Feasibility of hybrid renewable heating system application in poultry house: a case study of East Midlands, UK

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Abstract

Currently, there has been an obvious lack of innovation within poultry houses heating, ventilation and air conditioning (HVAC) system design that deals with both energy efficiency and poultry welfare issues. This paper presents an innovative and renewable heating system for poultry house application to meet the welfare living environment requirement for breeding, to improve the energy efficiency of HVAC system and to decrease fossil fuel consumption and harmful gas emissions. The purpose of this study is to design, develop and test the highly efficient hybrid heating system via an integrated photovoltaic/thermal array with polyethylene heat exchanger coupled to geothermal heat pump system based on the East Midlands’ climate conditions in the UK. The numerical model is established based on finite volume method and solved by using Engineering Equation Solver, and a good agreement with less than 15% difference between the numerical and experimental results is achieved. The results indicate that the annual electrical and thermal output are 11867 kWh and 30245 kWh, respectively, which not only could fulfil the poultry house electrical need, but also can provide \( \sim 43.5\% \) electricity demand of the heat pump compressor operating.

Keywords: poultry house; renewable heating system; PV/T with PHE; geothermal heat pump; energy efficiency;

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1. INTRODUCTION

Global-scale warming and climate change are unequivocal over the past century and are continuing under the continued emissions of greenhouse gases (GHGs). They have exerted adverse issues on human society, animals and plants, economy and ecosystems all over the world, which are mostly associated to changes in climate extremes [1]. The growing atmospheric concentration of carbon dioxide (CO\(_2\)) is widely known as the largest contributor of global warming. Energy saving has become more than significant nowadays due to shortage of energy reserves, considerably soaring energy prices and growing significance of environmental issues [2]. With the growth in awareness of the governments and individuals in UK [3], it has been accepted that conventional method based on fossil fuels has exacted a heavy cost on agriculture [4, 5], and thus they have started looking for new solutions to decrease CO\(_2\) emission and investigate novel renewable energy technologies in these fields, such as poultry houses [6–8], greenhouses [9–12] and pig houses [13–15]. Animal production is responsible for GHG emission reaching 22% of global total emission of which nearly 80% comes from poultry industries involving transport of livestock and feed [16, 17]. Poultry industry is one of the energy-intensive industries that consume large quantities of fuel in the UK, especially for the broilers sector. Broiler poultry houses heavily consume fuel—gas for their heating systems in order to maintain the desired temperatures for the breeding of the chicks. At present, conventional electricity and gas are the main sources for heating in poultry house. During the period from 2010 to 2017, poultry growers experienced the change in the running costs due to the rise in fuel bills. Energy produced from fuels has induced raised GHG emission in the atmosphere, which in turn gives rise to the growth of environmental contamination.
As a result, renewable energy sources are considered to apply in the poultry production field, where solar (55%), biomass (27%), geothermal (13%) and wind (5%) are commonly being considered as presented in Figure 1 [18–20]. Recently, many studies and applications, mostly on solar energy [7, 21–23] and geothermal energy [24–26], are proposed and provide a simple, sustainable, effective strategy for poultry houses. In poultry houses, heating is crucial importance in the first week for the growth of the chicken before they become able to adjust their own body temperature. Solar energy technology is an ideal solution for heating a poultry house as it is both efficient and cheaper to run than conventional energy sources like oil and liquefied petroleum gas. The cost of heating a poultry house by traditional method is becoming ever more expensive as the boilers and fuels that are normally used are less efficient. A solar energy heating system that is integrated into a roof could meet up to 80% of birds’ heating and energy demand. Mirzae-Ghaleh et al. [21] proposed a novel hybrid system that is composed of solar hot water system, auxiliary heater and heat exchanger and discovered that solar radiation can provide at least 20% of the required energy in winter. Chen and Sheng [22] illustrated a commercial solar hot water system with high-efficiency vacuum tube collectors and demonstrated that this system is more effective than the traditional tungsten lighter unit; meanwhile, it is found that the system could save ∼148.6 kg of CO₂ emissions for a thousand chicks. Fawaza et al. [23] studied the efficiency and performance of a solar-assisted localized heating and ventilation system for chicken brooding based on the computational fluid dynamic software. Their results indicate that this system could save 74% of the energy needed and cover 84% of the required load in the winter flock. Kapica et al. [24] presented a numerical model of a hybrid solar-wind system to investigate the CO₂ emission based on the Matlab/Simulink software for a poultry house and found that larger systems provide higher CO₂ but the energy utilization ratio reduces.

Geothermal energy is a potential heat source for the economic heating of poultry houses with optimum production performance. Because soil has a comparatively constant temperature, heat exchange between soil and poultry house is more efficient than that of traditional solutions. Ground source heat pump (GSHP) systems are environmentally friendly, require comparatively little power, help maintain a desired temperature within poultry house and decrease fossil fuel requirement and CO₂ emissions. Furthermore, GSHP system requires minimal maintenance over their long lifetimes and encapsulation of the heat exchange tubes in concrete prevents accidental release of coolants. Therefore, some case studies [25, 26] are investigated and conducted to estimate the impact of a heating system using a GSHP on production performance and housing environment of poultry houses. Choi et al. [25] developed a GSHP system applied in a commercial broiler house for heating space in Korea and found that the ranges of maximum temperature can be kept from 26.4 to 33.5°C, whereas the minimum is in the range from 22.4 to 30.9°C. Kharseh and Nordell [26] proposed a hybrid solar collector and GSHP system in order to investigate the heating and cooling requirements for a chicken farm in Syria and confirmed that this system could produce 92 MWh for heating and 13 MWh for cooling, which could fulfill the poultry house demands. Sustainable poultry house is an alternative for solving fundamental and applied issues related to food production in an ecological way. This involves design and management procedures that work with natural processes to conserve all resources and minimize waste and environmental damage as well as maintaining or improving poultry house profitability. The innovative hybrid renewable heating system has been installed in a real poultry shed, and the experiment data has been collected. Therefore, this paper focuses on developing the numerical model for each component of the heating system. A combining photovoltaic/thermal (PV/T) module with polyethylene heat exchanger (PHE) loop coupled to geothermal heat pump enable provide electricity and heat output for poultry house in the UK. There are few studies on designing a PV/T integrated with a geothermal heat pump system and employing in poultry house. Currently, there is a research gap to assess the system performance for poultry house over a year. Hence, the objective of this paper is to fill the above research gap, which is used for predicting the annual system thermal and electrical output.

2. NUMERICAL MODELS

The work presented in this paper is an Innovate UK Agri-Tech Catalyst funded project named ‘Welfare Enhanced Living Conditions for Healthier Chickens (WelChic)’. A basic design schematic of the ‘WelChic heating system’ is shown in Figure 2. The novel hybrid heating system provides highly efficient heating using a heat pump integrated with a PV/T array employing a new type of inexpensive PHE and a new low-cost ground pipe array. The PV/T array could provide electrical output to drive the operation of heat pump compressor and some of the evaporator thermal input along with the ground copper pipe array to the heat pump. The ground pipe array is applied to supplement the thermal output from the PV/T array, such as at night time or periods of low solar radiation. The condenser output of heat pump is utilized to produce hot...
2.1. PV/T array model

Thermal energy transfer occurred in the ‘WelChic heating system’ has two processes including conversion of solar radiation into thermal energy by PV panels and transporting absorbed thermal energy towards PHEs. This numerical model is established by using Engineering Equation Solver (EES) Software. This model is capable of well-suited for the physical structure of the proposed system. Figure 3 presents the heat transfer process between the PV cell and PHE.

To simplify the model structure, the following assumptions are made during the model set-up process:

1) The system is at a quasi-steady state condition.
2) The surface temperature of all material layers is defined as uniform.
3) The air layer between PV and heat exchanger is assumed stagnant.
4) A 6-hour sunshine day in winter (October–April) and 8 hours in summer (May–September).
5) Water flow in PHE is 5 L/min.
6) Initial PHE water temperature is 5 °C.

The PV/T module consists of the incident model of sunlight, energy conservation equations of glass cover, PV modules, EVA plastic back, PHE and roof support. Therefore, this dynamic mathematical equation is given by

$$\frac{\partial Q_t}{\partial t} = m_{PV/T} C_{PV/T} \frac{\partial T_{PV/T}}{\partial t} = Q_{abs} - Q_{PV/T-loss} - Q_{ele},$$  \hspace{1cm} (1)

where $Q_t$ is the useful heat energy (kW), $m_{PV/T}$ is the mass of PV/T (kg), $C_{PV/T}$ is the heat capacity of PV/T (J/(kg K)), $T_{PV/T}$ is the temperature of PV/T (°C), $t$ is the time (s), $Q_{abs}$ is the solar energy absorbed at the front of PV/T surface (kW), $Q_{PV/T-loss}$ is the total heat loss (kW) and $Q_{ele}$ is the overall electricity output (kW);

$$Q_{abs} = \tau_c \alpha_{abs} A_{eff} I,$$ \hspace{1cm} (2)

where, $\tau_c$ is the PV/T transmittance, $\alpha_{abs}$ is the PV/T absorptivity, $A_{eff}$ is the PV/T effective area (m²) and $I$ is the incident solar radiation (W/m²);

$$Q_{PV/T-loss} = Q_{conv, c} + Q_{c, sky} + Q_{conv, p, heo} + Q_{p, heo},$$ \hspace{1cm} (3)

where $Q_{conv, c}$ is the forced convection between the cover layer and ambient temperature (kW), $Q_{c, sky}$ is the emission power between the cover layer and sky (kW), $Q_{conv, p, heo}$ is the forced convection between the EVA plastic back of a PV model and outer wall of...
Boltzmann's constant, $5.67 \times 10^{-8}$ W m$^{-2}$ K$^{-4}$.

The heat flows through conduction across the PV cover, cell layer and EVA plastic layer in the back of the module. The heat gain of water in the PHE is as equal as the useful heat energy is given as

$$Q_l = A_{\text{eff}} \cdot h_l \cdot (T_a - T_w),$$  

(15)

where $h_l$ is the overall heat transfer coefficient between PV cell and water (W m$^{-2}$ K) and $T_w$ is the water temperature in the PHE ($^\circ$C);

$$\eta_l = \frac{Q_l}{A_{\text{eff}} \cdot T_f},$$  

(16)

where $\eta_l$ is the thermal efficiency (%).

2.2. Geothermal closed loop

To provide sufficient heat source for heat pump system, a novel ground coper pipe array is proposed. It consists of 25 mm diameter of vertical copper tubes sunk into the ground to a depth of 5–6 m and water/glycol solution as working fluid. One of the most promising aspects of the ground pipe array system describes its low-cost, fast and easy installation. In order to develop the heat transfer model, the simplified assumptions in this study are made as follows:

1) The ground is regarded as a homogeneous medium with the mean thermal physical properties.
2) The initial soil temperature is assumed as a function of depth.
3) Heat transfer in the solid region is regarded as pure heat conduction and the effect of groundwater flow is negligible.
4) A profile of velocity in copper pipe is uniform.

Ground surface temperature would be directly affected by the solar radiation when the ground surface is fully exposed to outdoor environment. However, when the ground surface is covered by a building, all the radiations from nearby buildings and sky would be sheltered resulting in a low temperature fluctuation near the ground surface. Therefore, in this study, the ambient temperature is just used to represent the ground surface temperature.

In terms of the working fluid flow region, energy formulations of the inlet and outlet pipes need to be setup separately because of the flow directions, and governing equations in the solid and fluid regions are also established based on reference [27].

2.2.1. Energy balance in solid region

For heat transfer analysis, the geothermal closed loop is classified into two regions: solid region and fluid region. The solid region includes soil and copper pipe, where heat transfer is regarded as three-dimensional (3D) transient heat conduction. The soil is divided into 100 layers in the vertical direction in order to interpret the effect of fluid temperature variation. Therefore, the

$$h_{cv} = 5.7 + 3.8 \cdot V_{\text{wind}}$$  

(5)

$$T_e = 30 + 0.0175 \times (I - 300) + 1.14 \times (T_a - 25),$$  

(6)

where $V_{\text{wind}}$ is the wind velocity (m/s); $T_e$ is the PV cover temperature ($^\circ$C) and $T_a$ is the ambient temperature($^\circ$C);

$$Q_{c,\text{sky}} = \varepsilon_c \cdot \sigma \cdot (T_{e}^4 - T_{s}^4)$$  

(7)

$$T_s = 0.037536 \cdot T_{a}^{1.5} + 0.32T_a,$$  

(8)

where $\varepsilon_c$ is the PV/T cover layer emissivity, $\sigma$ is the Stefan-Boltzmann's constant, $5.67 \times 10^{-8}$ W m$^{-2}$ K$^{-4}$ and $T_s$ is the sky temperature ($^\circ$C);

$$Q_{\text{conv,pl,heo}} = h_{\text{air}} \cdot (T_{pl} - T_{\text{heo}}),$$  

(9)

where $h_{\text{air}}$ is the convective heat transfer coefficient of natural air layer between the PV module and PHE (W/m$^2$ K) and $T_{pl}$, $T_{\text{heo}}$ are the EVA plastic layer temperature and outer wall of PHE temperature, respectively ($^\circ$C);

$$h_{\text{air}} = \frac{N_{u} \cdot \lambda_{\text{air}}}{\delta_{\text{air}}},$$  

(10)

where $\lambda_{\text{air}}$ is the thermal conductivity of air (W/m K) and $\delta_{\text{air}}$ is thickness of the air gap between cover and PV layer (m).

$Nu$ is the Nusselt number as obtained according to

$$N_{u} = \left[ 0.06 - 0.017 \left( \frac{\beta_s}{90} \right) \right] Gr^{1/3},$$  

(11)

where $\beta_s$ is the title–angle of PV panels and $Gr$ is the Grashoff number as given as

$$Gr = \frac{g \cdot (T_{pl} - T_{\text{heo}}) \cdot \delta_{\text{air}}^3}{\nu_{\text{air}}^2 \cdot T_{air}},$$  

(12)

$$Q_{pl,\text{heo}} = \varepsilon_{pl} \cdot \sigma \cdot (T_{pl}^4 - T_{\text{heo}}^4),$$  

(13)

where $\varepsilon_{pl}$ is the EVA plastic layer emissivity;

$$Q_{\text{ele}} = \eta_e A_{\text{eff}} I,$$  

(14)

where $\eta_e$ is the PV cells electrical efficiency (%) and $A_{\text{eff}}$ is the effective area of PV panels (m$^2$).
energy balance equation of the soil domain is given as

$$\rho_{\text{soil}} c_{\text{soil}} \frac{\partial T_s}{\partial t} = \frac{\partial}{\partial x} \left( k_{\text{soil}} \frac{\partial T_s}{\partial x} \right) + \frac{\partial}{\partial y} \left( k_{\text{soil}} \frac{\partial T_s}{\partial y} \right) + \frac{\partial}{\partial z} \left( k_{\text{soil}} \frac{\partial T_s}{\partial z} \right)$$

(17)

Heat transfer through the copper pipe is treated as pure heat conduction as well and defined as 3D heat conduction versus time. Thus, the corresponding energy conservation equation can be written as

$$\rho_{\text{pipe}} c_{\text{pipe}} \frac{\partial T_p}{\partial t} = \frac{\partial}{\partial x} \left( k_{\text{pipe}} \frac{\partial T_p}{\partial x} \right) + \frac{\partial}{\partial y} \left( k_{\text{pipe}} \frac{\partial T_p}{\partial y} \right) + \frac{\partial}{\partial z} \left( k_{\text{pipe}} \frac{\partial T_p}{\partial z} \right),$$

(18)

where $\rho_{\text{soil}}$ and $\rho_{\text{pipe}}$ are the densities of soil, grout and pipe (kg/m$^3$); $c_{\text{soil}}$ and $c_{\text{pipe}}$ are the thermal capacities of soil, grout and pipe (J/kg·°C); $k_{\text{soil}}$ and $k_{\text{pipe}}$ are the thermal conductivities of soil, grout and pipe (W/m·K); and $T_s$ and $T_p$ are the temperatures of soil and pipe (°C), respectively.

2.2.2. Energy balance in fluid region

In the fluid region, heat convection occurs between the rectangular pipe and working fluid. The fluid flow direction of inlet pipe is opposite to the direction of outlet pipe. Thence, the energy balance equations of inlet pipe and outlet pipe need to be developed separately.

The fluid in the inlet pipe (downward flow) can be modelled as follows:

$$\rho_{\text{fluid}} c_{\text{fluid}} \frac{\partial T_{\text{inlet}}}{\partial t} + (\rho c_v) f \frac{\partial T_{\text{inlet}}}{\partial z} = k_{\text{fluid}} \frac{\partial^2 T_{\text{inlet}}}{\partial z^2} + b_{ig} \left( T_{\text{grout}} - T_{\text{inlet}} \right).$$

(19)

Similarly, the fluid in the outlet pipe (upward flow) is also modelled as

$$\rho_{\text{fluid}} c_{\text{fluid}} \frac{\partial T_{\text{outlet}}}{\partial t} + (\rho c_v) f \frac{\partial T_{\text{outlet}}}{\partial z} = k_{\text{fluid}} \frac{\partial^2 T_{\text{outlet}}}{\partial z^2} + b_{og} \left( T_{\text{grout}} - T_{\text{outlet}} \right),$$

(20)

where $\rho_{\text{fluid}}$ is density of working fluid (kg/m$^3$); $c_{\text{fluid}}$ is thermal capacities of working fluid (J/kg·°C); $k_{\text{fluid}}$ is thermal conductivity of working fluid (W/m·K); $T_{\text{inlet}}$ and $T_{\text{outlet}}$ are inlet and outlet fluid temperatures (°C), respectively; $b_{ig}$ is reciprocal of thermal resistance $R_{ig}$ between inlet pipe and grout (W/m$^2$·K); and $b_{og}$ is reciprocal of thermal resistance $R_{og}$ between outlet pipe and grout (W/m$^2$·K).

2.2.3. Initial and boundary conditions

Cecinato and Loveridge [28] illustrated that the hetero-thermal zone should be accounted for EP design. The ground temperature is a sinusoidal wave function of time and depth, and can be expressed as [28, 29]

$$T_{\text{soil}} (Z, t_{\text{year}}) = T_{\text{mean}} - T_{\text{amp}} \cdot \exp \left( -Z \times \sqrt{\frac{\pi}{365}} \right) \cdot \cos \left[ \frac{2\pi}{365} \left( t_{\text{year}} - t_{\text{shift}} - \frac{Z}{2} \times \sqrt{\frac{365}{\pi \cdot \alpha}} \right) \right],$$

(21)

where $T_{\text{soil}} (Z, t_{\text{year}})$ is undisturbed ground temperature at time (t) and depth (Z) (°C), $T_{\text{mean}}$ is mean surface temperature (average air temperature) (°C), $T_{\text{amp}}$ is amplitude of surface temperature ([maximum air temperature—minimum air temperature]/2) (°C), Z is depth below the surface (surface = 0) (m), $\alpha$ is thermal conductivity of soil (J/kg·K), $t_{\text{year}}$ is current time (day) and $t_{\text{shift}}$ is day of the year when the coldest air temperature occurs (day).

Boundary conditions are classified into two categories: the first is expressed in terms of temperature at the boundary while the second is presented in terms of temperature gradient. In the case of the first boundary condition, at $z = 0$, the inlet pipe temperature is equal to the fluid temperature:

$$T_{\text{inlet}} (0, t) = T_{\text{fluid}} (t).$$

(22)

In terms of the second boundary condition, at $z = 0$, heat flux at the exit of outlet pipe is depicted as

$$\frac{\partial T_{\text{outlet}} (0, t)}{\partial z} = 0.$$  

(23)

2.3. Heat pump

The heat pump is coupled with PV/T and geothermal closed loop to provide a comfortable climate for the poultry house in the winter. A water-to-water heat pump model is used in this study and its parametric model reflecting the effect of compressor rotation speed is adopted [30].

$$m_r = V_r \omega_r \rho_{r,suc} \left[ 1 + C_r \left( 1 - \frac{P_{r,cond}}{P_{r,cond}} \right)^{\frac{1}{n}} \right]$$

(24)

$$\Delta \xi_{\text{comp}} = \xi_{r,dis} - \xi_{r,suc} = \frac{n}{n - 1} \cdot \frac{P_{r,cond}}{P_{r,suc}} \cdot \left( \frac{P_{r,cond}}{P_{r,cond}} - 1 \right)^{\frac{n-1}{n}}$$

(25)

$$Q_{\text{el}} = \frac{m_r \Delta h_{\text{comp}}}{\eta_{\text{comp}}}. $$

(26)
where $m_r$ is the refrigerant mass flow rate (kg/s), $V_c$ is the compressor swept volume (m³), $\omega$ is the compressor rotational speed (rev/s), $\rho_{suc}$ is the compressor suction refrigerant density (kg/m³), $C_c$ is the compressor volumetric coefficient, $P$ is the pressure (kPa), $\xi$ is the specific enthalpy (kJ/kg), $n$ is the polytropic compression coefficient, $\eta_{comp}$ is the compressor mechanical efficiency. $\Delta \xi$ is the specific enthalpy change (kJ/kg) and $Q_{el}$ is the electrical energy consumption (kW).

The coefficient of performance $\text{COP}_h$ of heat pump is defined as

$$\text{COP}_h = \frac{Q_{heating}}{Q_{el}}, \quad (27)$$

where $\text{COP}_h$ is the heating COPs and $Q_{heating}$ is the heating capacities (kW).

### 2.4. Energy storage buffer tank

In the energy storage buffer tank, the energy is stored mainly as sensible heat and a fully mixed water tank is assumed, thus the outlet water temperature from the storage tank is equal to the mean water temperature in buffer tank. Based on the conversion of energy, the equation of the buffer tank can be given as follows:

$$C_s M_s \frac{dT_s}{dt} + U_s A_s (T_s - T_o) + Q_{HP} = Q_{PV/T} + Q_{GSHP} \quad (28)$$

where $C_s M_s$ is the thermal capacity of storage medium of buffer tank (kJ/K), $T_s$ is the mean water temperature of buffer tank (°C), $t$ is the time (s), $U_s$ is the heat losses coefficients of buffer tank (W/m² °C), $A_s$ is the heat losses area of buffer tank (m²), $Q_{HP}$ is the heat output from heat pump (kW), $Q_{PV/T}$ is the heat production from PV/T system (kW) and $Q_{GSHP}$ is the heat production from GSHP system (kW).

### 3. METHODOLOGY

#### 3.1. John Wright Ltd. chicken farm

This ‘WelChic heating system’ is applied to a real chicken farm which is named John Wright Ltd. (JWL), located in Newark-on-Trent in Nottinghamshire in the East Midlands of England, UK. JWL is an intensive family farming site, growing broilers on a 9 weeks’ cycle, the pullets leave at 4 weeks and the cockles at 7 weeks. There are four sheds that vary slightly in size and have 40000 chicken in at a time.

An aerial photograph of JWL is presented in red region in Figure 4a. Shed 1 has been selected for ‘WelChic project’ as it is the smallest shed on-site and therefore has the lowest heating requirement in cold condition as presented in Figure 4a–c. This means that for the budget available the ‘WelChic heating system’ is able to provide the highest proportionate demand. The photograph of young pullets, the heating control system and 66 kW gas burner used in shed 1 are presented in Figure 4e–g, respectively. The required internal desired temperature within shed 1 is 32°C for the first 2 days of the broiler chicks at the eye level of the chicks, thereafter, the temperature is then dropped by 2–3°C each week, down to ~20°C at the age of 7 weeks. The desirable relative humidity (RH) range is between 50% and 70% in shed 1. Currently, the temperature required for the chicken brooding are on mains provided by 2 × 66 kW gas burners and hot air blowers. A gas burner is also used in fall and winter periods to aid the lamps in heating during the first 2 weeks of brooding. In fact, the current system seems to be inefficient due to the low energy factor of the electrical lamps and because the two gas burners are utilized to heat the whole poultry space instead of the microenvironment surrounding the chicken. On the other hand, fresh air is supplied to the house through mechanical ventilation; fans are installed on wall openings and supply variable flow rates according to ventilation requirements that also ventilate the whole space instead of that at the chicken level. The thermal comfort requirements and air quality requirements for shed 1 are presented in Table 1.

#### 3.2. Meteorological and shed 1 detailed information

Based on the meteorological database system, the input data including the average ambient temperatures, available solar irradiation and wind velocity used for the calculations are shown in Tables 2 and 3. The shed 1 input information for the numerical model with their dimensions is presented in Table 4.

#### 3.3. Program algorithm

The hybrid numerical model is solved based on the EES. The basic geometrical parameters, initial and boundary conditions are the main input data. Figure 5a shows the flow chart of the solution procedure for estimating the performance of ‘WelChic heating system’. The computational processes of PV/T and geothermal closed loop models are illustrated in Figure 5b and c. In the present simulation, the ‘WelChic heating system’ operates to collect the solar energy from 9:00 to 15:00 (solar time). The detailed solving process is shown as follows:

1) Input the initial conditions and weather conditions. The initial conditions involve temperature of each component, and the weather conditions include solar radiation intensity, ambient temperature, soil initial temperature, etc.
2) Solve the energy balance equations of PV/T. When solving the mathematical model of PV/T collection system, the Newton’s backward interpolation formula is applied to discretize the energy balance equations, then solve the simultaneous equations to obtain the temperature of each component at time $t + \Delta t$ based on time $t$.
3) Solve the simultaneously energy balance equations of geothermal closed loop components. The nodal temperature is calculated at each step until the time required for the fluid to flow through the pile heat exchangers is reached. And also, the program will output the simulation data if the results meet the air quality requirements.
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**Table 1.** Thermal comfort requirements and air quality requirements in shed 1 at JWL.

| Brooding time       | Body mass (kg) | Temperature (°C) | Air quality          | Air speed (m/s) |
|---------------------|----------------|------------------|----------------------|-----------------|
| The first two days  | 0.045          | 32               |                      | 0.17            |
| First week          | 0.045 – 0.182  | 30 – 32          | CO₂ ≤ 2500 ppm       | 0.17 ~ 0.47     |
| Second week         | 0.227 – 0.455  | 28 – 30          | NH₃ ≤ 25 ppm         | 0.58 ~ 0.94     |
| Third week          | 0.500 – 0.864  | 26 – 28          |                      | 0.58 ~ 1.52     |
| Forth week          | 0.909 – 1.364  | 24 – 26          |                      | 1.58 ~ 2.15     |
| Fifth week          | 1.409 – 1.909  | 22 – 24          |                      | 2.20 ~ 2.76     |
| Sixth week          | 2.000 – 2.273  | 20 – 21          |                      | 2.86 ~ 3.15     |
| Seventh week        | N/A            | 20               |                      | N/A             |

**Table 2.** Temperatures variations in Newark throughout a year.

| Jan    | Feb    | Mar    | Apr    | May    | Jun    | Jul    | Aug    | Sep    | Oct    | Nov    | Dec    |
|--------|--------|--------|--------|--------|--------|--------|--------|--------|--------|--------|--------|
| Mean temperature (°C) | 5.1    | 4.9    | 6.7    | 8.9    | 12.3   | 15.1   | 16.8   | 17.1   | 14.4   | 11.2   | 7.3    | 4.7    |
| Absolute maximum temperature (°C) | 13.1   | 13.85  | 15.3   | 18.3   | 21.1   | 22.8   | 28.4   | 25.5   | 24.6   | 19.6   | 15.6   | 12.12  |
| Absolute minimum temperature (°C) | -5.1   | -4.4   | -3.9   | 1.7    | 1.5    | 3.3    | 5.9    | 6.7    | 4.4    | 4.7    | 0.6    | -3.57  |

**Table 3.** Solar radiation and wind velocity in Newark throughout a year.

| Jan    | Feb    | Mar    | Apr    | May    | Jun    | Jul    | Aug    | Sep    | Oct    | Nov    | Dec    |
|--------|--------|--------|--------|--------|--------|--------|--------|--------|--------|--------|--------|
| Solar radiation (W/m²) | 27.7   | 48.5   | 98.6   | 165.2  | 193.6  | 207.2  | 195.8  | 153.9  | 126.4  | 67.7   | 37.1   | 24     |
| Wind speed (m/s)      | 5.2    | 4.8    | 4.7    | 4.3    | 4.3    | 3.8    | 3.7    | 3.6    | 3.9    | 4.4    | 4.7    | 4.5    |

4) The parameters of heat pump, including the inlet and outlet state parameters of refrigerant, compressor power and refrigerant flow rate, are solved based on the performance parameters provided by manufacturers.

5) Results outputs including daily resolution data for the system electrical and thermal output, heat pump output and annual system energy assessment.
Fig. 5. Flowchart of the computational procedure: (a) whole system, (b) PV/T system and (c) geothermal closed loop system.
In this study, the simulation is implemented by an Intel 2 Duo 3 GHz processor with 64-bit operation system, one step calculation would take $\sim 10$ s for proposed mode.

4. EXPERIMENTAL WORK

The thermal input to the evaporator of heat pump could be obtained by heat sources from the PV/T array and geothermal closed loop. Figure 6a presents a layout of the ‘WelChic heating system’ for shed 1 at JWL, which complete with all components and parts at the end of May 2016. Due to structural reasons, the PV/T array is installed on the workshop roof. A ‘WelChic plant room’ is built at the south-west corner of shed 1. The heat pump, buffer tank, system connections and instrumentation devices are all installed in the plant room. As shown in Figure 6b and c, the electrical and thermal output from the PV/T array have been piped down the side of the workshop, under the access road and through shed 1 to the ‘WelChic plant room’. The fan coil is vertically mounted, half way down the western portion of shed 1 on the southern façade wall.
Table 5. Solar optical parameters of PV cells.

| PV module type   | Absorbance, \( \alpha \) | Emissivity, \( \varepsilon \) | Thickness, \( \delta \) | Reference efficiency (%) | Temperature coefficient, \( \beta_p \) (°C⁻¹) |
|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|
| Polycrystalline | 0.9             | 0.96            | 0.04            | 15.88           | 0.035           |

Fig. 8. Schematic diagram of (a) PHE heat mate size and structure and (b) PHE heat mat connection.

4.1. PV/T module
Because the survey presented that none of the chicken sheds are suitable for the PV/T array. Thereby, only the workshop, which is situated next to shed 1 as shown in Figure 7a is deemed suitable pending structural reinforcement work.

As presented in Figure 7b, the completed PV array and PHE heat mats configuration are installed on the workshop roof. There are 52 (260Wp) Canadian solar PV panels installed on the roof and they are mounted at a 15-degree angle oriented to the south to improve the energy capture. The solar cell type is polycrystalline silicon absorber with the cell efficiency of 15.88%. The operation temperature of PV module ranges from −40°C to 85°C. The solar optical parameters of the PV module are illustrated in Table 5. PV/T electrical and thermal output is in shuttering on the external wall of workshop along with inverter as depicted in Figure 7c.

Furthermore, the SU10 PHE heat mats are shown in Figure 8a. The PHE has a 20 mm diameter of main manifold tube at each end. 4.3 mm diameter of capillary tubes are used to connect between the two manifold tubes, and the distance between the capillary tubes is 10 mm. Shuttering around the edge of the PV array is added in order to minimize heat loss and improve PHE performance. According to Figure 8b, beneath the PV array, four 1 m × 12 m PHE mats are installed between the PV panel mounting rails, and then the PV panels are placed on top. The two heat mats in each row will be connected in series. The two rows (and whole array) will be piped as a Tichelman reverse return. This is to ensure equal water flow rates throughout the PHE. The water flow in the PHE will be between 1.5 L/min (0.1m³/hour) and 7.5 L/min (0.43m³/hour). The water circulated through the PHE will be used as a thermal input to the heat pump evaporator. The technical parameters of the PV modules and PHE can be depicted in Table 6.

Table 6. Technical parameters of the PV and PHE system.

| Component | Description | Value |
|-----------|-------------|-------|
| PV modules | Module dimensions | 1650 × 992 × 40 mm |
| No. of PV panels | 52 |
| Cell type | Polycrystalline |
| Packing factor | 0.92 |
| Conversion efficiency | 15.88% |
| Nominal max. power | 260 W |
| Maximum voltage | 30.4 V |
| Maximum current | 8.56 A |
| Open circuit voltage | 37.5 V |
| Short circuit current | 9.12 A |
| PHE | Module dimensions | 1 × 12 m |
| No. of PHE | 4 |
| The diameter of main manifold tube | 20 mm |
| The diameter of capillary tubes | 4.3 mm |
| Spacing between tubes | 10 mm |
| Max temperature allowed in PHE | 60°C |
| Max pressure allowed in PHE | 1 MPa |

4.2. Ground copper pipe array
The ground array consists of 50 2.5 m long (3.5 m deep) vertical 15 mm diameter of copper pipes. The surface area of the ground array is roughly 10 m × 10 m. The vertical copper ground pipes are connected up and the pipe run back to the ‘WelChic plant room’. Thermocouples have been placed in the vertical borehole heat exchangers to measure soil temperatures. The thermal physical properties of the material used in the simulation are illustrated in Table 7.

Table 7. Technical parameters of the PV and PHE system.

4.3. Heat pump and fan coil
In the ‘WelChic heating system’, a 15 kW F1145 NIBE heat pump is selected as it is the largest capacity the consortium and could
Table 7. The input parameters of ground copper loop utilized in the simulation.

| Fluid (R407 refrigerant) |  |
|--------------------------|--|
| Density                  | 512 kg/m³ |
| Kinematic viscosity      | 209.09x10⁻⁶ m²/s |
| Heat capacity            | 1550 J/(kg·K) |
| Thermal conductivity     | 0.103 W/(m·K) |

| Copper                   |  |
|--------------------------|--|
| Density                  | 950 kg/m³ |
| Heat capacity            | 2300 J/(kg·K) |
| Thermal conductivity     | 0.45 W/(m·K) |

| Soil                     | Depth (m) | Thermal conductivity | Density |
|--------------------------|-----------|----------------------|---------|
| Mixed gravel and coarse sand, 0–2.22 m | 1.30 W/(m·K) | 2277 kg/m³ |
| Sand gravel, 2.22–3.3 m   | 1.15 W/(m·K) | 2094 kg/m³ |
| Gravelly clay, 3.3–5.5 m  | 1.68 W/(m·K) | 2223 kg/m³ |
| Weighted mean             | 1.50 W/(m·K) | 2260 kg/m³ |

Fig. 9. Experimental photos of (a) labelled plant room contents, (b) buffer tanks system and (c) 15 kW NIBE heat pump.

afford while considering adequate thermal input from the PV/T and geothermal closed loops. The NIBE unit can also be used in reverse to provide summer time cooling if required. The heat pump uses R407C as the refrigerant. The hot water produced by the heat pump at its condenser is stored in a buffer tank, and then is circulated through a fan coil within shed 1. During periods of no thermal demand, the PV/T thermal output is used to recharge the soil for maintaining the heat balance. Figure 9a presents a labelled photograph of the ‘WelChic pant room’ interior. A 300 L NIBE UKV buffer tank has been installed between the heat pump condenser water loops as presented in Figure 9b. This volume could provide adequate thermal storage capacity to ameliorate variations in heating supply and demand. Figure 9c exhibits the ideal performance of the 15 kW NIBE heat pump with 3.14 kW compressor. The heat pump produces hot water at a temperature between 35°C and 65°C through the heat pump condenser. The heat pump also has an in-line electrical immersion heater (9 kW) to boost the water flow temperature on the condenser side if required. Table 8 presents technical data of the NIBE F1145.

Table 8. Nominal specification of the NIBE F1145 heat pump [31].

| Description                          | Value       |
|--------------------------------------|-------------|
| Supplied output at 0/35°C            | 3.4 kW      |
| Delivered power at 0/35°C            | 15.4 kW     |
| COP 0/35°C                           | 4.5         |
| Immersion heater                     | 9 kW        |
| Refrigerant R407C mass flow rate     | 0.085 kg/s  |
| Operating voltage                    | 400 V       |
| Max temperature heating medium (flow/return pipes) | 24.7 bar |
| (flow/return pipes)                  | 70/58 °C    |

What is more, the fan coil is installed vertically on the south façade wall, roughly one quarter of the way down shed 1 at the beginning of July 2016 as shown in Figure 10a, and the installed timer switch for the fan coil unit is presented in Figure 10b. This indicates that for the fan coil to work the temperature must be outside the set temperature on the temperature controller and be within the time frame set on the timer control. As its winter
to measure incident solar radiation. This data can then be used to calculate the PV/T electrical and thermal efficiency. The initial design parameters are given in Table 9.

5. RESULTS AND DISCUSSION

In this section, the 3D numerical model is validated in light of PV array electrical production, PHE, geothermal cooper pipes and heat pump thermal outputs by the experimental data. Meanwhile, the error analysis is obtained via Eq. (29). What is more, the whole system performance is investigated and predicted over a whole year.

\[
\text{Error} = \left| \frac{T_{\text{Numerical}} - T_{\text{Experimental}}}{T_{\text{Experimental}}} \right| \quad (29)
\]

5.1. PV/T array

Before the 3D numerical model is used to simulate the annual system performance, it needs to be validated at first. Hence, these comparisons between experimental and numerical results are illustrated during system operation period from 1 November 2016 to 31 January 2017.

5.1.1. PV array electrical output

Figure 11 shows the daily PV electrical generation comparison between the numerical results and experimental data, and this data is collected from the GSM online interface. As shown in Table 10, the maximum error is 14.93% noticed at the end of operation stage and the average error is 9.26%. Meanwhile, the comparison between the two curves confirms that the numerical results are in very good agreement with those given by the experiment results, which verifies the reliability of the developed model. The total PV electrical generation in the period from 1 November 2016 to 31 January 2017 is 1125.89 kWh, based on the experimental data. By comparison, the simulated results present a close proximity of value, reaching 1247.51 kWh within a 10% error. This indicates that the PV array electrical efficiency of 14.56% has been achieved over the period.

5.1.2. PHE thermal output

The PHE thermal output has a trend of fluctuation during the whole operation period as shown in Figure 12. In terms of the experimental analysis, the highest working fluid temperature is \( \sim 15.75^\circ C \) on 10 November 2016 whereas the lowest one is \( \sim 0.75^\circ C \) on 25 January 2017.

The temperatures from the simulation have the similar variation pattern with the experimental data for the whole heating period, the maximum temperature difference between the simulation and experimental results reaches \( \sim 14.72\% \) on 14 November 2016, the mean error being 9.11%, while the minimum error is \( \sim 3.29\% \) on 31 December 2016, as indicated in Table 11.
Table 9. **Initial condition and basic parameters.**

| Component                | Description                              | Value       |
|--------------------------|------------------------------------------|-------------|
| Mean mass flow rate      |                                          | 5 L/min     |
| Specific heat of R407    |                                          | 4190 J/kg K |
| PV/T                     | Thermal conductivity of cover layer      | 1.0 W/m K   |
| Mass of R407             |                                          | 45 kg       |
| Initial ground surface temperature | Initial ground surface temperature | 10.4°C      |
| Geothermal closed loop   | Soil body temperature (soil far field boundary) | 15.0°C      |
|                          | Soil bottom temperature                  | 15.5°C      |
|                          | Fluid inlet temperature                  | 1.2°C       |

Table 10. **Relative errors of PV array electrical output from 1 November 2016 to 31 January 2017.**

| Date            | Numerical results (kWh) | Experimental results (kWh) | Errors (%) |
|-----------------|-------------------------|---------------------------|------------|
| 03 November 2016| 8.84                    | 10.38                     | 14.84      |
| 20 November 2016| 3.93                    | 4.10                      | 4.15       |
| 28 November 2016| 26.74                   | 29.40                     | 9.05       |
| 15 December 2016| 4.91                    | 5.35                      | 8.22       |
| 23 December 2016| 6.38                    | 6.61                      | 3.48       |
| 31 December 2016| 1.43                    | 1.68                      | 14.88      |
| 08 January 2017 | 2.85                    | 3.35                      | 14.93      |
| 24 January 2017 | 25.03                   | 27.34                     | 8.45       |
| 26 January 2017 | 5.46                    | 5.76                      | 5.38       |
| 31 January 2017 | 6.32                    | 6.62                      | 4.53       |

Table 11. **Relative errors of PHE thermal output from 1 November 2016 to 31 January 2017.**

| Date            | Numerical results (°C) | Experimental results (°C) | Errors (%) |
|-----------------|------------------------|---------------------------|------------|
| 03 November 2016| 12.73                  | 14.53                     | 12.39      |
| 14 November 2016| 2.55                   | 2.99                      | 14.72      |
| 28 November 2016| 6.32                   | 7.12                      | 11.23      |
| 15 December 2016| 4.44                   | 4.92                      | 9.76       |
| 23 December 2016| 4.98                   | 5.47                      | 8.96       |
| 31 December 2016| 8.23                   | 8.51                      | 3.29       |
| 08 January 2017 | 7.31                   | 7.99                      | 8.51       |
| 21 January 2017 | 5.63                   | 6.18                      | 8.90       |
| 26 January 2017 | 10.78                  | 11.17                     | 3.49       |
| 31 January 2017 | 1.72                   | 1.79                      | 3.91       |

5.2. Geothermal cooper pipe thermal output
The temperatures from the simulation results have the similar variation trend with the experimental data during the operating period as given in Figure 13. To be more specific, the highest fluid temperature could achieve 14.23°C on 15 January 2017 while the lowest one is ~0.89°C on 08 November 2016 from experimental testing.

Furthermore, the maximum temperature difference between the simulation and experimental results could achieve ~11.33% on 16 December 2016, the mean error being 6.36%, while the minimum error is ~2.40% on 15 January 2017, as illustrated in Table 12. Based on the fluid flow rate within the ground array (0.13 kg/s total), a total (maximum) thermal output from ground array of ~4.2 kW is demonstrated.

5.3. Heat pump thermal output
Figure 14 presents the cumulative thermal output of the heat pump comparison between the numerical results and experimental data during the period from 1 November 2016 to 31 January 2017. Specifically, when the heat pump system can operate at
Table 12. Relative errors of geothermal copper pipe thermal output from 1 November 2016 to 31 January 2017.

| Date            | Numerical results (°C) | Experimental results (°C) | Errors (%) |
|-----------------|------------------------|---------------------------|------------|
| 03 November 2016| 9.11                   | 9.91                      | 8.07       |
| 20 November 2016| 7.23                   | 7.78                      | 7.07       |
| 28 November 2016| 6.24                   | 6.69                      | 6.73       |
| 16 December 2016| 3.74                   | 3.09                      | 11.33      |
| 23 December 2016| 5.37                   | 5.72                      | 6.12       |
| 31 December 2016| 6.68                   | 7.03                      | 4.98       |
| 08 January 2017 | 7.47                   | 7.82                      | 4.48       |
| 15 January 2017 | 14.23                  | 14.58                     | 2.40       |
| 26 January 2017 | 13.28                  | 13.62                     | 2.50       |
| 31 January 2017 | 1.92                   | 2.01                      | 4.48       |

Fig. 13. Thermal output of ground copper pipe from 1 November 2016 to 31 January 2017.

Fig. 14. Cumulative thermal output of the heat pump from 1 November 2016 to 31 January 2017.

the initial stage, the chickens within shed 1 are very young, thus they need a very high internal air temperature of around 30°C (refer to Table 1). It is also confirmed from Figure 14 that the heat pump thermal output at this early stage is higher than those at the later stages during system operating period. This is because the required internal shed temperature decreases when the chickens grow older, thus the required thermal input of the heat pump reduces.

As illustrated in Table 13, the maximum error is ~9.30% noticed on 16 January 2017, the average error being 7.71%, while the minimum error is ~5.49% on 05 December 2016. Furthermore, the maximum cumulative thermal output of the heat pump could achieve ~2211 kWh via experiment on 31 January 2017, by comparison, the maximum cumulative simulated result could reach ~2431 kWh within a 9.05% difference at the same day. As a result, the 3D numerical model is effectively supported by the experimental data and can be utilized to investigate and predict the energy performance of the ‘WelChic heating system’ for a whole year.

5.4. Year-round performance assessment

Figure 15 describes the year-round performance of PV electrical output and heat pump thermal output. The maximum and minimum average monthly PV electrical output is ~1800 kWh in July and 400 kWh in December, respectively. Based on the 3D model, the annual electricity generation from PV array could achieve ~11867 kWh.

Operating continually, the 15 kW heat pump (3.4 kW compressor), can operate for 43.5% of the year using PV electricity. The electrical supply from the PV array/demand from heat pump could be met on balance, using the grid as a form of electrical store. By contrast, the thermal generation from heat pump is ~30210 kWh. Moreover, it can be seen that the highest average monthly thermal output is achieved ~4550 kWh in July and the lowest is ~1089 kWh in December.

Year-round performance of thermal/electrical efficiency and COP variation is depicted in Figure 16. It can be seen that the
Table 13. Relative errors of heat pump cumulative thermal output from 6 January 2016 to 31 January 2017.

| Date               | Numerical results (kWh) | Experimental results (kWh) | Errors (%) |
|--------------------|-------------------------|---------------------------|------------|
| 03 November 2016   | 1326                    | 1427                      | 7.08       |
| 20 November 2016   | 1480                    | 1580                      | 6.33       |
| 28 November 2016   | 1599                    | 1700                      | 5.94       |
| 05 December 2016   | 1721                    | 1821                      | 5.49       |
| 15 December 2016   | 1877                    | 2063                      | 9.02       |
| 31 December 2016   | 2061                    | 2247                      | 8.28       |
| 08 January 2017    | 2108                    | 2313                      | 8.86       |
| 16 January 2017    | 2145                    | 2365                      | 9.30       |
| 26 January 2017    | 2190                    | 2410                      | 9.13       |
| 31 January 2017    | 2211                    | 2431                      | 9.05       |

Fig. 16. Heat pump thermal output and COP variation over a year.

highest average monthly electrical efficiency is achieved in July (~15%) and the lowest in December (~6.3%). Increasing PV cell temperatures result in voltage and electrical efficiency reduction. The use of the PHE under the PV array means reasonably low PV cell temperatures are observed in summer. This is because the circulating water is taking some of the cell heat way from the array and therefore PV electrical efficiency is enhanced. The highest PV/T array thermal efficiency is achieved in summer (28.3% in July) and the lowest in winter (7.3% in December). Operating in conjunction with the ground pipe array, a maximum heat pump condenser output of 10.25 kW is obtained in July (COP = 5.01) and the lowest of 6.71 kW in December (COP = 1.97). As a result, some gas burner capacity would be required alongside the heat pump to heat the poultry houses sufficiently, in particular, from December to February.

6. CONCLUSIONS

In this paper, an innovative hybrid PV/T module with a new type of inexpensive PHE coupled to a low-cost geothermal heat pump system is installed and tested in a real poultry house. A detailed numerical model for the ‘WelChic heating system’ is developed based on finite volume method and solved by EES software. The results indicate that a good agreement with less than 15% difference between the numerical and experimental results is obtained. Furthermore, the annual electrical and thermal output can be predicted and investigated via the 3D model based on the East Midlands’ climate conditions in the UK. Some key findings are summarized as follows:

- The PV electrical output from the ‘WelChic heating system’ could achieve 11867 kWh per annum. It not only can fulfil the poultry house electrical demand, but also can provide ~43.5% electricity demand of the heat pump compressor operating.
- The maximum average monthly system electrical output is ~1800 kWh in July, whereas the minimum is ~400 kWh in December. Moreover, the highest mean monthly system thermal output is obtained ~4550 kWh in July, whereas the lowest is attained ~1089 kWh in December.
- The thermal output of the heat pump system is ~30210 kWh per annum, which can meet the poultry house thermal requirement.
- Operating in conjunction with the PHE and ground pipe array, a maximum heat pump condenser output of 10.25 kW is achieved in July (COP = 5.01) and a lowest of 6.71 kW in December (COP = 1.97). As a result, some gas burner capacity would be required alongside the heat pump to heat the poultry houses sufficiently, in particular, from December to February.

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