Environmental conditions in a growing-finishing swine building ventilated with and without earth tube heat exchanger

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Lemay, S.-P., Marquis, A. and D’Allaire, S. 1994. Environmental conditions in a growing-finishing swine building ventilated with and without earth tube heat exchanger. Can. Agric. Eng. 36:263-271. An earth tube heat exchanger was coupled with a slotted air inlet in three rooms of a growing-finishing swine building. A slotted air inlet provided fresh outside air to three other rooms without heating. Exchangers 1, 2, and 3 consisted of three, four, and four pipes of 0.3 m diameter non-perforated plastic drainage pipes, respectively. Each line was 61 m long and buried at 3 m depth. The air temperature, relative humidity, and ventilation rate of each room were measured between January 18, 1990 and February 14, 1991. Growth performance and health of pigs were also evaluated. For different combinations of outside air temperature and pig mass, there was no significant effect of the exchanger on ventilation rate compared to the conventional system. Pneumonia lesions, atrophic rhinitis degrees, and growth performance of pigs were not affected by use of the exchanger. In this study, no difference was observed between the two ventilation systems. The construction and use of similar earth tube heat exchanger is therefore not recommended in regions with equivalent energy costs and climatic conditions to that of Quebec.

Les entrées d’air du système de ventilation d’une porcherie d’engraissement étaient composées d’échangeurs souterrains et d’entrées d’air continues à contre-poids. La porcherie était divisée en six salles: trois d’entre elles bénéficiaient du conditionnement par un échangeur tandis que les trois autres fonctionnaient avec des entrées d’air continues sans conditionnement. Les échangeurs 1, 2 et 3 étaient constitués de trois, quatre et quatre tuyaux de drainage non-perforés de 0.3 m de diamètre, d’une longueur de 61,0 m et enfouis à 3,0 m dans le sol. La température, l’humidité relative de l’air intérieur et le débit de ventilation ont été mesurés. Parallèlement aux conditions environnementales, la santé et les performances des porcs ont été évaluées. Entre le 18 janvier 1990 et le 14 février 1991, pour différentes valeurs de température extérieure et de poids des porcs, les résultats n’indiquaient aucune différence de débit de ventilation des salles entre les deux systèmes de ventilation n’a pu être dégagée des résultats. Les échangeurs souterrains n’ont pas modifié significativement l’environnement et les performances des porcs. C’est pourquoi, l’utilisation d’échangeurs équivalents n’est pas recommandée dans les régions ayant un climat et des coûts d’énergie semblables à ceux du Québec.

INTRODUCTION AND LITERATURE REVIEW

Background

An important aspect of indoor swine production is adequate control of air temperature and relative humidity to maintain conditions for optimal health and growth. In Quebec’s climate, since typical winter design temperature is -24°C (St-Hyacinthe region, NRC 1985), supplemental heating is theoretically essential and can represent a significant production cost. However, in practice, the Quebec growing-finishing swine buildings are not heated. During cold periods, the producers shut down the ventilation rate to maintain the air temperature. Consequently, the relative humidity increases, some diseases may appear and the building’s structure may decay.

An earth tube heat exchanger (ETHE) system was retrofitted to a growing-finishing building designed for all-in/all-out (AIAO) production. Quebec swine producers are rapidly adopting this method of production since it lowers mortality and incidence of diseases (Nicks and Dechamps 1986). The AIAO production method is based on placing several small groups of young pigs into separate rooms, rearing them, and finally shipping them all out when market weight is reached. The whole building is then completely cleaned and disinfected and a new set of pigs is introduced. This method reduces the possibility of spread of disease; however, construction costs for the building are higher because of the full walls.

Effects of exchanger on environment

In general, the literature indicates that optimum air conditions for growth performance and health of growing pigs are 15-20°C in temperature and 70% relative humidity. Thus, heating and ventilation loads for AIAO systems may be higher when scheduling coincides with the presence of young pigs during cold periods or the presence of those close to market weight during hot periods.

Dulosy and Glawischnig (1980) installed an ETHE system in a fattening building for 140 swine. They used 20 pipes (200 mm diameter, 35 m length). The system maintained pen temperatures at 15 to 21°C throughout the year without supplemental heating, where maximum and minimum pen temperatures of 40°C and 5°C had been previously measured. The change of indoor temperature from 15 to 21°C occurred progressively throughout the year and presented no problem of acclimatization to the animals.

In Belgium, Neukermans et al. (1989) used a heat exchanger made of PVC pipes buried under a farrowing pig
building. In winter, the exchanger heated the ventilation air from -4.8°C to 10°C which, along with the heat output of the pigs, was sufficient to maintain adequate ambient air temperature without supplementary heating. They also reported that at an outside temperature of 28.3°C, the inside air temperature was reduced slightly to 26°C.

In Poland, Besler and Gryglewicz (1985) studied three systems: 1) a mechanical ventilation system without heating or cooling, 2) a mechanical ventilation system with a traditional cooler, and 3) a mechanical ventilation system with a heat and mass exchanger. They reported that in a beef growing building, the exchanger maintained the inside temperature in the range 15 to 22°C while the outside temperature ranged from -15 to 30°C. They noted that the lowest feed conversion by young beef cattle and pigs occurred in buildings equipped with earth heat exchangers.

**Effects of temperature**

For the same feed intake, a decrease in the surrounding air temperature from 28 to 20°C, resulted in a reduction of growth of 6 g·d⁻¹·°C⁻¹ (Le Dividich et al., 1985). Between 20 and 12.5°C, this reduction was 14.3 g·d⁻¹·°C⁻¹. The growth rate was maintained constant by an additional supply of food of 16 g·d⁻¹·°C⁻¹ in the 28 to 20°C temperature interval and of 38 g·d⁻¹·°C⁻¹ for the 20 to 12.5°C range.

Nichols et al. (1982) studied pigs fed ad libitum at temperatures ranging from 0 to 35°C. Pig mass was 75 kg at the beginning of each trial. No significant difference in gain and feed/gain ratio was observed at temperatures between 10 and 25°C. These authors concluded that the decision on the type and degree of environmental modifications must be based on the relative price of food versus fuel. Their results indicated that heating or cooling may not be justified for temperatures from 10 to 25°C for finishing swine. Nienaber et al. (1987) obtained similar results for air temperatures between 5 and 30°C. With pigs having a mass of 43 to 86 kg and being fed ad libitum, the maximum growth rate reached 0.749 kg/d and the minimum feed conversion was 3.0 kg of food per kg of gain at an air temperature of 20°C. However, the growth rate and feed conversion did not change significantly between 5 and 20°C and 15 and 25°C, respectively.

**Effects of relative humidity**

Very few experiments concisely describing animal health and performance at different humidities have been reported. In some cases, the conclusions of these experiments are somehow contradictory (Wathes et al., 1988). Christison (1988) specified that if relative humidity rises, there will be an inevitable rise in concentration of other air pollutants such as odor, noxious gases, dust, and pathogens. Microbe viability is enhanced by high humidity. Nicks and Dechamps (1986) suggest that the humidity affects the lifetime of microbes in the air and on the ground.

Recommended values of relative humidity vary and each situation must be considered individually. In practical terms, livestock may thrive perfectly well in temperate climates over a wide range of humidities extending from 30% to 90% (Sainsbury and Sainsbury 1988). On the contrary, Nicks and Dechamps (1986) advise that to reduce the possibility of infection, producers should practice the all in-all out technique, must maintain relative humidity between 60 and 80%, decrease animal density and maintain proper ventilation.

The possibility of coupling ETG to a standard slotted air inlet-based heating and ventilation system was therefore investigated. The objectives of the project were to compare environmental conditions (air temperature, relative humidity, ventilation rate), growth performance and health of pigs (pneumonia lesions, atrophic rhinitis degree) with two mechanical ventilation systems: an earth tube heat exchanger combined with a slotted air inlet and a slotted air inlet introducing outside air directly without a heating system in the room. Considering the heating capacities of the exchanger, a higher ventilation rate and lower relative humidity were assumed to be associated with an exchanger system (Figs. 1 and 2).

**THEORY**

The sensible heating or cooling rate of the exchangers is given by:

\[
Q_{\text{exch}} = M_{\text{exch}} \cdot C_p \cdot (T_{pi} - T_{ou})
\]

where:

\[
Q_{\text{exch}} = \text{heating or cooling rate of the exchanger (kW)},
\]

\[
M_{\text{exch}} = \text{mass air flow rate of the exchanger (kg/s)},
\]

\[
C_p = \text{specific heat of air} = 1.005 \left( \frac{\text{kJ}}{\text{kg} \cdot \text{K}} \right),
\]

\[
T_{pi} = \text{air temperature at outlet of pipes (°C)}, \quad T_{ou} = \text{outside air temperature (°C)}.
\]

Murray and Britton (1985) and Britton et al. (1987) measured the performance of four PVC pipes of 30 m long with diameter of 150 mm and 250 mm. The heating rate of energy obtained varied from 1.00 to 1.26 kW per pipe for an outside temperature of -20°C. To calculate the theoretical heating rate of the exchangers, we have considered an energy rate of 1.25 and 0.0 kW per pipe for outside temperatures of -20 and 5°C, respectively (pipe: 300 mm diameter, 61 m long).

The determination of ventilation rates is based on evaluation of heat and moisture balances in the building. The ventilation rate for the temperature control depends on heat production of pigs, heat losses of building and heating rate of the exchanger (Hellickson and Walker 1983):

\[
V_t = \frac{V_s \left[ Q_{pi} - Q_b + Q_{\text{exch}} \right]}{C_p \left[ T_{in} - T_{ou} \right]}
\]

where:

\[
V_t = \text{ventilation rate for temperature control (m}^3/\text{s)},
\]

\[
V_s = \text{specific volume of air (m}^3/\text{kg dry air),}
\]

\[
Q_{pi} = \text{total sensible heat production of pigs (kW),}
\]

\[
Q_b = \text{heat losses of building (kW), and}
\]

\[
T_{in} = \text{air temperature (°C)}.
\]

Ventilation rate can be plotted against outside air temperature for a given inside air temperature. Also, the ventilation rate for relative humidity control is given by (Hellickson and Walker 1983):

\[
V_h = \frac{V_s \cdot W_{pi} - W_{ih} - W_{ou}}{W_{in} - W_{ou}}
\]
where:
\[ V_h = \text{ventilation rate for humidity control (m}^3/\text{s)}, \]
\[ W_{pi} = \text{total water production of pigs (kg water/s)}, \]
\[ W_{in} = \text{mass of water per unit mass of inside air (kg water/kg dry air), and} \]
\[ W_{ou} = \text{mass of water per unit mass of outside air (kg water/kg dry air).} \]

The mass of water per unit mass of air depends on the air temperature. Therefore, the two ventilation rates can be plotted against outside air temperature to maintain a specific air temperature and relative humidity. Figures 1 and 2 present an example for room C filled with 160 pigs with a mass of 70 kg, an inside air temperature of 20°C, and relative humidity of 70%.

These figures show that with the exchanger, the relative humidity should be lower and ventilation rate should be higher than without a heating system. In fact, for an outside temperature lower than -10°C, the relative humidity in a conventional system (without heating) is greater than 80%. These differences should be observed experimentally and constituted the basic hypothesis of this work.

Because the heat output from exchangers is the prime interest for this study, Figs. 1 and 2 are drawn for outside temperatures of -25 to 5°C. In summer, the ventilation systems are designed to maintain an inside air temperature 3°C higher than outside temperature. Since the air flow rate through an ETHE is much lower than the summer ventilation rate, the cooling effect from the ETHE is insignificant.

![Figure 1. Theoretical ventilation rates of room C to maintain a temperature of 20°C and relative humidity of 70%.

![Figure 2. Theoretical relative humidity in room C with temperature control of ventilation rate, with and without exchanger.

![Figure 3. Plan and elevation views of the swine building and the exchangers 1, 2, and 3, St-Hyacinthe, QC.

MATERIALS AND METHODS

Building
The swine building was divided into six rooms A, B, C, D, E, and F (Fig. 3) which housed 120, 120, 160, 160, 150, and 150 pigs, respectively. The building was 15 years old. The wall and ceiling heat losses were estimated to be 0.361 and 0.340 W·m\(^{-2}·\text{°C}^{-1}\), respectively. The rooms were connected with plywood doors and they were sealed from one another with a

film of polyethylene installed in the wall. The exchangers were made of non-perforated plastic drainage pipes, 0.3 m diameter and 61 m in length (Lemay and Marquis 1993). The pipes were spaced at 3.0 m and buried at 3.0 m depth. Exchangers 1, 2, and 3 conditioned a part of the ventilation air for rooms A, C, and E. They consisted of three, four, and four pipes, respectively. These exchangers could provide 15% of the summer total ventilation rate of each room. The ventilation worked in sequence; at minimum ventilation rates the heat exchanger supplied all of the ventilation air, and after 15% of summer ventilation was reached, the slotted air inlets opened.
The supplemental heat requirements could be calculated according to ventilation rates needed to control temperature and humidity. For an outside temperature of -20°C and relative humidity of 70%, the heat requirements of rooms A, B, and C to maintain inside conditions of 20°C, 70% R.H. and pig mass of 70 kg were 6.0, 4.7, and 6.0 kW, respectively. The heat requirements of rooms D, E, and F were similar to room C. However, rooms B, D, and F were equipped with slotted air inlets only and were not provided with any heating system.

A constant speed centrifugal fan (Delhi model Bi-13, Les industries Aston Inc., St-Léonard d’Aston, QC) drew the air through each exchanger. The fan motors of exchangers 1, 2, and 3 developed nominal powers of 1.2, 1.5, and 1.5 kW, respectively. Each fan operated in conduction with two shutters to modulate the exchanger air flow rate: a shutter downstream of the wood duct for incoming air from the exchanger and the other for recirculation of the room air (Fig. 4). A static pressure control (BCB, Les équipements agricoles BCB, Dunham, QC) operated simultaneously and inversely on both shutters through a linear actuator to maintain pressure between 10 and 15 Pa.

![Inlet air system diagram with earth tube heat exchanger.](image)

**Fig. 4. Inlet air system diagram with earth tube heat exchanger.**

The ventilation of each room operated by negative pressure. Rooms A and B had two 502 mm diameter variable speed fans (model 501-6, Ziehl Abegg GmbH & Co. Kg, Kunzelsan, Germany). A constant speed 451 mm diameter fan (1750 rpm, Ventilateur Victoria Ltée, Victoriaville, QC) completed the summer ventilation rate. The other rooms had two 451 mm diameter variable speed fans (model 451-4, Ziehl Abegg GmbH & Co. Kg, Kunzelsan, Germany) and one 451 mm constant speed fan. The maximum ventilation rates of the model 501-6 and 451-4 fans were 1082 and 1434 L/s at 30 Pa, respectively. The maximum ventilation rate of the constant speed fan was 1258 L/s at 30 Pa. The ventilation rate of the fans was controlled by the room air temperature. The controls were adjusted to reach the maximum rotation speed of the fan when the inside temperature was 3°C higher than the air temperature set point.

Instrumentation and measurements

Two probes measured the temperature and relative humidity of the air at the exchanger inlet and outlet (model 207, Campbell Scientific Inc., Logan, UT). The airflow rate was determined by measuring the head loss through an orifice plate, located in the wood duct, with an analogue manometer (model T-10, Modus instruments Inc., MA). Dry and wet bulb room temperatures were measured with copper constantan thermocouples.

The fan flow rates were also determined. An apparatus corresponding to CSA-C320-M86 standard (CSA 1986) was used to determine fan flow rate from the static pressure drop across the fan and the rotation speed. The fans were calibrated with their shutters and hood in both dirty and clean conditions. A regression analysis produced an equation of the air flow versus the static pressure and rotation speed for each fan. On the farm, the room static pressure was also measured with an analogue manometer. Magnetic sensors placed in front of the variable speed fans measured their rotation speed. The constant speed fans were connected to relays and the state of the relay indicated when the fan was operating or not. All the instruments were connected to a data logger (model CR7, Campbell Scientific Inc., Logan, UT).

**Measurement errors**

Periodic recalibration was done during the experiment. An error analysis was performed based on Jordan and Sewell’s (1983) procedure. Supplementary details about the error analysis are presented in Lemay (1991).

The temperature/humidity probes at the exchanger inlets and outlets measured the temperature to 0.2°C and the relative humidity to 5%. The error on the exchanger flow rates was 70, 40, and 40 L/s for exchangers 1, 2, and 3, respectively. With respect to the heating or cooling power of the exchangers, the corresponding errors were 0.9, 0.8, and 0.8 kW. The accuracy of the thermocouples placed in the psychrometers was 0.5°C. Consequently, the relative humidity was measured to 8%.

The flow rate error for the outlet fans was calculated from the experimental error associated with the regression analysis. This analysis showed a probable error of 60 L/s for the 451 mm constant speed fan, 60 L/s for the 451 mm, and 50 L/s for the 502 mm variable speed fans, respectively.

**Effects of exchanger on inside environment**

Measurements were taken from January 18, 1990 to February 14, 1991. This period was divided into three seasons: winter 1990 (W90), summer 1990 (S90) and winter 1991 (W91). The data logger recorded all parameters every five minutes. The mean, maximum, and minimum values of each measurement were saved every hour. For the W90 season, the analysis covered 10, 14, and 11 days for the room groups A-B, C-D, and E-F. These periods corresponded to days between January 18 and March 31 where the mean outside air temperature was lower than -4°C. For the S90 and W91 seasons, the five days considered for each group corresponded to the five hottest days available for S90 and the five coldest days for W91.

The days for data analyses were selected with respect to the reliability of measurements, mean outside air temperature, and pig mass. For example, to compare the measurements of two rooms for a specific day, all sensors had to be working properly, the outside air temperature had to be relatively cold, and the difference between the mean mass of
pigs in each room must have been less than 5 kg.

During the project, the mean initial mass of pigs was 27 kg. The mean daily mass of animals in each room was calculated with the inventory of pigs, the mean initial mass, and a growth model (ASAE 1993).

The ventilation system was adjusted in the fall for winter conditions and in the spring for summer conditions. The slotted air inlets responded to 25 Pa in winter and 10 Pa in summer providing air speeds at the inlet of approximately 5.0 m/s in winter and 2.5 m/s in summer (Hellickson and Walker 1983). The 5.0 m/s winter inlet air speed resulted in the inlet air being well mixed before reaching the pigs. In summer, the 2.5 m/s inlet air speed constrained wind effects and the fans provided a higher flow rate at their maximum rotation speed.

During the experiment, the farm ran normally without any specific intervention on mass and number of pigs in the room. Therefore, this experiment was conducted under normal working conditions and the youngest pigs were not necessarily present in the rooms when the outside air temperature was coldest.

**Effects of climate on the pigs**

Six cohorts of two groups of pigs each involving 416 pigs were observed. In each cohort, one group was housed in a room with the exchanger and a slotted air inlet and the other in a room with only a slotted air inlet. The pen animal density was 0.8 m²/pig. Growth performance, atrophic rhinitis (score of 1 to 5), and pneumonia (% of lung lesions) were evaluated at slaughter.

A 2 X 3 X 2 factorial experiment design was used: two ventilation systems, three cohorts of pigs, and two periods of the year. An analysis of variance permitted comparison of the two ventilation systems.

**RESULTS AND DISCUSSION**

**Heating environment**

Figures 5 to 8 show air temperature and ventilation rate for the room groups A-B and C-D for three days of the W90 season. The measurements for the groups E-F are not discussed because the results of this group were similar to the group C-D.

Figures 5 and 7 demonstrate that the air temperature in the building averaged 15°C. During W90 season, an outbreak of swine pleuropneumonia occurred. In an attempt to control the disease, the ventilation rate was increased and consequently the temperature was decreased from the original set point of 20 to 15°C.

Moreover, these graphs show the quality control of the ventilation system. Even though the exterior air temperature fluctuated widely, the inside temperature was relatively constant. In general, the hourly temperature variations were lower than 1°C indicating the ventilation system worked adequately.

Figures 5 and 6 show the results for rooms A and B on February 2 to 4, 1990. The mean mass of pigs in the two rooms was 93 kg. For an outside air temperature range of -19°C, the room B air temperature was 2°C warmer than room A air temperature. The outlet pipe air temperature was in the range of -2°C. In general, the exchanger heated outside air by 10°C. The relative humidity averaged 72 and 69% in rooms A and B, respectively (Table I). Considering the measurement accuracy on air flow rate (±1 L·s⁻¹/pig), the ventilation rate in room B was higher than the ventilation rate in room A. For these days, exchanger 1 provided a mean sensible heating power from 2.0 to 2.1 kW. The maximum heating power equalled 4.2 kW and was obtained on February 1 at 1300h.

The inside air temperature in room A was lower than in room B even though room A had a lower ventilation rate and supplemental heating. This was because room A had three exterior walls, the heat loss of room A was greater than from room B. To maintain the same temperature and ventilation rate, the insulation of room A should have been higher than that of room B.

Figure 7 presents the air temperature in rooms C and D on February 23 to 25, 1990 when the mean pig mass was 74 kg. With outside air temperature ranging from 1.0 to -16.5°C, inside air temperatures of rooms C and D were 16°C and 14°C, respectively. Air was warmer in room C than in room D. The mean relative humidity was 57% in room C and 60% in room D (Table I). Allowing for the psychrometer accuracy, relative humidity was considered to be the same in both
rooms. The ventilation rate in room C was lower than in room D (Fig. 8). The higher inside air temperature in room C explains the lower ventilation rate. Exchanger 2 provided a mean sensible heating power of 1.8 to 4.8 kW. The maximum energy output reached 7.7 kW on February 25 at 1500h.

Dullosy and Glawischang (1980) and Neukernans et al. (1989) showed that the inside air temperature was more constant with an earth tube heat exchanger than with a conventional ventilation system. However, the air temperature fluctuations in room D were not greater than those in room C. It seems that a conventional ventilation system with proper adjustments can produce the same air temperature control as an exchanger.

For February 25, Fig. 8 shows a ventilation rate in the range of 5 Ls⁻¹/pig. For the same conditions (climate, number, and mass of pigs), the ventilation rate calculated from the heat production would be 3 Ls⁻¹/pig according to Hellickson and Walker (1983) and 4 Ls⁻¹/pig according to CIGR (1984). Considering that the air flow rate measurement accuracy in our study was ±1 Ls⁻¹/pig, the sensible heat production observed in this experiment was much closer to the CIGR (1984) values. Clark et al. (1984) measured heat and moisture loads in four swine feeder barns equipped with two types of flooring arrangement. With solid concrete floors, they observed sensible heat productions ranging from 515 to 586 kJh⁻¹/pig for pigs of average liveweight of 56 kg at an inside temperature of 17°C. Under such conditions, mean heat productions would be 329 and 461 kJh⁻¹/pig, according to Hellickson and Walker (1983) and CIGR (1984), respectively. Since both Clark et al. (1984) and our measurements are slightly higher than those that would be predicted by CIGR (1984), the CIGR (1984) method appears to be more realistic today than Hellickson and Walker’s (1983) values. The heat production probably decreased the influence of the exchanger on ventilation rates and may explain why many growing-finisher pig buildings in Quebec are successful without supplementary heat.

### Table I. Mean relative humidity in the rooms and heating or cooling power of exchangers for winter 1990 and summer 1990 results

<table>
<thead>
<tr>
<th>Date</th>
<th>Rooms (i-j)</th>
<th>Exchanger</th>
<th>R. H. room i (%)</th>
<th>R. H. room j (%)</th>
<th>Mean power* (kW)</th>
<th>Maximum power* (kW)</th>
<th>Time of max. power (h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>02/02/90</td>
<td>A-B</td>
<td>1</td>
<td>71</td>
<td>68</td>
<td>2.1</td>
<td>4.2</td>
<td>1300h</td>
</tr>
<tr>
<td>02/03/90</td>
<td>A-B</td>
<td>1</td>
<td>72</td>
<td>69</td>
<td>2.1</td>
<td>3.3</td>
<td>1300h</td>
</tr>
<tr>
<td>02/04/90</td>
<td>A-B</td>
<td>1</td>
<td>72</td>
<td>69</td>
<td>2.0</td>
<td>4.1</td>
<td>1600h</td>
</tr>
<tr>
<td>02/23/90</td>
<td>C-D</td>
<td>2</td>
<td>55</td>
<td>60</td>
<td>1.8</td>
<td>3.0</td>
<td>1300h</td>
</tr>
<tr>
<td>02/24/90</td>
<td>C-D</td>
<td>2</td>
<td>56</td>
<td>60</td>
<td>4.2</td>
<td>6.7</td>
<td>1300h</td>
</tr>
<tr>
<td>02/25/90</td>
<td>C-D</td>
<td>2</td>
<td>59</td>
<td>61</td>
<td>4.8</td>
<td>7.7</td>
<td>1500h</td>
</tr>
<tr>
<td>06/12/90</td>
<td>A-B</td>
<td>1</td>
<td>59</td>
<td>62</td>
<td>-2.3</td>
<td>-4.7</td>
<td>1600h</td>
</tr>
<tr>
<td>06/13/90</td>
<td>A-B</td>
<td>1</td>
<td>60</td>
<td>63</td>
<td>-2.6</td>
<td>-3.9</td>
<td>1800h</td>
</tr>
<tr>
<td>06/14/90</td>
<td>A-B</td>
<td>1</td>
<td>59</td>
<td>62</td>
<td>-3.7</td>
<td>-4.6</td>
<td>1800h</td>
</tr>
<tr>
<td>08/24/90</td>
<td>C-D</td>
<td>2</td>
<td>60</td>
<td>68</td>
<td>-3.4</td>
<td>-8.0</td>
<td>1400h</td>
</tr>
<tr>
<td>08/25/90</td>
<td>C-D</td>
<td>2</td>
<td>62</td>
<td>68</td>
<td>-4.4</td>
<td>-8.4</td>
<td>1600h</td>
</tr>
<tr>
<td>08/26/90</td>
<td>C-D</td>
<td>2</td>
<td>64</td>
<td>70</td>
<td>-4.4</td>
<td>-8.5</td>
<td>1700h</td>
</tr>
</tbody>
</table>

R. H.: relative humidity.
Time of max. power: time when the maximum power of the exchanger was reached.
* Negative value means a cooling power.

**Cooling environment**

Results of summer data are shown in Figs. 9 to 12. Dates of June 12 to 14 and August 24 to 26, 1990 were selected to demonstrate room responses A-B and C-D. The general behaviour of room group E-F was comparable to group A-B. Therefore, the analyses of rooms A-B would explain the results of group E-F also.

Figure 9 demonstrates that inside air temperature was comparable in both systems of air inlets. Moreover, the inside air temperature exceeded slightly the outside air temperature during the hottest day. The ventilation system was designed to maintain air temperatures 3°C higher in the room.
1.5°C and 2.0°C lower than outside and room D air temperatures, respectively (Fig. 11). During the warmest part of the day, the exchanger cooled the outside air by 10°C. Relative humidities ranged from 60 to 70% in the two rooms (Table I). The constant speed fan in room C failed, resulting in low but constant ventilation during the day (Fig. 12). Between 0000h and 0900h, ventilation in room C equaled that in room D. Exchanger 2 provided a mean sensible cooling rate of 3.4 to 4.4 kW for room C. These results confirm that decreasing the ventilation rate reduced the inside air temperature. Effectively, when the ventilation rate was decreased, the percentage of warm air (standard inlet air, no cooling) was reduced. This observation raises a question about the lowest possible temperature in room C that could have been obtained.

The minimum ventilation rate was estimated at 6.3 L·s⁻¹/pig and consisted of the exchanger air flow rate of 4.2 L·s⁻¹/pig and the minimum room infiltration of 2.1 L·s⁻¹/pig. The infiltration rate was estimated from the difference between the fan ventilation rate (outlet flow rate) and the exchanger flow rate (a part of inlet flow rate) during the winter when the slotted air inlets were closed. If the ventilation rate had been adjusted to 6.3 L·s⁻¹/pig, calculations indicate that inside air than outside. The performance of the ventilation system was better than the design calculations. Relative humidity was 3% lower with the exchanger than with standard slotted air inlets (Table I). Consequently, it was comparable in both rooms.

The ventilation rate in room A using the exchanger was higher than the ventilation rate of room B (the control) in the morning from 0000h to 1000h (Fig. 10). Between 1000h and 2300h, the ventilation rate was equal in both systems. For these days, exchanger 1 provided a mean sensible cooling power of 2.3 to 3.7 kW. To obtain the same inside air temperature with higher ventilation rate in room A, only a part of incoming air was contributing to control temperature and moisture. The exchanger air flow rate was distributed in a wood duct, under the slotted air inlets (Fig. 3). The momentum of exchanger air flow maintained the slotted inlet air flow near the ceiling and prevented the fresh air to fall in the room and to increase in temperature and humidity. There was no distribution duct in room B. The room air was probably well mixed and the air flow rate needed to control temperature and humidity was lower than in room A. The geometry of the distribution duct is the most probable explanation for this over ventilation.

Figures 11 and 12 present the results for rooms C and D. Between 1200h and 1800h, the temperature of room C was
Table II. Health and performance of pigs with both ventilation systems for the cohorts started between March and May 1990 (St-Hyacinthe, QC)

<table>
<thead>
<tr>
<th>Cohort</th>
<th>Number of pigs</th>
<th>Initial mass (kg)</th>
<th>Final mass (kg)</th>
<th>ADG (g/d)</th>
<th>Days to market</th>
<th>Pneumonia lesions(a)</th>
<th>Atrophic rhinitis(a) score</th>
<th>MSE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
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<td></td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Room A</td>
<td>32</td>
<td>26.5</td>
<td>93.8</td>
<td>639.4</td>
<td>106.1</td>
<td>10.7</td>
<td>1.24</td>
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<tr>
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<td>1.2</td>
<td>16.1</td>
<td>1.7</td>
<td>2.2</td>
<td>0.14</td>
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</tr>
<tr>
<td>Room B</td>
<td>41</td>
<td>24.4*</td>
<td>94.0</td>
<td>628.7</td>
<td>111.4*</td>
<td>13.6</td>
<td>1.38</td>
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<td>1.7</td>
<td>18.8</td>
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<td>3.1</td>
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</tr>
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<td></td>
</tr>
<tr>
<td>Room C</td>
<td>28</td>
<td>25.5</td>
<td>93.4</td>
<td>653.2</td>
<td>105.5</td>
<td>2.6</td>
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<td>1.1</td>
<td>17.0</td>
<td>2.8</td>
<td>1.0</td>
<td>0.16</td>
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<td></td>
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<tr>
<td>Room D</td>
<td>29</td>
<td>25.0</td>
<td>93.4</td>
<td>666.4</td>
<td>104.4</td>
<td>6.1</td>
<td>0.94</td>
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</tr>
<tr>
<td>MSE</td>
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<td>1.7</td>
<td>22.2</td>
<td>2.5</td>
<td>1.5</td>
<td>0.17</td>
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<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Room E</td>
<td>30</td>
<td>31.7</td>
<td>98.6</td>
<td>708.8</td>
<td>95.6</td>
<td>6.9</td>
<td>1.43</td>
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<td>1.1</td>
<td>16.0</td>
<td>2.5</td>
<td>3.4</td>
<td>0.16</td>
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<td></td>
</tr>
<tr>
<td>Room F</td>
<td>38</td>
<td>32.2</td>
<td>101.9</td>
<td>785.2*</td>
<td>88.0*</td>
<td>1.4</td>
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<td>0.12</td>
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</tr>
</tbody>
</table>

\(a\) These variables have a gamma distribution. Therefore, the analysis of variance is not applied.
*The difference between the groups of the same cohort is significant at P \(\leq 0.05\).

MSE: mean standard error.

Table III. Health and performance of pigs with both ventilation systems for the cohorts started between July and September 1990 (St-Hyacinthe, QC)

<table>
<thead>
<tr>
<th>Cohort</th>
<th>Number of pigs</th>
<th>Initial mass (kg)</th>
<th>Final mass (kg)</th>
<th>ADG (g/d)</th>
<th>Days to market</th>
<th>Pneumonia lesions(a)</th>
<th>Atrophic rhinitis(a) score</th>
<th>MSE</th>
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<tr>
<td>Room A</td>
<td>51</td>
<td>25.5</td>
<td>95.4</td>
<td>681.7</td>
<td>102.1</td>
<td>6.4</td>
<td>1.25</td>
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<td>1.0</td>
<td>15.4</td>
<td>2.5</td>
<td>1.3</td>
<td>0.16</td>
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<td></td>
</tr>
<tr>
<td>Room B</td>
<td>41</td>
<td>28.4*</td>
<td>93.6</td>
<td>717.1</td>
<td>92.1</td>
<td>9.7</td>
<td>1.24</td>
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<td>2.4</td>
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<tr>
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<td></td>
</tr>
<tr>
<td>Room C</td>
<td>34</td>
<td>28.7</td>
<td>96.6</td>
<td>673.7</td>
<td>101.6</td>
<td>5.4</td>
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<td>17.4</td>
<td>2.3</td>
<td>1.1</td>
<td>0.15</td>
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<tr>
<td>Room D</td>
<td>30</td>
<td>29.2</td>
<td>95.7</td>
<td>658.2</td>
<td>101.9</td>
<td>14.3</td>
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</tr>
<tr>
<td>MSE</td>
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<td>1.6</td>
<td>18.9</td>
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<td>0.18</td>
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<td>6</td>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Room E</td>
<td>34</td>
<td>23.8</td>
<td>104.5</td>
<td>674.9</td>
<td>121.5</td>
<td>6.3</td>
<td>1.10</td>
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<td>Room F</td>
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<td>24.1</td>
<td>102.3</td>
<td>629.6</td>
<td>126.6</td>
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<td>25.4</td>
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<td>3.0</td>
<td>0.15</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

\(a\) These variables have a gamma distribution. Therefore, the analysis of variance is not applied.
*The difference between the groups of the same cohort is significant at P \(\leq 0.05\).

MSE: mean standard error.

The temperature would have been 3°C lower than outside air temperature. To obtain 6.3 L\(\text{m}^{-1}\)\(\text{pig}\), the constant fan should have been stopped (as in the experiment) and the speed of the variable fan decreased. However, the energy savings would have been low and the setup complicated. Moreover, the effect of a 1°C decrease in temperature has probably little influence on pigs and does not justify this adjustment.

The exchanger of Neukermans et al. (1989) maintained an inside air temperature of 26°C for an outside air temperature of 28.3°C. The exchanger used by these authors conditioned the total air ventilation flow. In the present experiment, with the high ventilation rate in room A, exchanger 1 had a small influence on the inside air temperature. Exchanger 2 provided only 15% of the summer total ventilation rate. Their performances are therefore considered similar to that in the literature.

The cooling rate of the current exchanger was not significant under Quebec conditions. In fact, even with the minimum ventilation rate of a room as in room C, when the constant speed fan was not operational, the temperature differences between inside and outside air were small and could be observed only for a short period of time during the day (Fig. 11). A larger ETHE with a higher flow rate operating alone without conventional sidewall ventilation may provide significant cooling on hot days. However, economic benefit would have to be studied to determine if the extra cost of a larger ETHE system would be justified.

Health and growth performance

A total of 416 pigs was studied between March 2, 1990 and January 23, 1991. Growth performance, atrophic rhinitis degrees, and percent of pneumonia lesions were evaluated at slaughter (Tables II and III). The average daily gain varied from 0.629 kg/d for the room B of cohort 1 to 0.785 kg/d for the room F of cohort 3. These performances are comparable with the results of Nienaber et al. (1987). The average percentage of pneumonia lesions varied from 1.4 to 14.3%. These values were obtained with conven-
tional ventilation system. The atrophic rhinitis scores ranged from 0.59 to 1.43. Both of these extreme values occurred with the earth tube heat exchanger. Although some differences in productivity and health variables were observed between the two groups in a few cohorts, it was concluded that the two types of ventilation systems gave similar results. These differences were observed in only a few cohorts and the effects were often in opposite directions. Therefore, the use of an exchanger does not seem to affect the performance and health of pigs.

CONCLUSIONS

Use of the earth tube heat exchanger to heat or cool a growing-finishing swine building did not significantly influence air temperature, relative humidity, or ventilation rate. The cooling performance of the exchanger was of poor value for the Quebec climate. In fact, the temperature difference between inside air with an exchanger and without was less than 2°C. Moreover, the temperature difference between inside and outside air was small and was observed only for a short time of the day. Also, the performance and health of pigs were similar with the exchanger system and standard slotted air inlet alone. The absence of significant differences or lack of consistency in the results can be explained by a similar quality and control of the environment with both ventilation systems. Therefore, the use and construction of similar earth tube heat exchangers is not recommended in regions with equivalent energy costs and climatic conditions to that of Quebec.

The similarity of the results (air temperature, relative humidity, ventilation rate, and pig performance) and the current heat production of pigs, indicate that proper management of a standard ventilation system without heating can provide adequate environmental conditions for pig production in Quebec.

REFERENCES


