, by using the same control signal as Figure 7, the response cannot track the same response (real condition) if there exists uncertainties in the outside air.

From Figure 10, it is seen that closed loop control with uncertainties compensation is necessary for the air condition control in the poultry house.

4.2 Case 2 no variation of outside condition

Simulation results of the control system of this case are shown as Figure 11 and Figure 12. It is not surprising that the temperature and moisture responses in Figure 11 track their set point very well. Small steady state errors in moisture are present because integral action of the control law did not include it and from mismatch between the plant and control law parameter. Figure 12 shows the corresponding control input of Figure 11. The inputs are

flattening out after about 400 seconds due to its control response having reached the reference value.

4.3 Case 3 sinusoidal variation of outside condition

In this case, the uncertainty is given as a sinusoidal function as stated earlier. The results of the simulation for this case are shown in Figure 13 and Figure 14.

From the response in Fig.13, there is slight fluctuation after a time of about 200 sec. This is because of variation of outside conditions. For a transient period (before 200 sec.), there is a spike of both responses and saturation of the control input signal. This transient behavior occurs due to the added uncertainties making the larger input signal in the first period compared to the second. The input thus becomes saturated. However, the adaptation law that has been operating since the beginning of the control action will give suitable input signals after the temperature and moisture reach the set point. Therefore the transient spike responses disappear eventually. This case shows that the controller can overcome the uncertainties and nonlinearities in the plant and make inside air conditions comfortable for chickens according to the operating designing conditions.

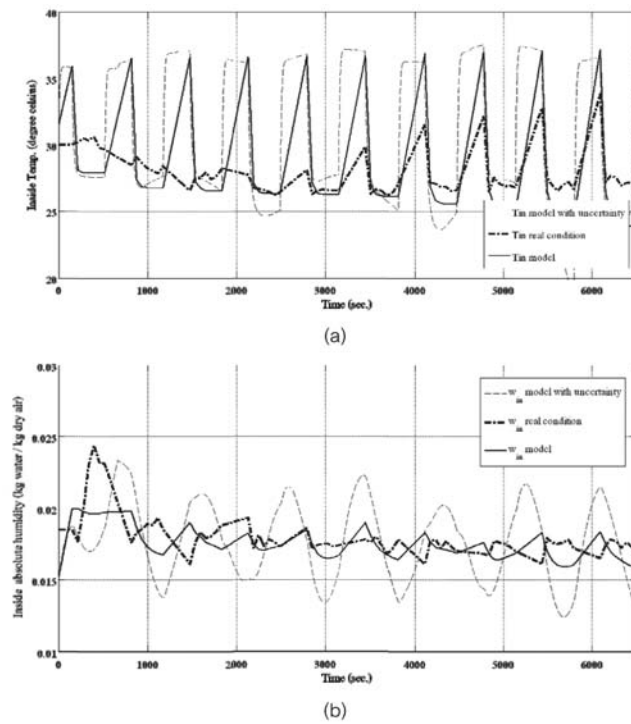


Figure 10 (a) The response of inside temperature (T_{in}) and (b) The response of inside absolute humidity (w_{in}) of case 1.

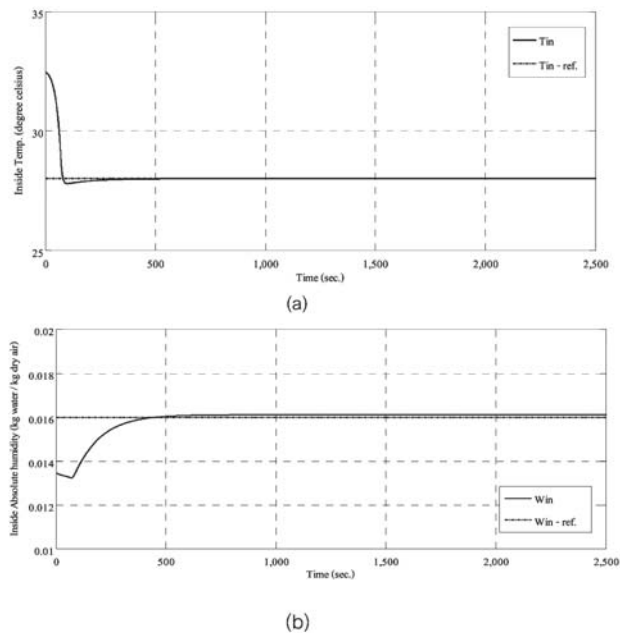


Figure 11 (a) The response of inside temperature (T_{in}) and (b) The response of inside absolute humidity (w_{in}) of case 2.

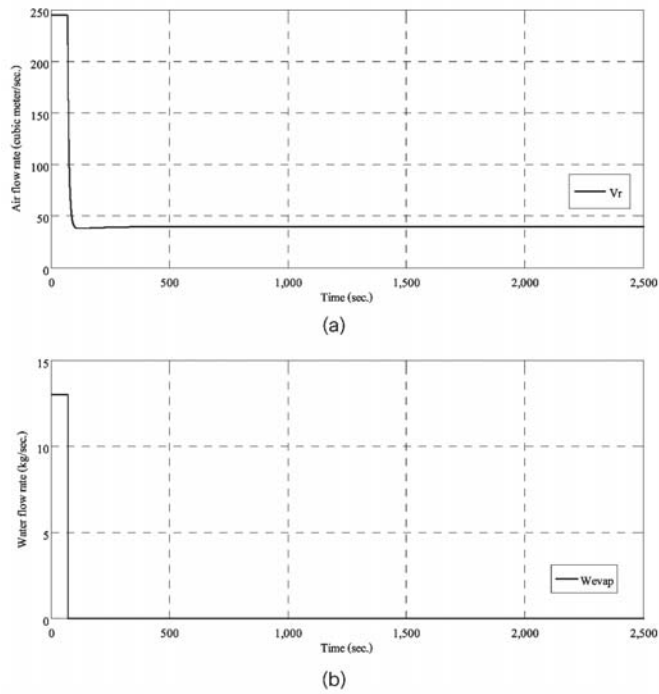


Figure 12 The control signal (a) V_R and (b) w_{ev} of case 2.

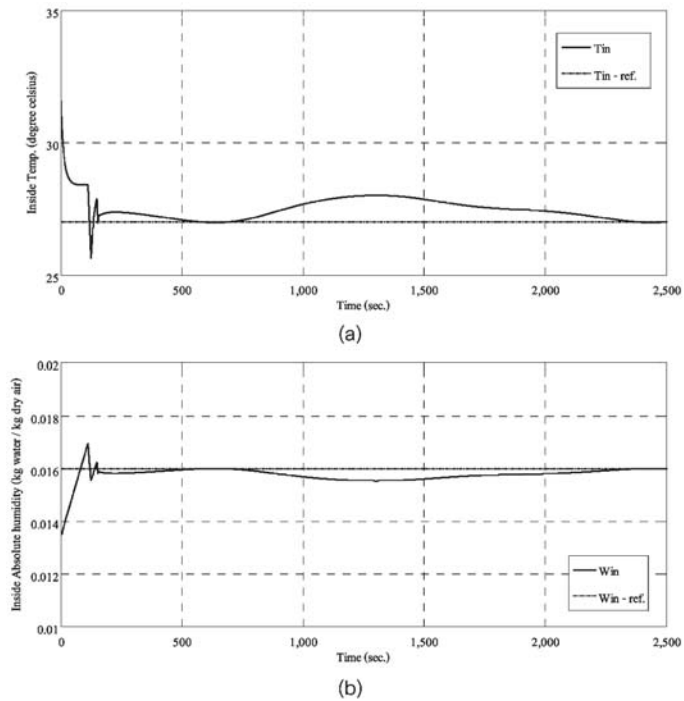


Figure 13 (a) The response of inside temperature (T_{in}) and (b) The response of inside absolute humidity (w_{in}) of case 3.

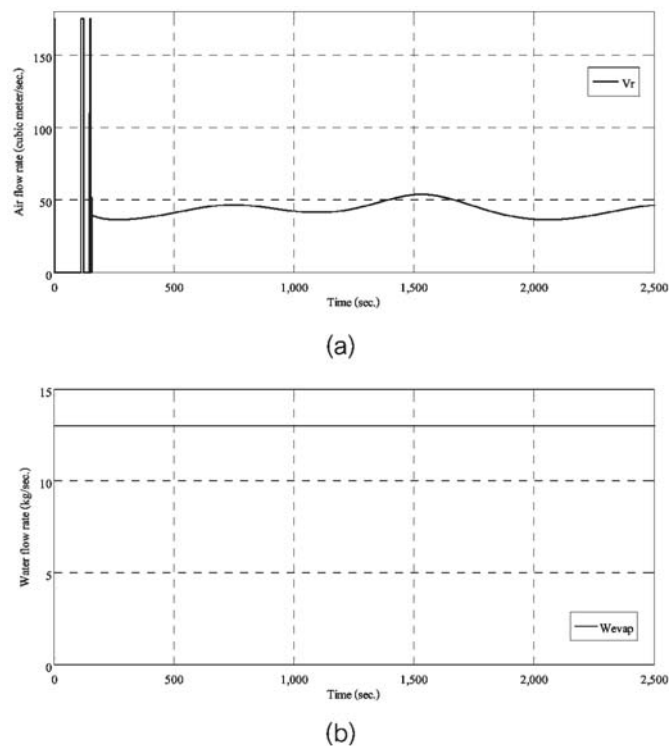


Figure 14 The control signal (a) V_R and (b) w_{ev} v of case 3

5. Conclusion

In this work, the mathematical models of air conditioning in a closed type poultry house are presented. For close loop control, the energy and mass of water balance was used to derive the mathematical model. The constructed model was verified using the experimental input/output data. The errors of output between simulation and collected data are 0.5% and 2.9% for temperature and moisture, respectively. Therefore this acceptable model could be used for controller design. Due to the presence of nonlinearity, uncertainty and coupling structure, the SMC and adaptation law with simple decoupling technique was applied to force the

complex system to reach the desired condition. It has been noted here that the control volume in this work is an active mixing volume at about 60% of total volume. For larger size of poultry house that has much deviation of temperature and humidity, one may divide it into small volumes before applying this technique. The simulation results show that the SMC can force the system to the set point against variation of ambient conditions.

6. Acknowledgement

This research was supported by Cleaner Technology Internship Program, Khon Kaen University Node and National Metal and Materials

7. References

- [1] Rattanathanaopat A. Improving Efficiency of Evaporative Cooling in Poultry, Department of Energy Technology, King Mongkut's University of Technology Thonburi, Bangkok. 2546. (In Thai)
- [2] Barmish B. R., Leitmann G. On ultimate boundedness control of uncertain systems in the absence of matching assumption. *IEEE Transactions on Automatic Control*. 1982;27(1):153-158.
- [3] Bom Gret Jan et al, Evaporative Air-Conditioning, Washington, D.C., U.S.A, The World Bank. 1999
- [4] Camargo J R, et al. Experimental Performance of a direct evaporative cooler operating during summer in a Brazilian city. *Intl. J. Refrig*. 2005 ;28(7): 1124-1132.
- [5] Corless M J, Leitmann G. Continuous state feedback guaranteeing uniform ultimate boundedness for uncertain dynamic systems. *IEEE Transactions on Automatic Control*. 1981;26(5):1139-1144.
- [6] Daskalov, et al. Non-linear Adaptive Temperature and Humidity Control in Animal Buildings *Bio systems Engineering* 2005;93 :1-24
- [7] Giabaklou Z, Ballinger J A. A Passive Evaporative Cooling System by Natural Ventilation. *Building and Environment* 1996;31(6):503-507.
- [8] João M, Pimenta D, Wagner P. De Castro Analysis of Different Applications of Evaporative Cooling Systems, Department of Mechanical Engineering, University of Brasilia. 2005.
- [9] Lawrence S A, Tiwari G N, Performance Study of an Evaporative Cooling System for a Typical House in Port Moresby. *Solar & Wind Technology*. 1989;6(6):717-724.
- [10] Mills A F, Basic Heat & Mass Transfer, California, Prentice Hall, 1999;274-276, 842-844.
- [11] Roengruen P et al, A Study Of an Automatic Control System for a Poultry House. Department of Instrumentation Engineering, Faculty of Engineering, King Mongkut's Institute of Technology Ladkrabang Bangkok. (In Thai)
- [12] Neamtang P. Reduction of Heat Loads Through Poultry Roof, Department of Energy Technology, King Mongkut's University of Technology Thonburi, Bangkok. 2546. (In Thai)
- [13] Radpukdee T, Jirawattana P. Uncertainty learning and compensation: an application to pressure tracking of an electro-hydraulic proportional relief valve. *Control Engineering Practice*. 2009;17:291-301
- [14] Simmons J D, Lott B D, Evaporative Cooling Performance Resulting From Changes In Water Temperature. *Applied Engineering in Agriculture*. 1996;12(4):497-500.

- [15] Slotine J J E, Li W. Applied Nonlinear Control. New Jersey: Prince-Hall International Inc. USA. 1991
- [16] Sodha M S et al. Thermal Performance of a Room Coupled to an Evaporative Cooling Tower. Proceedings of The Biennial Congress of The International Solar Energy Society. 1991 August 19-23, Colorado, USA. 1991;3: Part 3 p. 3095-3100.
- [17] Yunus A C, Michael A B, Thermodynamics: An Engineering Approach. Singapore 1989.