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Full length Research Paper

Computerized data acquisition to demonstrate the influence of coil density on hog cooling pad heat transfer and long-term operational sustainability

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Abstract

Researchers at Purdue University have developed a device which removes excess heat from sows and helps to mitigate thermal stress. The optimal amount of cooling coil beneath the top pad in the device needed further investigation. A second prototype with eight coolant pipes was compared to the original six-pipe prototype design. Bench tests of thermal and heat transfer properties similar to the initial prototype testing were conducted with instrumented heat exchangers and completed using an animal simulator that could maintain a constant temperature. The results indicated that at steady state operating conditions, the heat rejection (heat transfer rejected through coolant) in the eight-pipe unit was significantly greater than the six-pipe unit, heat exchanger efficiency in the eight-pipe unit was significantly larger than in the six-pipe unit, the effectiveness in the eight-pipe unit was significantly greater than the six-pipe unit, and the conductive heat transfer coefficient was larger than that of the six-pipe unit. Intermittent operation testing showed improvement with increasing off periods in peak heat rejection, average heat rejection, efficiency, and effectiveness. This work demonstrated the superiority of the eight-pipe design over that of the six-pipe design, and further design prototypes have incorporated this additional eight-pipe coil length.

Keywords: Biological thermo-physics, conduction, convection, cooling systems, heat transfer, swine, thermal stress.

INTRODUCTION

Background on Thermal Stress and Animal Cooling Technologies

Many researchers from numerous fields believe that thermal stress in the animals is the single biggest negative factor on profitability in modern hog production (Knox et al., 2013; Nardone et al., 2010; Prunier et al., 1997; Quiniou & Noblet, 1999; Stinn & Xin, 2014). Multiple cooling technologies, included but not limited to, low fat diets, misting, snout cooling, radiative cooling, swamp coolers, and enhanced air circulation, have been attempted (Bjerg et al., 2019). The undesirable side effects from some of these technologies in commercial settings, including lower milk production, increased mess and safety concerns, complicated infrastructure additions with low utility, high operational costs, increased ambient humidity, and general ineffectiveness, have prevented the

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large-scale acceptance of these technologies (Bjerg et al., 2019). Although many swine operations rely on some form of evaporative cooling, it is fair to note that these systems have significant problems and are less than ideal. Several studies have demonstrated that floor cooling is a successful strategy in mitigating the effects of high temperatures on animals (Huynh et al., 2004; Mondaca et al., 2013; Ortiz et al., 2015; Perano et al., 2015; Rojano et al., 2011; Rojano et al., 2019; Shaffer, et el., 2001; Shaffer et al., 2017; Silva et al., 2006; Silva et al., 2009; Van Wagenberg et al., 2006). These researchers were experimenting with the general concept of floor cooling or cooling for dairy animals, and they did not design devices that were scalable or practical to commercialize within production swine operations. Significant work from both the engineering and animal sciences perspective has been invested at Purdue University to create a controllable cooling pad that is durable within a production environment and can be utilized by large-scale commercial enterprises. Multiple hog equipment manufacturers have examined the feasibility of the system and have been interested in commercial production. One firm is currently in the process of finalizing a licensing arrangement with the university. Schinckel and Stwalley (2015) produced the original design of the device, and an Agricultural & Biological Engineering (ABE) capstone team experimented with internal heat transfer variance and assembly methodology and constructed the first testing prototype (Geis et al., 2015).

One of the first issues associated with earlier floor cooling concept designs by other researchers was the responsiveness of the units. Cooling pipes buried within a solid concrete floor, attached to a slotted floor grate, or housed within some other thermally massive specialized device react too slowly to environmental or physiological changes to be useful for heat stressed animals. These designs can take a significant amount of time to change the animal interface temperature from the ambient condition to one capable of effective cooling, due to the large thermal heat capacity within the mass of these devices. Depending upon the size of the barn and the cooling system, it can take up to three days to appreciably lower the temperature of a solid concrete floor. Additionally, if the cooling device is in direct conductive contact with another portion of the larger housing structure, it will be trying to draw energy from that source as well, limiting its ability to change temperature and draw heat away from the animal. These issues were squarely addressed in the first Purdue prototype design, and the general features of that unit have been retained in subsequent design iterations. The heat exchanger design is somewhat unique in that it is a single-phase fluid stream indirect recuperative cooler, drawing heat from an extended flat plate surface in contact with the subject animal into what might be described as a flattened pipe coil. An exploded schematic of the device is shown in figure 1, and a cross-section is shown in figure 2.

Previous Cooling Pad Device Testing

Testing by Cabezon et al. (2017a) demonstrated that the Purdue hog cooling pad successfully overcame the thermal heat capacity shortcomings of many previous attempts at floor cooling. The first Purdue prototype device was tested under a variety of flow conditions, and it was shown that the time constant of the device was fairly uniform, averaging about 240 *s*. The specific heat capacity of the device was shown to be two orders of magnitude lower than the equivalent footprint of the solid 15 *cm* thick concrete barn floor slab used by Silva et al. (2006) in the initial Brazilian floor cooling study. It was demonstrated that the heat transfer of the Purdue device was directionally focused on the top plate, absorbing little heat from the facility's structural supporting members or ambient environment. It was also shown that the temporal character of the thermal response at various levels throughout the depth of the unit followed a natural logarithmic curve, and therefore, the overall device exhibits a character similar to that described by Newton's Law of Cooling. Additionally, the top surface plate temperatures across the length and width of the device were reasonably uniform and did not exhibit warm or cold spots (Cabezon et al., 2017a).

The heat transfer characteristics of the pad operating under a constant temperature heat source were reported by Cabezon et al. (2018). Although heat transfer increased with flowrate, regression analysis showed that lower flows were most effective at removing energy from the heat source, and intermittent flows performed even better under bench testing. Biot numbers (**Bi**) based upon the coolant exit temperature are an appropriate measure of the internal operation of the swine cooling pad, because the top plate temperature should ideally remain relatively constant, dropping only slightly from the skin temperature, and all the coolant contained within the piping should warm to the same temperature in intermittent operation. **Bi** values for the cooling pad provide information regarding the internal heat transfer activities in a combined conductive / convective process and represent the conductive resistance over the convective resistance. This does not consider any contact resistance between the sow's skin and the pad, but it does include contact resistances between the aluminum plate and clip and between the clip and copper pipe. **Bi** number showed high temperature gradients in the metal and low gradients in the liquid coolant. The previous bench experiments of Cabezon et al. (2017a, 2018) indicated that the heat transfer in the device was limited by conduction.

The prior work of Cabezon et al. (2017a, 2018) with constant heat source testing demonstrated that the logarithmic temperature decay of the device remained under both steady state and intermittent coolant flows. The 'end' or steady state temperatures throughout the device under bench testing were primarily controlled by the entering coolant temperature and did not vary significantly with flowrates or mode of operation. Cabezon et al. (2017a, 2018) determined that low coolant flowrates and intermittent flow conditions were the preferred means of heat transfer for the first prototype cooling pad. This work was followed by an initial live animal test, and then extended testing with multiple animals. Initial results using multiple animals on individual cooling pads with steady coolant flow have been published (Cabezon et al., 2017b; Cabezon et al., 2017c; Maskal et al., 2018; Parois et al., 2018). At the close of the series of bench testing reported in Cabezon et al. (2017a, 2018), it was recognized that the conduction process was seriously limiting the overall heat transfer, and the number of cooling coil passes within the device was a variable that had remained constant. These bench testing experiences shaped the evolution of the device by providing a better understanding of the heat transfer characteristics of the initial six-pipe prototype device. The accuracy of the experimental apparatus and the means of data collection has been verified in a separate set of experimentation (Seidel et al., 2020). Good design practice certainly dictated that more than one variation of the concept be evaluated, and this paper presents the bench-testing results of this cooling coil number comparison.

Experimental Rationale and Scope

Coolant coil length is clearly a key element in the design of this heat exchanger, but simply adding tubing length and assuming proportional changes in key metrics is not a valid process for a heat exchanger design of this complexity. Although it can be reasonably hypothesized that an increase in coil length will improve the heat exchanger efficiency in continuous operation, the effects of intermittent operation on resource effectiveness cannot be easily predicted from theoretical considerations alone. Preliminary computational investigation also demonstrated that changes in pipe length, width between pipes, and coil layout beneath the top plate could affect the overall Biot number, top plate temperature variance, and top plate temperature drop in the device (Field & Deneke, 2018). Further experimentation was clearly required to optimize the design of the device and reduce coolant use while in operation. No other considerations were examined during this experimentation.

Experimental Objectives

A second prototype cooling pad for experimental testing was created increasing the number of pipes to eight from the previous six. This increased the length of the active coil from 6.5 *m* to 8.6 *m*, an enhancement of one third. This modification provided more contact length with the top panel and increased the volume of contained water from 1.3 *L* to 1.7 *L*. Other minor improvements in manufacturing techniques were included in the new design, but the heat path design from the top pad through the clip to the coolant pipes remained the same, to ensure a reliable comparison between the new eight-pipe device and the older six-pipe unit. Other currently recognized engineering concerns such as component deterioration within the highly corrosive animal confinement environment and closed system coolant recirculation were not considered during this experimental program. Overall, this study was undertaken to determine the difference in heat transfer performance between these two designs through bench testing. Specifically, the objectives of this series of experimental testing were to:

- measure thermal traces and coolant flow measurements for the six-pipe and eight-pipe designs at the nominal 2.6 *L/min* for a steady flow and one *min* on /one *min* off (two *min* total), one *min* on / two *min* off (three *min* total), and one *min* on / three *min* off (four *min* total) intermittent flows;
- determine the Biot number, heat rejection, top plate temperature drop, top plate temperature variance, heat exchanger efficiency, coolant effectiveness for the various test conditions (heat transfer rate per mass flowrate of coolant), and the conductive and convective coefficients for heat transfer of the devices operating at steady state; and
- perform a statistical comparison of heat exchanger metrics between the six-pipe and eight-pipe designs at steady state flow with the following statistical test hypotheses:

Heat Rejection:	H_0: $\dot{Q}_6 = \dot{Q}_8$	H_1: $\dot{Q}_6 < \dot{Q}_8$
Efficiency:	H_0: $\varepsilon_6 = \varepsilon_8$	H_1: $\varepsilon_6 < \varepsilon_8$
Effectiveness:	H_0: $\Psi_6 = \Psi_8$	H_1: $\Psi_6 < \Psi_8$

MATERIALS AND METHODOLOGY

Testing Apparatus

All bench testing comparisons between the six and eight-pipe designs were conducted using the same experimental protocols. From an experimental design perspective, the original six-pipe device represented the 'control' treatment, and the eight-pipe design was the comparison treatment. Each prototype had the same upper surface area, 0.74m^2, although the actively cooled

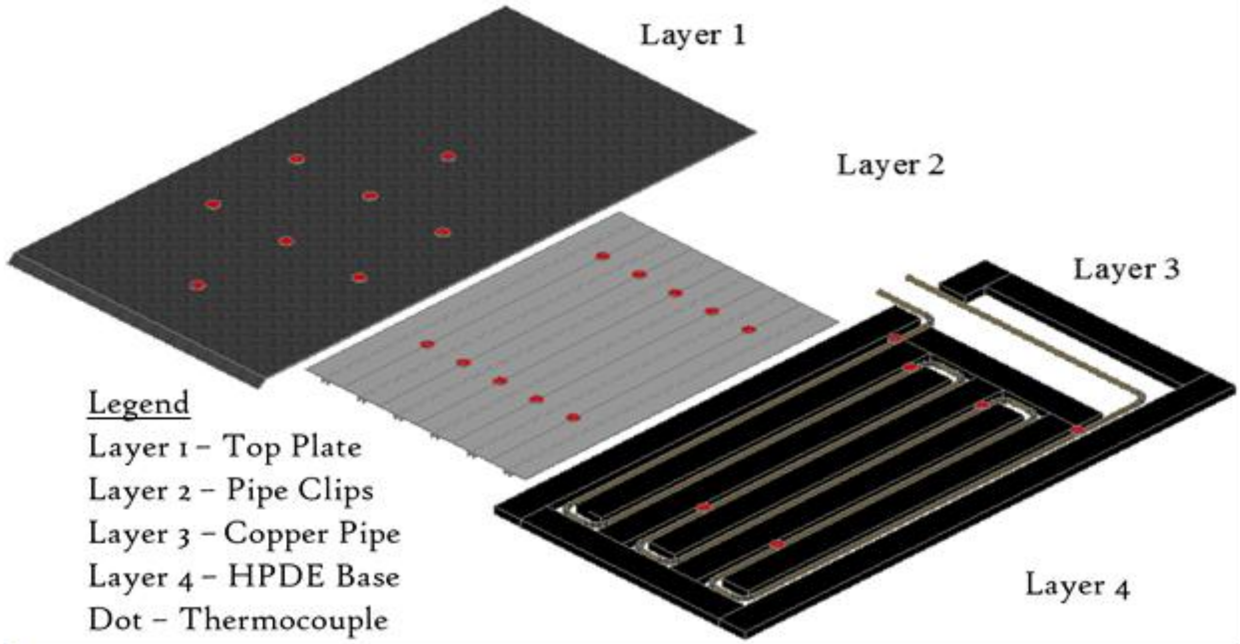

Fig. 1.Exploded view of the first prototype (six-pipe) Purdue hog cooling pad showing the four basic layers of construction, plus the thermocouple locations used in bench testing the device for heat capacity and heat transfer.

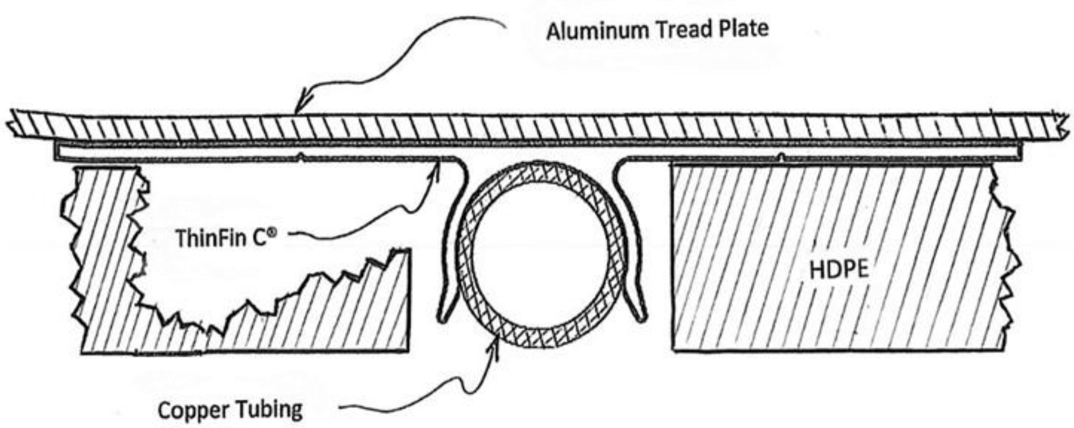

Fig. 2. Cross-section of the Purdue hog cooling pad showing the four basic layers of construction.

area was only 0.56 m^2. The bench test apparatus (the 'virtual pig') was set to the same source temperature, 40^oC, as previous experiments. Bench testing of the six-pipe first prototype cooling pad design, shown in Cabezon et al. (2017a), and the eight-pipe second prototype, shown in figure 3, was conducted at the Purdue ABE Department's ADM research facility. The exterior of the second prototype Purdue hog cooling pad is shown on the left side of figure 3, and from the outside, it looks nearly identical to the first prototype pictured in Cabezon et al. (2017a). The right side of figure 3 shows the internal coil configuration of the second prototype. Coolant pipes were placed at 7.5 *cm* intervals across the width of the second prototype device, while they were distributed at 10.0 *cm* intervals on the first prototype. Coils were placed lengthwise to extend from approximately the rear of the animal to the shoulders. The top plate under the animal's head was not actively cooled, as the length of the coolant coils did not extend that far forward on the animal. The depth-wise cross-

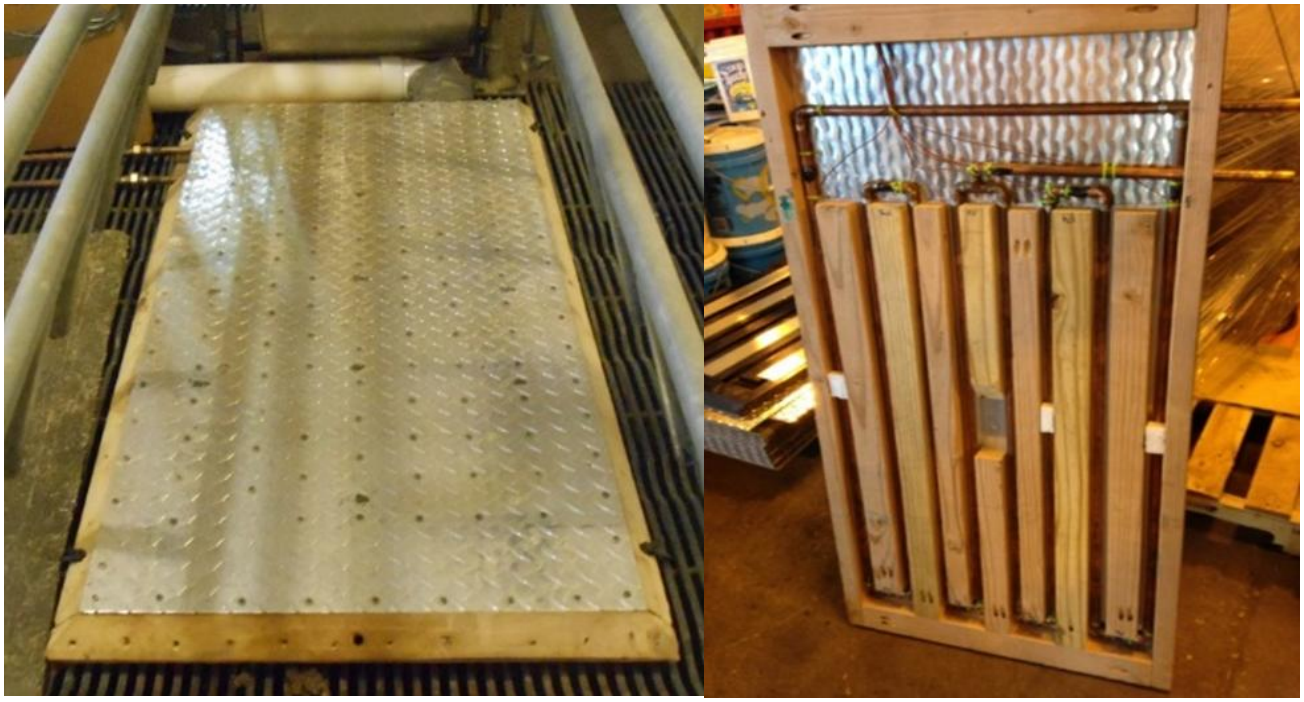

Fig.3. Exterior of the second prototype (eight-pipe) Purdue hog cooling pad installed in a farrowing crate at the Purdue Animal Sciences Research and Education Center (left), and the operational interior of the same unit showing cooling coil layout (right).

sectional profile of the two prototypes remained identical, with the active portions consisting of a 1/8*”* nominal 6061-T6 aluminum tread plate top, ½*”*nominal copper pipe (wall thickness – 0.125 *cm*), and a specialty aluminum clip connecting the two components. Construction grade lumber comprised the base of the prototype units used in bench testing and provided the environmental heat path insulation, although the base was converted to high density polyethylene (HDPE) in later subsequent versions of the cooling pad. The completed units have an active cooling pad area 61 *cm* wide by 91 *cm* long and were fitted with identical system connections to enable testing with the existing bench apparatus.

The bench testing apparatus was fully described in detail in Cabezon et al. (2018). Concisely, it consists of a wooden platform for the cooling pad. The virtual pig was constructed from a wooden frame and EPDM membrane, and it was set upon the cooling pad and filled with water. The virtual pig was approximately the same size as a small parity one sow or modest gilt. The testing apparatus in no way attempted to duplicate the complexities of a live animal. It merely represented a constant temperature source for laboratory experimentation. The water in the virtual pig box was heated using a swimming pool heater, which was held to a constant temperature. For bench testing, coolant water was provided by the university potable water system, kept in a separate supply tank maintained at a constant temperature using insulation and periodic resupply, and delivered to the experiment through a separate, recirculating, constant pressure circuit. During bench testing, coolant water was dumped into the university sanitary waste drain after a single use.

Both prototype pads were instrumented identically, with Type K thermocouples from REOTEMP® (San Diego, CA) distributed across on the upper surface of the top plate, at the plate and pipe clip interface, and on the coolant pipe. Figure 4 shows the placement of the thermocouples for the top plate and pipe positions. An Omega DAQ-56® (Norwalk, CN) PC computer-based automatic data acquisition (ADA) system recorded the pad’s thermal response to testing, as well as the virtual pig and ambient conditions temperatures. Coolant flow was measured with a Blancett® B110-500-1/2 (Racine, WI) turbine flow meter connected to the same Omega® ADA system.

Experimental Procedure

The experiments reported here follow an experimental progression from the no heat source and stored energy investigation detailed in Cabezon et al. (2017a) to the constant temperature source scenario described in Cabezon et al. (2018). As such, the testing procedures were similar. The coolant water supply system was started, and the outlet back pressure was set to generate the required 2.6 *L/min* (2.6 *kg/min*) of flow. The coolant supply system was then turned-off. The virtual pig tank heater was started and allowed to equilibrate to the set temperature of $40^{o}C$. Data analysis from Cabezon et al.

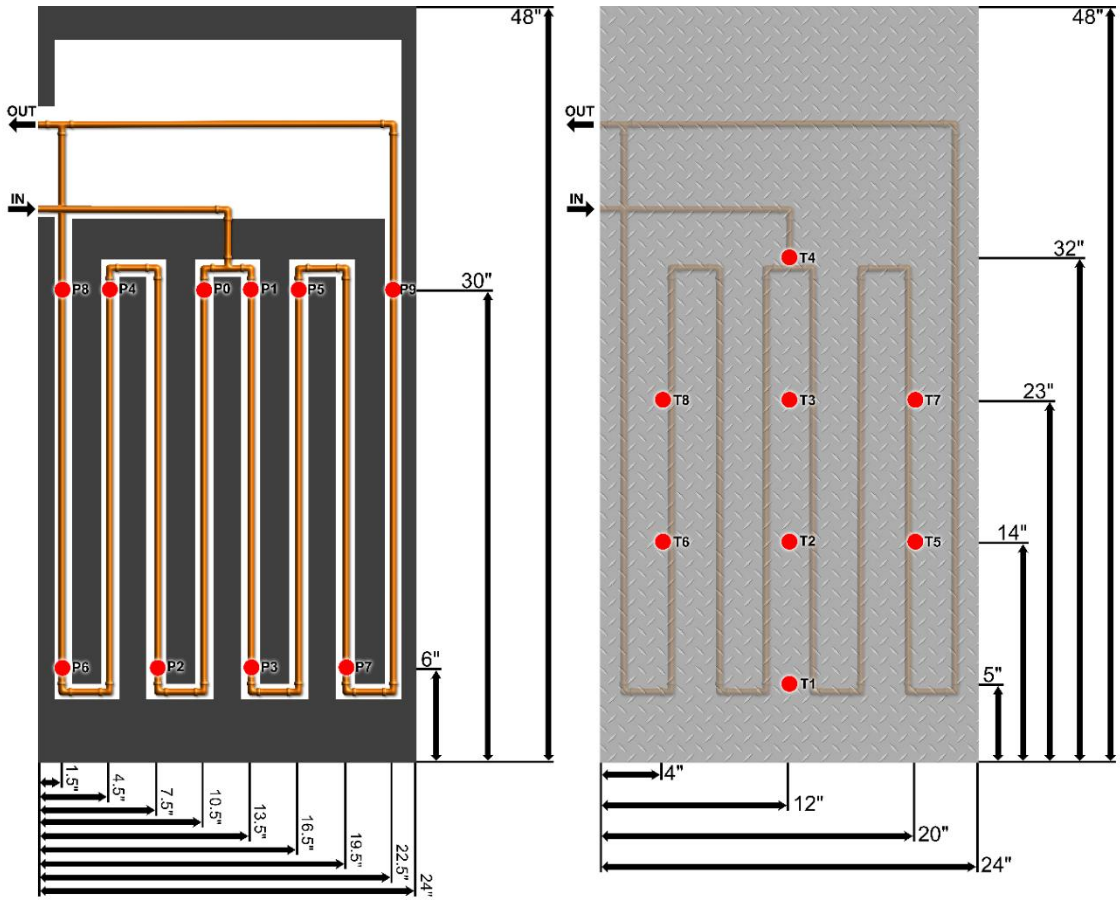

Fig. 4. Pipe thermocouple positions used for the bench testing the second prototype (eight-pipe) Purdue hog cooling pad.

(2018) showed the original 12 *h* system equilibration times were unnecessary for good results. It took 6 *h* for the cooling pad to reach a steady-state thermal equilibrium with the virtual pig, while only 1 *h* was required between steady-state runs to re-establish an elevated temperature within the coolant pipes.

The experiments were conducted by beginning the data acquisition and then establishing the coolant flow, already set for the predetermined flowrate. The ADA system recorded a time averaged value on all data channels every 6 *s* during active experimentation. Twenty to thirty minutes was generally sufficient to establish a new steady state thermal equilibrium in the first six-pipe prototype device (Cabezon et al., 2017a). The intermittent flow protocols were controlled manually using a ball valve on the coolant supply system triggered by the lapsed time value displayed by the ADA system. Two repetitions of each experimental treatment combination were conducted, as it had been previously demonstrated by Cabezon et al. (2017a) that variance between runs was minimal.

Instrumentation Calibration

Thermocouple and turbine flow meter calibration testing was performed prior to experimental measurements in Cabezon et al. (2017a, 2018), along with verification testing following the experimentation. Those calibration results were essentially identical and indicated that the instrumentation and ADA system had remained constant throughout the testing series. The series of experiments reported here occurred several months later. Thermocouple and turbine flow meter calibration testing were repeated prior to and following the new series of experimentation to ensure that no variation in the system occurred during storage or over the course of the investigation.

A bad thermocouple in position P1 on the six-pipe device was replaced. Temperature calibration was performed using ambient conditions, boiling water, and an ice bath. ADA channel readings were successfully checked against a handheld thermocouple reader for correspondence at an ambient temperature of $27^{o}C$. All sensors were within the expected accuracy during the calibration testing. A dedicated ambient condition thermocouple and an ice bath thermocouple were additionally checked in an ice water bath and boiling water. These two sensors recorded the expected temperatures during the appropriate phases of the testing. An indication of consistency across the time frame of the experiment was provided by the post-test and pre-test offset of reading value differences between the ADA system and the reading with a certified, handheld Tegam® 817A (Geneva, OH) digital thermocouple reader adjusted for Type K sensors. A summary of averaged offsets is shown in table 1, and they are comparable to the previous results. The temperature calibration testing results showed very good consistency through time with average offsets of less than 0.1^oC for the bench testing of the six-pipe and eight-pipe devices, respectively. The offset values were grouped tightly with all standard deviations approximately 0.1^oC.

Scatter in the ADA readings was determined through the calculation of variance. ADA readings during calibration were taken every second for a period of roughly 30 *min*. Removing the data sections where the handheld reader was in use, provided over 1500 individual readings per channel which were used to calculate standard deviation and variance. Average variances on specific channels during calibration were also small, ranging from 0.1$^oC^2$ to 0.2$^oC^2$. Although a few of the thermocouples exhibited slightly wider variances, all signals were deemed within acceptable limits. The published accuracy limit of the K type sensor is ±1.1^oC (REOTEMP Instrument Corporation, 2011).

The coolant flowrate calibration also utilized an experimental procedure to determine the accuracy of the turbine flow meter sensor output. Although the coolant plumbing was slightly different than in previous experimental efforts, the results of the current testing were consistent with the earlier work and are shown in figure 5. There were no significant differences noted between pre-test and post-test data, and the sensor performed similarly between the six-pipe and eight-pipe device testing. The published accuracy of the Blancett® flow meter is ±1% of the flow reading. The uncertainty shown in figure 5 represents the 2σ error band across the full range calibration from the Cabezon et al. (2017a) testing and is the equivalent to ±0.33 *L/min*.

Unlike previous studies, the testing conducted in this study used only a single average operational coolant flowrate of 2.6 *L/min*. The internal diameter of the ½*"* nominal cooling coil tubing is 1.34 *cm*. The mass flowrate and one-dimensional velocity were 0.05 *kg/s* and 0.34 *m/s*, respectively. The Reynolds Number (**Re**) for the test condition was 4370. The coolant flow at this level was turbulent and within the range of previous testing. The flowrate data collection system performed without incident during the testing periods. The combined experimental uncertainty in heat transfer with this instrumentation would be the product of the uncertainty in both instruments at the normal temperature and pressure (NTP: 20^oC, 101.325 *kPa*) specific heat of water (4.18 *kJ/kg·K*). The range of the heat transfer uncertainty was previously determined to be ±4.6 *W* (Cabezon et al., 2017a; 2018).

Previous testing indicated that the source heater on the virtual pig maintained a temperature within the accuracy limits of its circuitry, ±0.5^oC. Coolant temperature remained constant throughout the testing at a nominal 18^oC. To establish a comparison between the six-pipe coolant coil design and the eight-pipe design, a partial repetition of the testing series completed in Cabezon et al. (2018) was repeated using the second prototype design. Flowrates between 1.5 and 3.5 *L/min* were previously established as being the range of interest, and for the current experimentation, steady-state and transient responses were limited to a single, steady 2.6 *L/min* coolant flow. Intermittent mode tests were limited to one *min* on / one *min* off, one *min* on / two *min* off, and one *min* on / three *min* off cycles, also at the 2.6 *L/min* flowrate, when coolant circulation was active.

Data Analysis Protocols

Analysis of the collected data for the two prototype cooling pad devices was performed in a manner consistent with earlier work of Cabezon et al. (2017a, 2018). As recommended in Incropera & DeWitt (1981), temperature data was converted to a non-dimensional temperature difference, Θ_η, to determine the time constant of the devices. Temperatures at specific sensors under analysis over time within an experiment are normalized using:

$$\Theta_\eta = \frac{T - T_\infty}{T_i - T_\infty} \quad (1)$$

where: T is the time varying temperature of the sensor of interest (^oC);
T_∞ is the final temperature of the sensor of interest (^oC); and
T_i is the initial temperature of the sensor of interest (^oC).

The time constant for a temperature changing component is equal to a period of time required for the non-dimensional temperature to drop to a value of 0.368 (Incropera & DeWitt, 1981; Lewis et al., 2004). The top plate temperature drop, ΔT_t(^oC), was of interest in analysis and is calculated as:

$$\Delta T_t = T_{ti} - T_t \quad (2)$$

where: T_{ti} is the initial top plate temperature (^oC); and
T_t is the time varying top plate temperature (^oC).

Heat rejection, $\dot{Q}$ (*kW*), from the virtual pig is experimentally determined by:

$$\dot{Q} = \dot{m} \cdot c_p \cdot (T_{in} - T_{out}) \quad (3)$$

where: $\dot{m}$ is the mass flowrate of coolant (*kg/s*);
c_p is the specific heat of the coolant (*kJ/kg·^oC*);
T_{in} is the inlet temperature of the coolant (^oC); and
T_{out} is the outlet temperature of coolant (^oC).

Biot number, **Bi**, is a non-dimensional parameter used to examine conductive / convective heat transfer devices and is particularly useful in the analysis of extended surface heat transfer devices (Incropera & DeWitt, 1981) The only assumption regarding the use of Biot number under these circumstances is that all heat transfer passes through the path under analysis, an assumption

Table 1. Average thermocouple validation results for the Purdue hog cooling pad coil design bench testing calibration protocol, showing offset between the automatic data acquisition system and a handheld reader and the variance during the pre-test and post-test calibration at an ambient room temperature (27°C).

	6-Pipe Cooling Pad			**8-Pipe Cooling Pad**		
Thermocouple	Post to Pre- Test Offset (*°C*)	Pre-Test Variance (*°C²*)	Post-Test Variance (*°C²*)	Post to Pre-Test Offset (*°C*)	Pre-Test Variance (*°C²*)	Post-Test Variance (*°C²*)
Top	0.0	0.1	0.2	0.1	0.1	0.1
Pipe	0.1	0.1	0.1	0.1	0.1	0.1

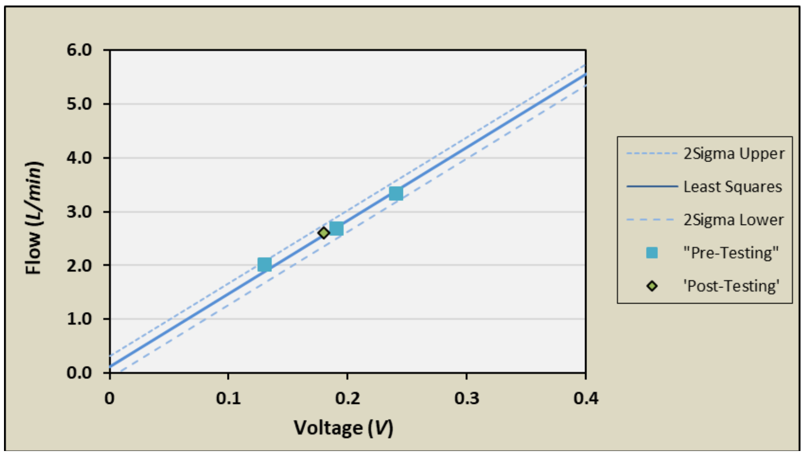

Figure 5. Pre and post-testing calibration data for the Blancett® turbine flow meter, plus linear regression and error bars, for the coolant flow measurement system used in the bench testing apparatus for the Purdue hog cooling pads during the coil design testing.

shown to be reasonable accurate in the present physical scenario (Cabezon et al., 2017a; 2018). Where all temperature measurements are made according to the same scale, the **Bi** definition may be algebraically rearranged and determined in this situation experimentally by:

$$\mathbf{Bi} = \frac{T_{top} - T_{pipe}}{T_{pipe} - T_{out}} \qquad (4)$$

where: T_{top} is the temperature of the top plate (*°C*);
T_{pipe} is the temperature of the coolant pipe (*°C*); and
T_{out} is the outlet temperature of coolant (*°C*).

Heat exchanger efficiency, ε, is a metric consisting of the ratio of actual heat transfer divided by the potential for heat transfer and is used to compare different heat transfer devices or similar devices operating under different conditions (Incropera & DeWitt, 1981). Assuming that the maximum potential in this situation is given by the differential between the top plate and coolant inlet temperature and all temperature measurements are made according to the same scale, heat exchanger efficiency for the hog cooling pad is defined as:

$$\varepsilon = \frac{T_{out} - T_{in}}{T_{top} - T_{in}} \qquad (5)$$

where: T_{out} is the outlet temperature of coolant (*°C*);
T_{in} is the inlet temperature of the coolant (*°C*); and
T_{top} is the temperature of the top plate (*°C*).

Effectiveness of a heat transfer device, Ψ (*kJ/L*), for this specific analysis is defined as the ratio of energy rejection rate to coolant flowrate:

$$\Psi = \frac{\dot{Q}}{\dot{v}} \qquad (6)$$

where: $\dot{Q}$ is the heat rejection rate (*kW*); and
$\dot{v}$ is the volumetric flowrate of the coolant (*L/s*).

The conductive heat transfer coefficient, k (*W/m·°C*), for generalized heat transfer analysis is defined by the rate of heat transfer per area and thermal gradient:

$$k = \frac{\dot{Q}}{A \cdot \frac{dT}{dx}} \quad (7)$$

where: $\dot{Q}$ is the heat rejection rate (*W*);
A is the effective area for heat transfer; and
$\frac{dT}{dx}$ is the solid material temperature gradient in the direction of heat transfer.

The convective heat transfer coefficient, h ($kW/m^2 \cdot {}^oC$), for generalized heat transfer analysis is defined by the rate of heat transfer per area and thermal driving potential:

$$h = \frac{\dot{Q}}{A \cdot (T - T_\infty)} \quad (8)$$

where: $\dot{Q}$ is the heat rejection rate (*kW*);
A is the effective area for heat transfer;
T is the temperature entering the convective system; and
T_∞ is the exit temperature of the convective fluid.

Statistical analysis was used to compare the hypotheses for heat rejection, heat exchanger efficiency, and effectiveness of the eight-pipe versus six-pipe designs in the steady flow condition. The two different designs represented the two different treatment combinations, or independent variables in the statistical determination. The measured experimental data and the calculated metrics under comparison are continuous and represented the dependent variables. The individual data readings collected in time were mutually independent in these experiments, because no single data collection event affected any other data collection event. In repeated measure experimental designs, sequential measurements are considered independent, if no information about a preceding measurement is used in the collection of a successive measurement (Field, 2009). Fifty data points from the end of the steady state portion of the experiments were selected as comparison data sets. SPSS® was used to perform the statistical computations. The normality of the data sets was confirmed with the Shapiro-Wilk test. The homogeneity of variances between sets were checked with Levene's test for the equality of variances (f-test), and then based upon these results, appropriate t-tests were conducted for discrimination between the means of the six-pipe and eight-pipe designs (Field, 2009). A 95% confidence interval was selected for the statistical inferences.

RESULTS AND DISCUSSION

Temporal Temperature Traces for Continuous Coolant Operation

Figure 6 shows that the temperature change over time between the two designs exhibited the same general temporal characteristics as predicted by Newton's Law of Cooling and shown by the first prototype in the original bench testing (Cabezon et al., 2017a). Curves for the top plate and the cooling pipes both exhibited a logarithmic decay toward a steady value, although the actual devices lag the ideal value slightly during the experiments, indicating data collected from a real device instead of a predicted response from an ideal one. The average error between the predicted values and the experimental values over the course of the individual experimental runs was small (< 0.1^oC), with the largest values near the beginning of each experiment. Additionally, the standard deviation of the errors was small, between 0.20^oC and 0.36^oC.

Figure 7 illustrates the differences in top plate and pipe temperatures for the two devices under identical operational conditions from a no coolant flow condition to a steady state continuous coolant flow at 2.6 *L/min*. Data displayed on the left side of figure 7 represents the averaged temperature readings from the two measured layers within the device every 6 *s*. The temperature traces were switched into the corresponding non-dimensional temperature difference, Θ_η, based upon the initial top plate temperature and the driving potential of the coolant inlet, and these values are also shown on the right side of figure 7. Non-dimensional temperature is an engineering metric useful in heat exchanger analysis, because it normalizes the temperature against the potential existing within the design under the existing conditions. The eight-pipe device clearly exhibits a lower equilibrium temperature on the top of the top plate. The temperature reduction on the top plate between the initial condition and the steady state condition, defined by equation 2, was 8.1^oC for the eight-pipe device and 5.5^oC for the six-pipe. Less pronounced, but clearly indicated, is the increase in pipe temperature of the eight-pipe design, with an increased temperature gain of 1.5^oC compared to the six-pipe. Between the two devices, the eight-pipe design has a smaller total internal spatial gradient in temperature, implying a heat transfer device with less overall resistivity to the passage of heat. A lower top plate temperature implies an improved ability to move heat through the device. This can only be attributed to the increased active coil length, coolant volume, and internal contact area within the pad, as all other details of the prototypes' construction were identical.

Pad Performance Metrics for Continuous Coolant Operation

Utilizing the non-dimensional temperatures and the accepted 36.8% final value (Lewis et al., 2004), the time constants for the top plate temperature of the six-pipe and eight-pipe designs at the 2.6 *L/min* coolant flowrate were 156 *s* and 100 *s*, respectively. The time constants for the pipe temperatures showed less differential at 60 *s* and 50 *s*, respectively. The eight-pipe design clearly

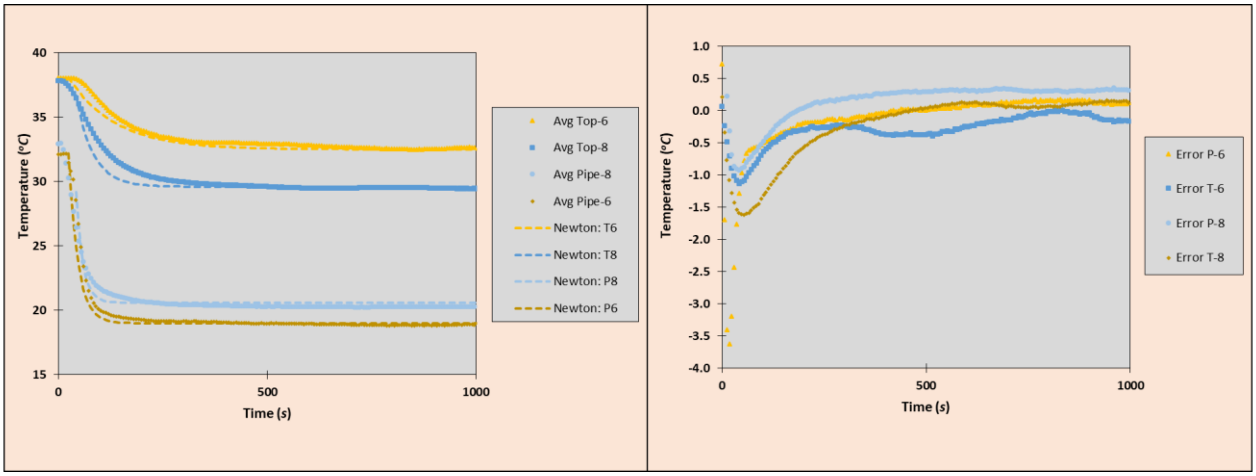

Fig. 6. Comparison of the average temperature traces for the top plate (8 sensors) and coolant pipe (9 sensors) thermocouples with Newton's Law of Cooling for the six-pipe and eight-pipe designs of the Purdue hog cooling pad during steady 2.6 L min^{-1} coolant flow testing at 6 s intervals on the left and the error between the prediction and data on the right.

moved to its steady state temperature faster than the six-pipe design, provided a slightly cooler top plate temperature for the animal and improved the heat transfer, and showed a more effective use of the coolant. Additionally, the Biot Number, calculated by equation 4, for the eight-pipe design was 3.5, compared to a 12.2 for the six-pipe design, indicating a better balance between the conductive and convective sections of the device. Table 2 shows **Bi,** the top plate temperature drop, the steady state temperatures at the top plate and coolant pipe, the standard deviation in temperature readings on the top plate for the two designs, and the conductive and convective heat transfer coefficients, based upon an average of the last 50 data points. The operational Biot number results for the two designs clearly showed that although both devices were convection-dominated as heat exchangers, the eight-pipe unit was more balanced than the six by a significant degree.

The heat exchanger efficiency, ε, calculated with equation 5, shows that the eight-pipe cooling pad is slightly more efficient than the six-pipe, but both units are fairly inefficient compared with heat exchangers used in power generation or HVAC systems. However, this is perfectly acceptable in this situation, because these units will be in direct contact with a live animal. To achieve a higher efficiency, the top plate would need to drop more in temperature, and this would not be a productive strategy, as the animals would find the plate uncomfortably cold and not use it. Temperatures in the eight-pipe design did not have as steep a gradient and are spread more uniformly across the cross section, as is desirable. Unfortunately, the drop in top plate temperature in the eight-pipe design was slightly more than in the six-pipe, although not so much as to become unusable. The top plate temperature is a delicate balance between animal comfort and heat exchanger efficiency. These devices are not efficient as heat exchangers, because this would be highly undesirable from a mammal's perspective. A low skin / plate contact temperature inhibits the animal from laying on the device (Maskal et al., 2018). Anecdotal evidence suggests that animals subjected to excessively cool contact temperatures alter their behavior and begin to roll from side-to-side and then stand. When triggered by cold skin contact temperature, they begin to shunt blood away from the skin to the deep body, regardless of ambient conditions (Johnson et al., 2016; Quiniou & Noblet, 1999; Silva et al., 2006). The design and operational protocols for the Purdue cooling pad are designed to prevent over-chilling of the animal. Variance in the top plate temperatures was smaller for the eight-pipe design, demonstrating that the eight-pipe created a more uniform temperature profile on the top surface.

Heat Rejection, Coolant Use Effectiveness, and Heat Transfer Coefficients for Continuous Coolant Operation

Figure 8 shows the temporal trace of the heat rejection by the cooling pad and the effectiveness of the unit's use of coolant in continuous operation. The noise within the displayed metrics is the result of the instantaneous calculation of both $\dot{Q}$and ψ from the physically measured temperatures and flowrates at each 6 *s* interval. The data show that heat quantity removed through the coolant system was slightly enhanced by the eight-pipe design, but the eight-pipe unit also utilized the coolant in a slightly more conservative manner, by producing an improved coolant effectiveness. Both designs show similar transient responses, with heat rejection rates and effectiveness metrics spiking to fairly high values, before rapidly settling into a modest asymptotic approach to the steady state. The steady state point was determined by looking at the slope of the last fifty readings and adding fifty points to the time interval where the slope dropped to 0.001% of the initial value. This occurred for all metrics of interest by 2500 *s*. Fifty readings were selected for examination as a sample size sufficient to dampen the natural noise inthe data. The eight-pipe design pumped an average of 305 *W* away from the virtual pig in the steady state, and

Table 2. Variation of final Biot number, heat exchanger effectiveness, operational top plate temperature drop, top plate temperature, top plate temperature standard deviation, pipe temperature, and conductive and convective heat transfer coefficients with six-pipe and eight-pipe coil designs during bench testing of the Purdue hog cooling panel operating at a steady 2.6 L min^{-1} coolant flow.

Number of Passes	Bi	ε	Top Temp.(°C)	ΔT_{Top} (°C)	$SD_{(Top)}$ (°C)	Pipe Temp.(°C)	k (W/m·°C)	h (kW/m²·°C)
6-Pipe Cooling Pad	12.2	0.08	32.5	5.5	2.0	19.1	0.223	9.0
8-Pipe Cooling Pad	3.5	0.14	29.7	8.2	1.4	20.6	0.387	10.1

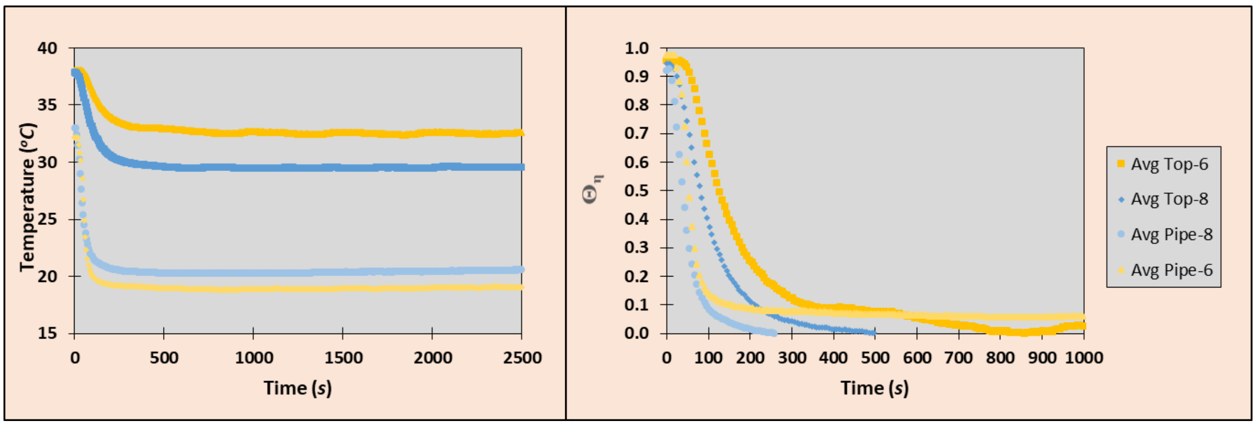

Figure 7. Comparison of the average temperature traces for the top plate (8 sensors) and coolant pipe (9 sensors) thermocouples for the six-pipe and eight-pipe designs of the Purdue hog cooling pad during steady 2.6 L min^{-1} coolant flow testing at 6 s intervals on the left and the same data in non-dimensional temperature (Equation #1) format on the right.

the six-piperemoved 262 *W*.

Although the heat rejection values shown in figure8 may seem a little high given the nominal level of heat production in adult swine, around 400-500 *W*, it is worth noting that these are continuous heat rejection capacities. Overheated animals will temporarily require larger heat transfer rates to cool-down, and lactating sows can produce over four times the heat of a similar sized animal fed a maintenance diet (Cabezon et al., 2017d; Johnson et al., 2019; Noblet et al., 1993; National Research Council, 2012). The control system protocols on production units will prevent the devices from over-chilling the animals. The eight-pipe design showed an improvement in efficiency over the six-pipe, 13.7% vs. 8.3%. However, this traditional design metric is not as important a driving criterion for this device for heat exchangers in mechanical applications, because a large drop in temperature across the unit, while raising efficiency, is not necessarily desirable for cooling animals. From an effectiveness perspective, the eight-pipe design could move 5.84 *kJ/L*, while the six-pipe could only transfer 5.03 *kJ/L*.

A statistical analysis within SPSS® was undertaken to determine if the eight-pipe design performed better than the six-pipe design using the last 50 data points as an approximation for the steady state. The statistical data set summary results from SPSS® are available in table 3 and preliminarily show superior performance in the eight-pipe device. A Shapiro-Wilk test for normality was performed to establish whether a normal distribution of the dependent variables for each pad design was present. A 95% confidence interval was selected for all of the discrimination testing. Table 3 results demonstrate that show normality can be assumed in all cases. The statistical analysis results from SPSS® to check the comparison test propositions are shown in table 4. Levene's test for equality of variances was used preliminarily to determine whether equal variances could be assumed in the t-test. The data set testing showed that equal variances could be assumed for rejection, but not for efficiency and effectiveness, since $p < 0.05$. The final t-tests, which determine results for significance, show that each test had a $p < 0.01$. These results demonstrate that the eight-pipe pad's rejection rate, efficiency, and effectiveness were statistically better than the six-pipe pad.

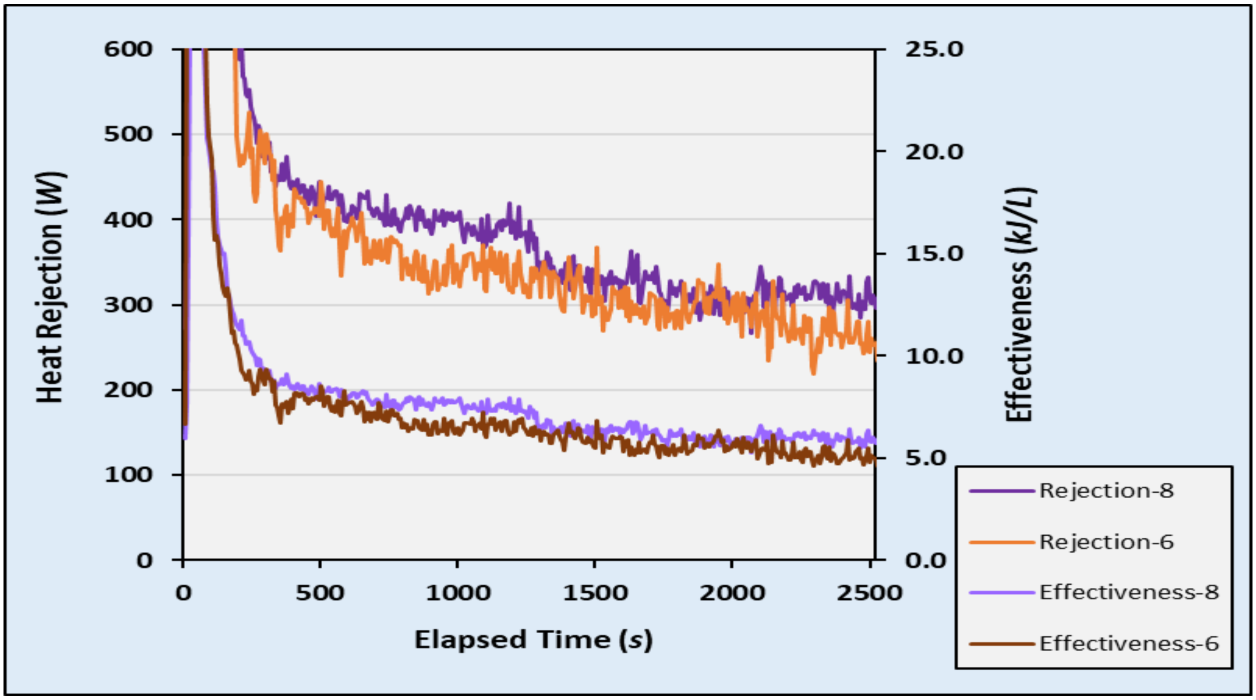

Fig. 8. Variation of heat rejection (equation 3) and effectiveness (equation 6) with six-pipe and eight-pipe coil designs during bench testing of the Purdue hog cooling panel at a steady 2.6 L min^{-1} coolant flow.

Table 3. Statistical mean summary results from SPSS® for heat removal (W), efficiency, and coolant effectiveness (kJ/L) data between the six and eight-pipe coil designs of the Purdue hog cooling panel operating at a steady 2.6 L min^{-1} coolant flow.

	Pad	N/df	Mean	Std. Deviation	Std. Error Mean	Shapiro-Wilk (w)	Sig. (p)
Removal	6-Pipe	50	262	16.4	2.31	0.98	0.52
	8-Pipe	50	305	11.4	1.61	0.97	0.23
Efficiency	6-Pipe	50	0.1	<0.01	<0.01	0.97	0.33
	8-Pipe	50	0.1	<0.01	<0.01	0.97	0.26
Effectiveness	6-Pipe	50	5.0	.28	.04	0.98	0.49
	8-Pipe	50	5.8	.20	.03	0.97	0.29

The conductive and convective heat transfer coefficients for the sow cooling pad can be calculated on a macroscopic basis from the physical characteristics of the units and the experimental data. The cooling coils do not engage with the full top plate area of the devices, as the front area, near the sow's head, is not actively cooled. However, the active area is the same, 0.56 m^2. In the conductive portion of the device, the heat moving from the top plate to the coolant coils passes linearly through 0.0064 *m*. Using equation 7, the average conductive heat transfer coefficient for the steady state condition in the six-pipe pad is 0.223 $W/m{\cdot}^{o}C$. The calculation for the eight-pipe pad provides a value of 0.387 $W/m{\cdot}^{o}C$. This is a 73% improvement in the conductive heat transfer coefficient of the eight-pipe device over the six-pipe unit. In the convective portion of the device, the heat moving into the coolant coils passes through a different area in each design. Using the geometry of the coolant pipe and assuming that 67% of the pipe circumference is engaged with the specialty clip, an effective area of 0.26 m^2 is available for the six-pipe device, and 0.34 m^2 is usable inin the eight-pipe. Using equation 8, the average

Table 4. SPSS® significance results for heat removal (W), efficiency, and coolant effectiveness (kJ/L) discrimination t-tests between the six and eight-pipe coolant coil designs of the Purdue hog cooling panel operating at a steady 2.6 L min^{-1} coolant flow.

	Levene (f)	Sig. (p)	Levene (f) Results	t	df	Sig. (1-tailed)
Removal	3.25	0.07	Equal variances assumed	-15.4	98	< 0.01
Efficiency	4.48	0.04	Equal variances not assumed	-50.8	93	< 0.01
Effectiveness	4.68	0.03	Equal variances not assumed	-16.6	89	< 0.01

convective heat transfer coefficient for the steady state condition in the six-pipe pad is 10.1 $kW/m^2 \cdot {}^oC$. The calculation for the eight-pipe pad provides a value of 9.0 $kW/m^2 \cdot {}^oC$. This is an 11% drop in the convective heat transfer coefficient of the eight-pipe device over the six-pipe unit. These differences are attributable only to the denser packing of the coolant coils in the eight-pipe design. Even though the convective coefficient drops in the eight-pipe design, the improvement in conductive coefficient is sufficient to improve the overall heat transfer in the aggregate device. These values corroborate the findings with Biot number.

Temporal Temperature Traces for Intermittent Coolant Operation

The researchers previously had significant success utilizing intermittent coolant operational cycles within the cooling pads. Because of the higher Biot number and modest time constant present in the overall device, continuous coolant flow wouldlikely not be useful or economicalfor cooling swine. Utilizing an intermittent flow allowed heat from the animal to further warm the coolant held within the coils before exhausting it. When placed into commercial operation where system coolant will likely be recycled, this will provide a further improvement in coolant utilization efficiency, contributing to the overall operational sustainability of the device. Three intermittent coolant flow cycle modes were investigated during this study. Coolant flows, when operational and on, remained at the same 2.6 *L/min* used previously. Intermittent flow cycles of one *min* on / one *min* off, one *min* on / two *min* off, and one *min* on / three *min* off were tested. The operational period was selected to ensure that at least a volume and a half of fresh coolant would be pushed through the full coil length in the eight-pipe design at the 2.6 *L/min*flowrate. Figure 9 presents the non-dimensional temperature for the one *min* on / one *min* off and one *min* on / three *min* off intermittent coolant flow cycles. The impact of the coolant flow cycles is clearly visible in the temperature data.

Pad Performance Metrics for Intermittent Coolant Operation

The same general temperature pattern between the traces for the two different designs shown in the steady coolant flow data also emerged under intermittent flow. The eight-pipe configuration under all intermittent testing scenarios demonstrated lower top plate temperatures and higher pipe temperatures than the six-pipe design. Since the intermittent flow process only reached a pseudo-steady state, average **Bi** and device temperatures were determined across five full ‘on’ portions of the cycle. Table 5 shows the results of this analysis. The six-pipe design results were extremely well matched with the results in Cabezon et al. (2018). Under each individual coolant flowrate cycle, the eight-pipe design performed better than the six. Both designs became more convectively dominated as the cycle off period was increased, even though there was a slight rise in the average temperature of the top plate and pipe in the devices. The temperature drop of the top plate, ΔT_{top}, fell with increasing ‘off’ period length, which is a positive result. The variance of the top plate temperature across the sensors was reduced from the six-pipe design to the eight, as happened in the steady flow. This result demonstrated a more uniform top plate temperature with the eight-pipe design under intermittent coolant flow operating conditions.

Heat Rejection, Coolant Use Effectiveness, and Heat Transfer Coefficients for Intermittent Coolant Operation

The heat rejection rate and effectiveness of coolant use for the one min on / one min off and one min on / three min off coolant flow cycle are shown in figure 10. Instantaneous rejection for the eight-pipe device was greater than the six throughout the testing series, and the instantaneous effectiveness of the eight was likewise superior. However, the addition of the on / off cycles

Table 5. Variations in final Biot number, heat exchanger effectiveness, operational top plate temperature reduction, top plate temperature, top plate temperature variance, and pipe temperature with six-pipe and eight-pipe coil designs during bench testing of the Purdue hog cooling panel during 2.6 L min^{-1} coolant flowrate with intermittent variation in operational cycle.

	6-Pipe Cooling Pad					**8-Pipe Cooling Pad**				
Intermittent Cycle *(on min / off min)*	Bi	Top Temp. (^oC)	ΔT_{Top} (^oC)	T_{top}Variance ($^oC^2$)	Pipe Temp. (^oC)	Bi	Top Temp. (^oC)	ΔT_{Top} (^oC)	T_{top}Variance ($^oC^2$)	Pipe Temp. (^oC)
1 / 1	5.2	33.4	4.7	3.0	20.5	2.3	30.7	7.5	1.7	21.8
1 / 2	3.4	33.7	4.3	3.7	21.6	1.9	31.6	6.3	1.6	22.7
1 / 3	2.7	34.1	3.8	3.7	22.4	1.7	32.4	5.9	1.3	23.3

Table 6. Variation of cooling pad operational metrics with changes in flow rate cycle with six-pipe and eight-pipe coil designs during bench testing of the Purdue hog cooling panel during2.6 L min^{-1}intermittentcoolant flow operating cycles.

	6-Pipe Cooling Panel				**8-Pipe Cooling Panel**			
Intermittent Cycle *(on min / off min)*	Peak Reject. *(W)*	Average Reject. *(W)*	Efficiency	Average Operational Effectiveness *(kJ/L)*	Peak Reject. *(W)*	Average Reject. *(W)*	Efficiency	Average Operational Effectiveness *(kJ/L)*
1 / 1	445	187	0.14	8.3	651	285	0.27	11.3
1 / 2	681	174	0.20	11.7	892	240	0.34	16.0
1 / 3	878	148	0.23	13.6	1193	222	0.37	18.9

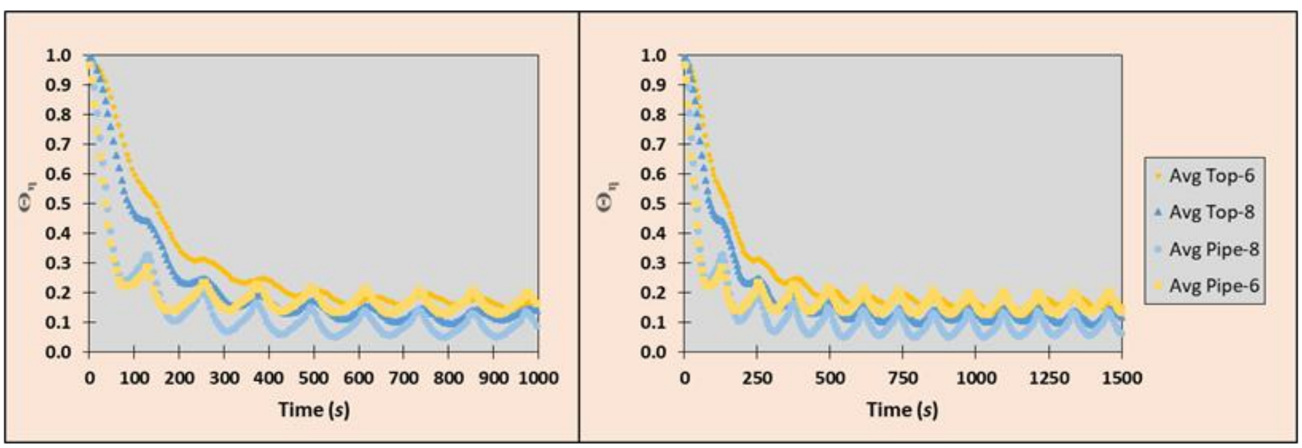

Fig. 9. Comparison of the non-dimensional temperature for the top plate and coolant pipe thermocouples for the six-pipe and eight-pipe designs of the Purdue hog cooling pad during intermittent one min on / one min off 2.6 L min^{-1} coolant flow on the left and intermittent one min on / three min off 2.6 L min^{-1} coolant flow on the right.

complicates numerical analyses, as the effectiveness becomes undefined when the unit is not running.

Table 6 provides comparison metrics that account for the intermittent nature of the coolant operation. Peak rejection was defined as the highest recorded amount of heat passed from the cooling pads. Average rejection was the total heat removed from the pads divided by the total time of operation. Effectiveness is only meaningful during operation, so average effectiveness was only calculated across the time when the unit was funtioning.

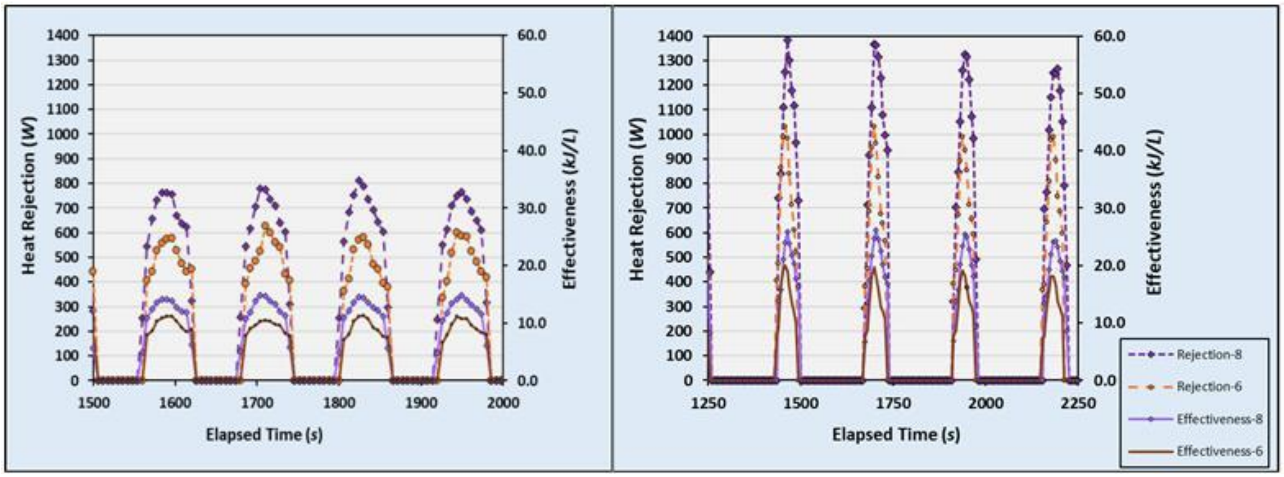

Fig. 10. Variation of heat rejection and effectiveness with six-pipe and eight-pipe coil design during bench testing of the Purdue hog cooling panel during one min on / one min off 2.6 L min-1intermittent coolant flow on the left and one min on / three min off 2.6 L min-1intermittent coolant flow on the right.

For boththe six-pipe and eight-pipe designs, the effectiveness of coolant use, ε, improved with the duration of the off portion of the intermittent cycle. In all cases, intermittent coolant flow was superior to continuous flow in the conservation of the coolant resource, and across the tested intermittent sequences, the eight-pipe device performed better than the six. Peak heat rejection rate of the devices increased with off period duration, and the eight-pipe exhibited higher rates than the six. In all cases, average heat rejection rate decreased with off period. Again, the eight-pipe device outperformed the six, but interestingly at the one *min* on / one *min* off intermittent mode, the eight-pipedevice outperformed itself in steady state, clearly demonstrating the efficiency of the intermittent mode of operation. This will be important for sustainability in commercial operation, where it will be likely that the coolant will be recycled and a coolant chiller system will be incorporated into the overall swine production facility, rather than dumping the coolant into the pit after one use as the researchers have done for simplicity. Heat exchanger efficiency also increased in both devices with intermittent operation. This result was beneficial and as expected, because the additional 'soak' time to absorb heat from the animal raised the exit temperature of the coolant.

CONCLUSIONS

Primary Inferences

Experimental testing demonstrated that the eight-pipe prototype was superior to the six-pipe device in several key metrics:

- The time constant of the eight-pipe unit was smaller than the six-pipe, indicating the capability of a faster reaction to changes in conditions.
- The Biot number of the eight-pipe prototype was decreased from that of the six-pipe, making it more dominated by the convective process and less limited by the conductive portion of the device.
- Heat exchanger efficiency also showed a slight, but significant, improvement for the eight-pipe design over the six-pipe.
- The eight-pipedevice removed more heat from the virtual pig than the six-pipe unit.
- The eight-pipedevice utilized coolant more effectively than the six-pipe unit.
- The eight-pipe prototype had a 73% higher conductive heat transfer coefficient than the six-pipe in steady state operation.
- The eight-pipe unit functioned better in the intermittent coolant flow operation than the six-pipe unit.

The current experimentation showed that the addition of two-passes of coolant coil improved the performance of the Purdue cooling pad. Further top plate temperature reductions are not desirable, because while a colder top plate would improve the heat exchanger efficiency, it would also run the risk of being objectionable to the animal, closing surface blood vessels and preventing laying behavior. The convective heat transfer coefficient of the eight-pipe device dropped slightly compared to the six-pipe, and Biot number for the eight-pipe device dropped modestly. However, it did not move enough to significantly change the character of the device away from a convective dominated device. The Biot number for the first pad had already indicated that the device was limited by the conductive portion of the heat transfer, and the enhancement in conductive heat transfer coefficient in the eight-pipe design was an improvement. Addingcapacity to the convective portion of the process wouldbe unnecessary and an unproductive use of resources. Additionally, the space beneath the aluminum top plate is limited in the current manifestation of the device.

Future Effort

This series of testing proved very useful in the development of the Purdue hog cooling pad. The eight-pipeprototype proved to function better in heat rejection, efficiency, and effectiveness under continuous coolant operation. Previously identified positive trends associated with increasing off periods during intermittent operation were shown also to exist in the eight-pipe design, and performance factors, such as peak heat rejection, efficiency, and effectiveness, increased. The eight-pipe design has been adopted and produced for further large-scale testing of the cooling pads at the Purdue Animal Sciences Research and Education Center. Current development efforts for the cooling pad design are now centered upon tightening the thermal reading band of error and improving the controller box functioning for the device. Future effort may investigate the utilization of alternative materials in the construction of the device, along with designing a unique device for boars. The current design precisely fits into a farrowing crate, so that the sows urinate and defecate behind the device. The anatomy of the boar will require a design modification to accommodate their ventral urinary tract.Results of initial heat transfer data from a preliminary live animal test with the second prototype should be published soon.

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REFERENCES

Bjerg, B., Brandt, P., Sorensen, K., Pedersen, P., & Zhang, G. (2019). Review of methods to mitigate heat stress among sows. *ASABE 2019 AIM - Boston.* St. Joseph: ASABE. doi:10.13031/aim.201900741

Cabezon, F. A., Schinckel, A. P., & Stwalley III, R. M. (2017a). Thermal capacity of hog-cooling pad. *Applied Engineering in Agriculture, 33*(6), 891-899. doi:10.13031/aea.12333

Cabezon, F. A., Schinckel, A. P., Richert, B. T., Peralta, W. A., & Gandarillas, M. (2017d). Application of models to estimate daily heat production of lactating sows. *Professional Animal Scientist, 33*(3), 357-362. doi:10.15232/pas.2016-01583

Cabezon, F. A., Schinckel, A. P., Smith, A. J., Marchant-Forde, J. N., Johnson, J. S., & Stwalley III, R. M. (2017c). Effect of floor cooling on late lactation sows under acute heat stress. *Livestock Science, 206*, 113-120. doi:10.1016/j.livsci.2017.10.017

Cabezon, F. A., Schinckel, A. P., Smith, A. J., Marchant-Forde, J. N., Johnson, J. S., & Stwalley III, R. M. (2017b). Initial evaluation of floor cooling on lactating sows under acute heat stress. *The Professional Animal Scientist, 33*(2), 254-260. doi:10.15232/pas.2016-01584

Cabezon, F. A., Schinckel, A. P., Stwalley, C. S., & Stwalley III, R. M. (2018). Heat transfer properties of hog cooling pad. *Transactions of the ASABE, 61*(5), 1693-1703. doi:10.13031/trans.12351

Field, A. (2009). *Discovering Statistics Using IBM SPSS.* London: Sage Publications, Ltd. doi:10.5555/2502692

Field, T., & Deneke, E. (2018). *Numerical analysis of thermal distribution for hog cooling pad design.* Term Paper, Purdue University, Mechanical Engineering 60800: Numerical Methods for Heat, Mass, and Momentum Transfer, West Lafayette.

Geis, E., Zumwalt, D., & Carter, J. (2015). *Sow Cooling Pad.* ASM 494-495 Capstone Experience Final Report, Purdue University, Agricultural & Biological Engineering Department, W. Lafayette.

Huynh, T. T., Aarnink, A. J., Spoolder, H. A., Verstegen, M. W., & Kemp, B. (2004). Effects of floor cooling during high ambient temperatures on the lying behavior and productivity of growing finishing pigs. *Transactions of ASAE, 47*(5), 1773-1782. doi:10.13031/2013.17620

Incropera, F., & DeWitt, D. (1981). *Fundamentals of Heat Transfer.* New York, NY: John Wiley & Sons.

Johnson, J. S., Sapkota, A., & Lay, Jr., D. C. (2016). Rapid cooling after acute hyperthermia alters intestinal morphology and increases the systemic inflammatory response in pigs. *Journal of Applied Physiology, 120*, 1249-1259.

Johnson, J. S., Zhang, S., Morello, G. M., Maskal, J. M., & Trottier, N. L. (2019). Technical note: development of an indirect calorimetry system to determine heat production in individual lactating sows. *Journal of Animal Science, 97*, 1609-1618. doi:10.1093/jas/skz049

Knox, R. V., Rodriguez Zas, N. L., Sloter, N. L., McNamara, K. A., Levis, D. G., & Singleton, W. L. (2013). An analysis of survey data by size of breeding herd for the reproductive management practices of North American sow farms. *J. Anim. Sci., 91*, 433-445. doi: 10.2527/jas.2012-5189

Lewis, R. W., Nithiarasn, P., & Seetharamu, K. N. (2004). *Fundamentals of the Finite Element Method for Heat and Fluid Flow.* New York, NY: John Wiley & Sons. doi:10.1002/0470014164

Maskal, J., Cabezon, F. A., Schinckel, A. P., Smith, A. J., Marchant-Forde, J. N., Johnson, J. S., & Stwalley III, R. M. (2018). Evaluation of floor cooling on lactating sows under mild and moderate heat stress. *The Professional Animal Scientist, 34*, 84-94. doi:10.15232/pas2017-01661

Mondaca, M., Rojano, F., Choi, C. Y., & Gebremedhin, K. G. (2013). A conjugate heat and mass transfer model to evaluate the efficiency of conductive cooling for dairy cattle. *Transactions of the ASABE 56*(6), 1471-1482. doi:10.13031/trans.56.10178

Nardone, A., Ronchi, B., Lacetera, N., Ranieri, M. S., & Bernabucci, U. (2010). Effects of climate changes on animal production and sustainability of livestock system. *Livest. Sci., 130*, 57-69. doi:10.1016/j.livsci.2010.02.011

National Research Council. (2012). *Nutrient Requirements of Swine.* Washington, DC: National Academy Press. doi:10.17226/13298

Noblet, J., Shi, X. S., & Dubois, S. (1993). Energy cost of standing activity in sows. *Livestock Production Science, 34*, 127-136. doi:10.1016/0301-6226(93)90041-F

Ortiz, X. A., Smith, J. F., Rojano, F., Choi, C. Y., Bruer, J., Steele, T., . . . Collier, R. J. (2015). Evaluation of conductive cooling of lactating dairy cows under controlled environmental conditions. *Journal of Dairy Science 98*(3), 1759-1771. doi:10.3168/jds.2014-8583

Parois, S. P., Cabezon, F. A., Schinckel, A. P., Johnson, J. S., Stwalley III, R. M., & Marchant-Forde, J. N. (2018). Effect of floor cooling on behavior and heart rate of late lactation sows under acute heat stress. *Frontiers in Veterinary Science, 5*(223), 1-8. doi:10.3389/fvets.2018.00223

Perano, K. M., Usack, J. G., Angenent, L. T., & Gebremedhin, K. G. (2015). Production and physiological responses of heat-stressed lactating dairy cattle to conductive cooling. *Journal of Dairy Science 98*(8), 5252-5261. doi:10.3168/jds.2014-8784

Prunier, A. M., de Braganca, M., & Le Dividich, J. (1997). Influence of high ambient temperature on performance of reproductive sows. *Livest. Prod. Sci., 52*, 123-133. doi:10.1016/S0301-6226(97)00137-1

Quiniou, N., &Noblet, J. (1999). Influence of high ambient temperatures on performance of multiparous lactating sows. *J. Anim. Sci., 77*, 2124-2134. doi:10.2527/1999.7782124x

REOTEMP Instrument Corporation. (2011). *Thermocoupleinfo.com/thermocouple-accuracies.htm.* Retrieved 2017, from Thermocoupleinfo.com.

Rojano, F., Choi, C., Ortiz, X. A., & Collier, R. J. (2019). Development of a water pipe network serving as a conductive cooling system applied to dairy farms. *Ingenieria Agricola y Biosistemas 11*(2), 127-145. doi:10.5154/r.inagbi.2018.06.012

Rojano, F., Mondaca, M., & Choi, C. Y. (2011). Feasibility of a dual cooling system for dairy cows in Arizona. *ASABE 2011 AIM – Louisville#1111661.* St. Joseph: ASABE. doi:10.13031/2013.38149

Schinckel, A. P., & Stwalley III, R. M. (2015). *USA Patent No. 62/268,066: Purdue University Office of Technology Commercialization.*

Seidel, D. S., Field, T. C., Schinckel, A. P., Stwalley, C. S., & Stwalley III, R. M. (2020). Effects of temperature probe orientation on the Purdue hog cooling pad data acquisition. *Computers and Electronics in Agriculture 175*, 105609. doi:10.1016/j.compag.2020.105609

Shaffer, C. S., Riskowski, G. L., & Harrison, P. C. (2017). Effect of conductive cooling pads on heat and moisture production of gilts in how and thermoneutral environments. *CIRG Journal, 19*(4), 13-19.

Shaffer, C. S., Riskowski, G. L., Harrison, P. C., &Su, J. (2001). Benefits of conductive cooling pads for sows. *ASAE Paper 01-4108.* St. Joseph, MI: ASAE.

Silva, B. A., Oliveira, R. F., Donzele, J. L., Fernandes, H. C., Lima, A. L., Renaudeau, D., &Noblet, J. (2009). Effect of floor cooling and dietary amino acids content on performance and behavior of lactating primiparous sows during summer. *Livest. Sci., 120*, 20-34. doi:10.1016/j.livsci.200804.015

Silva, B. A., Olivera, R. F., Donzele, J. L., Feranandes, H. C., Abreu, M. L., Noblet, J., & Nunes, C. G. (2006). Effect of floor cooling on performance of lactating sows during summer. *Livest. Sci., 105*, 176-184. doi:10.1016/j.livsci.2006.06.007

Stinn, J. P., & Xin, H. (2014). Heat and moisture production rates of a modern U.S. swine breeding-gestation-farrowing facility. *Transactions of ASABE 57*, 1517-1528. doi:10.13031/trans.S7.10711

Van Wagenberg, A. V., van der Peet-Schwering, C. M., Binnendijk, G. P., &Claessen, P. P. (2006). Effect of floor cooling on farrowing sow and litter performance: Field experiment under Dutch conditions. *Trans. ASABE, 49*, 1521-1527. doi:10.13031/2013.22044.