The Effects of Pen Partitions and Thermal Pig Simulators on Airflow in a Livestock Test Room

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The aim of this work was to investigate the influence of pen partitions and heated simulated pigs on airflow in a slot ventilated test room and to evaluate computer fluid dynamics (CFD) as a tool to predict airflow in livestock rooms. To obtain two-dimensional flow in the occupied zone, four guiding plates were mounted beneath the ceiling in the test room. Experiments were carried out in three arrangements: (a) the room with guiding plates; (b) the room with guiding plates and eight heated pig simulators; and (c) the room with guiding plates, eight heated pig simulators and 0.8 m high partitions which divided the room into four equal-sized pens.

The guiding plates beneath the ceiling were efficient in creating two-dimensional flow in the occupied zone, but they increased the differences between measured and simulated air velocity close to the ceiling and close to the floor. Both measurements and CFD simulations showed that the introduction of pen partitions and thermal pig simulators reduced the air velocities in the occupied zone of the test room. Detailed geometric modelling of the animals might often be unnecessary for simulation of airflow in livestock rooms. This will especially be the case when the animals are located close to pen partitions or other large obstacles in the occupied zone.

Poor ability to predict recirculating zones limits the expected precision of CFD calculations with the k–ε turbulence model in livestock rooms where recirculating zones often occurs.

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1. Introduction

Control of climate parameters such as temperature, air velocity and air contaminants is important to maintain animal health, well-being and productivity in modern livestock rooms. Therefore, the prediction of climate parameters in animal houses has been the aim of a lot of research work during the last 30 years. Computational fluid dynamics (CFD) is one of the newest modelling tools used for this purpose, and it is a very fascinating tool because it includes potential possibilities to handle all important conditions which may influence the climatic environment in a livestock building. However, the use of CFD in relation to livestock houses has, so far, been concentrated on rather uncomplicated models where the CFD simulations have been compared with measurement in full scale or model scale test rooms, see e.g. Bjerg et al. (1999), Hoff et al. (1992), Worley and Manbeck (1995), Heber et al. (1996) and Harral and Boon (1997). The airflow in these test rooms was generated by a wall jet and usually without influences by heat supply or any kind of blockades such as animals, pen partitions or other equipment which are normally present in livestock rooms.

Smith et al. (1999) measured the effect of pigs on airflow in a full-scale cross-section of a livestock building. To maximize the stability of a rotary airflow pattern a ventilation rate of 50 air changes/h was used. This relatively high air change rate resulted in air velocities in the occupied zone (20 cm above floor) of about 0.7 m/s in a test room without pigs. Introduction of pigs reduced the air velocity at positions close to the animals with
typically 40%. Due to the high ventilation rate it is likely that the buoyancy effect from the pigs heat production played a minor role in their investigation and the major influence of pigs was caused by the physical blocking of the airflow.

Zhang et al. (1999) studied buoyant flow generated by thermal convection of a simulated pig by measurements and CFD simulations. This investigation took place in a room with no forced convection and room inside surface temperature of 18°C. The thermal convection generated by a pig simulator with the same surface temperature (35°C) and surface area as a pig of 100 kg created a maximum air velocity above the simulator of about 0.22 m/s.

The aim of the work presented in this paper was to investigate the influence of pen partitions and heated simulated pigs on airflow in a slot ventilated test room, and to evaluate CFD as a tool to predict airflow in livestock rooms. To achieve a reasonable interpretation of the result it was decided that the reference case for the investigation should be a two-dimensional airflow near the floor in an empty test room. The performance of the used empty test room is investigated in earlier works (Bjerg et al., 1999; Zhang et al., 2000) which showed a strong three-dimensional flow in the occupied zone although the boundary conditions of the room were two-dimensional. In the present work, guiding plates beneath the ceiling were used to obtain two-dimensional flow in the occupied zone.

Growing pigs usually spend more than 75% of the time lying (Van Ouwerkerk, 1992) and therefore, physical models which simulated lying pigs were used in this investigation.

2. Methods

2.1. The test room

The test room was 8.5 m deep, 3 m high, 5 m wide and equipped with a slot inlet beneath the ceiling at one entire end of the room and a slot outlet in the floor near the inlet wall. To obtain two-dimensional flow in the occupied zone four guiding plates were mounted beneath the ceiling. The plates were 7.65 m long and reached 0.5 m vertically down from the ceiling. Further details of the test rooms are given by Zhang et al. (2000).

Three different arrangements were included in the investigation (Fig. 1): A — the room with guiding plates; B — the room with guiding plates and eight heated pig simulators; C — the room with guiding plates, eight heated pig simulators and 0.8 m high closed partitions which divided the room into four pens of equal size.

2.2. Air inlet and airflow rate

Two different ventilation rates were used: a low ventilation rate of 0.14 m³/s and a high ventilation rate of 0.36 m³/s corresponding to about 4 and 10 air changes/h. The height of the inlet slot was 0.008 m in the low ventilation experiments, and 0.019 m in the high ventilation experiments and the Reynolds numbers were 1900.
and 4800, respectively (Reynolds number was calculated as \( Re = \rho V_0 h/\mu \), where \( \rho \) is the density, \( V_0 \) the inlet air speed, \( h \) the inlet height and \( \mu \) the laminar viscosity). The pressure drop over the inlet was about 15 Pa in all experiments. The two ventilation rates were expected to generate air velocities in the occupied zone of about 0.2 and 0.4 m/s, respectively.

2.3. Pig simulators and heat supply

The pig simulators were semi-cylindrical with diameter of 0.5 m and length of 1 m closed in the ends with metal plates and insulated against the floor by 9 mm plywood plate. The simulators were painted dark red and heated by four 100 W electric bulbs. The voltage was controlled to keep the total heat supply at 125 W per simulator in all non-isothermal experiments. Assuming full mixture of the incoming air and no heat transfer through walls it was calculated that the heat supply would elevate the room temperature by 2.3 K at high ventilation rate and 5.7 K at low ventilation rate.

To separate the buoyancy effect from the deflections caused by the geometry of simulators and pen partitions there were corresponding measurements made at isothermal conditions.

2.4. Measurements

The airflow pattern was determined by smoke and the direction of the return air was measured with a flow direction sensor (Zhang et al., 2000), which was located at \( x = 6.5 \) m, \( y = 0.15 \) m and \( z = 2.5 \) m in arrangement A; and at \( x = 6.5 \) m, \( y = 0.15 \) m and \( z = 3.5 \) m in arrangements B and C, where \( x \) denotes the longitudinal axis, \( z \) the lateral axis and \( y \) the vertical axis.

Air speed was measured with 30 thermistor-based omni-directional air speed sensors. Nine sensors in three groups were fixed at permanent locations throughout all measurements. The sensors in each group were located in a line transverse to the inlet flow in the \( z \) direction, where the \( xz \) denotes the horizontal plane and \( y \) the vertical axis. One sensor was located in the symmetry plane \( (z = 2.5 \) m) and one sensor \( 0.5 \) m from each side wall \( (z = 0.5 \) and \( 4.5 \) m). One group of three sensors was mounted near the ceiling \( (x = 4.5 \) m and \( y = 2.98 \) m) and used for reference measurement of incoming flow. Another group of sensors was located at floor level near the end of the room \( (x = 7 \) m and \( y = 0.2 \) m). A third group of sensors were located above the floor near the inlet wall \( (x = 2 \) m and \( y = 0.2 \) m). The remaining 21 sensors were mounted at three moveable fixtures with seven sensors on each. The sensors on each fixture were placed at a distance of 0.1 m in between. Detailed locations of the measurement points are shown in Fig. 1. All data were recorded for at least 25 min with average values and standard deviations (SD). The data scan rate was 10 scan/s.

2.5. Numerical simulation

The airflow was calculated by means of the commercial numerical simulation code — CFX4 (AEA Technology plc). An inlet velocity of 4.8 m/s with 4% turbulence intensity and wall functions at all surfaces was used in all the simulation. The effect of turbulence was taken into account by the \( k-\varepsilon \) turbulence model. This is a widely used two-equation model based on transport equations for turbulent kinetic energy and the dissipation of turbulent kinetic energy (Lauder & Spalding, 1974). In the simulations at low ventilation rate, the effective inlet slot height was 0.006 m and in the simulation with the high ventilation rate the effective slot height was 0.015 m.

Isothermal flow was assumed in arrangement A, whilst both isothermal and non-isothermal conditions were simulated in arrangements B and C. In the non-isothermal condition, the heat transfer was assumed to be 63 W convective heat and 63 W radiative heat from each of the eight simulated pigs. The convective part was distributed equally from the surface of the pig simulators. The radiative part was assumed to be transmitted to the room surfaces and from there released as convective heat. To simplify the simulation model the radiative heat was distributed evenly across the ceiling as convective heat. All other surfaces were assumed to be adiabatic.

An overview over the simulations is shown in Table 1. A code is used to identify the simulations. The first letter denotes the arrangement (A, B or C), the second letter denotes the airflow rate (H for high and L for low) and the third letter denotes heat supply (Y for yes and N for no). A number is used if different grid constructions was investigated.

The geometry in arrangement A is rectangular and can be modelled in a relatively uncomplicated grid. Opposite to that, the round geometry of the pig simulator and the openings between the simulators makes it necessary to use a much more complicated and non-rectangular grid in arrangements B and C. The basic simulations in arrangement A were carried out in a rectangular grid of 21000 cells. In order to obtain a good basis for comparison between the arrangements the airflow also was calculated in arrangement A in a grid with the same cells distribution as used in arrangements B and C. In one of the simulations in arrangement C (CHY2, see Table 1), the geometry of the pig simulator was removed from the model and instead the heat was released at floor level.
Table 1
Overview of the simulations. The first letter in the simulation identification denotes the arrangement (A, B or C), the second letter denotes the airflow rate (H for high and L for low) and the third letter denotes heat supply (Y for yes and N for no). A number is used if different grid constructions was investigated

<table>
<thead>
<tr>
<th>Simulation</th>
<th>Arrangement</th>
<th>Airflow rate m³/s</th>
<th>Heat supply</th>
<th>Grid description</th>
<th>X</th>
<th>Y</th>
<th>Z</th>
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<td>21 000</td>
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<td>No</td>
<td>Rectangular</td>
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<td>32</td>
<td>60</td>
<td>96 000</td>
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<tr>
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<tr>
<td>BHIN</td>
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</table>

The non-rectangular grids has 95 040 cells if it includes the geometry of the pig simulators and 96 000 cells if the simulators are removed. Figure 2 shows the grid distribution on room surfaces using the hexagonal and the non-hexagonal grids.

3. Results and discussion
3.1. Arrangement A
3.1.1. Airflow pattern
Figure 3 compares measured and simulated airflow patterns. For both ventilation rates in arrangement A it can be seen that measurement and simulation agree on that the jet adhere to the ceiling nearly the entire way to the rear wall. In the experiment, however, a small recirculating zone beneath the ceiling close to the end wall was observed, and in simulations small regions with flow opposite to the inlet direction were developed in middle between guiding plates, but, close to the guiding plates the air followed the ceiling to the end wall. In two-dimensional cfd-simulations Chen (1996) found that the $k-e$ turbulence model has a tendency to underpredict recirculating zones and that the anisotropic Reynold-stress turbulence model is better to predict such zones. In this study, additional simulations confirmed this observation, but in three-dimensional simulations the Reynolds-stress turbulence model did not significantly improve the prediction of the recirculating zones in arrangement A.

The recirculating zones in arrangement A were small and, therefore, the disagreement between measurement and simulation is only of minor importance for the condition in the occupied zone.

Fig. 2. Grid distribution in (a) the rectangular grid and in (b) the non-rectangular grid
3.1.2. Airflow direction

At high airflow rate, the measured airflow direction at $x = 6.5$, $y = 0.15$ and $z = 2.5$ was $181^\circ$ on average over a sampling period of 75 min. In a corresponding measurement in the same test room without guiding plates beneath the ceiling, the measured flow direction at the same point was $213^\circ$ (Bjerg et al., 1999). This indicates that the introduction of guiding plates beneath the ceiling had the desired effect because it made the airflow in occupied zone much closer to be two-dimensional than without guiding plates. The sampling period for the flow direction measurement was 3 s and the measured values from a period of 15 min are shown Fig. 4.

The calculated airflow direction at $x = 6.5$, $y = 0.15$ and $z = 2.5$ m was $180^\circ$ in all simulation in arrangement A.

3.1.3. Air speed

Figure 5 compares the simulated and measured air velocities at high airflow rate. The arrows show the measured values, the bold lines show simulated results with the rectangular grid and the thin lines show the
simulated results with the non-rectangular grid. Near the ceiling and near the floor, it can be seen that the simulated air speed is higher than the measured — the difference is typically 0.2 m/s. About 0.5 m from the ceiling and about 0.5 m from the floor, however, there is a much better agreement between measurements and simulations.

In a corresponding measurement in the same test room without guiding plates, a very good agreement between measured and simulated air velocities was found near the ceiling and near the floor (Bjerg et al., 1999). It is not clear why the introduction of the guiding plates leads to the over estimation of air velocities close to the ceiling and close to the floor in the simulation. A number of additional simulations with increased grid intensity close to the guiding plates and close to the ceiling did not result in a better agreement with measurement. Figure 5 shows that there were no significant differences between the relatively coarse rectangular grid and the finer non-rectangular grid.

3.1.4. Inlet slot height and airflow rate

The simulated and measured velocities at the low airflow rate are compared in Fig. 6. The difference between measurements and simulation is very much like what was found at high flow rate.

3.2. Arrangement B

3.2.1. High flow rate

Measured and simulated results for the high flow rate in arrangement B at non-isothermal conditions are shown in Fig. 7. Compared with the results from arrangement A, it can be seen that the pig simulators only had a minor effect on both the simulated and the measured air velocity at the fixed measuring point beneath ceiling and at floor level in the rear part of the room.

At floor level between the simulators and the outlet slot, there was a significant difference in the experimental results hence a recirculating zone was developed where the flow near floor was opposite to the return flow (see also Fig. 3 for sketches of the airflow patterns). In the simulations there were only small recirculating vertexes developed immediately behind the simulators. After that the return flow reattached to the floor. Also in the simulations, the return airflow penetrates through the openings between the simulators with relatively high velocities, but this was not seen in the measurements (see Fig. 7). The difference in the size of the recirculating zone and the difference in the penetration between the simulators is in agreement with the previously mentioned experience that the $k$-$\varepsilon$ turbulence model underpredicts the tendency to develop recirculating zones.

In both measurements and simulations, the air speed level was generally reduced in the occupied zone behind the simulators compared to the set-up without simulators. 20 cm above the floor the air speed was reduced about 50% which is in good agreement with experiments made by Smith et al. (1999).

At high flow rate, both measured and simulated air velocity was nearly unaffected irrespective of whether or not isothermal conditions were used. This is due to the relatively small ratio of the thermal buoyancy force to the inlet jet momentum. The thermal load in the room provides only 2.3 K difference over the supplying air in the tests with the high ventilation rate.

3.2.2. Low flow rate

In Fig. 3, it can be seen that both the measured and the simulated airflow pattern was unaffected by the reduction of the airflow rate from 0.36 to 0.14 m$^3$/s, which indicates that, even at the low flow rate, the airflow was mainly determined by the ventilation system and the room geometry, and that, the buoyancy forces only played a minor role in this arrangement.

3.3. Arrangement C

3.3.1. High flow rate

Simulated and measured airflow patterns at high flow rate were very uniform in Arrangement C (Fig. 3). Figure 8 compares the measured and the simulated air velocities — the bold lines show result of simulations CHY1 and the thin line shows results of simulation CHY2 where the geometry of the simulators was neglected. Compared with arrangement A or arrangement B the
Fig. 5. Comparison of measured and simulated air speed in arrangement A at high airflow rate without heat, the arrows show the measured results and bold lines show the results of simulation AHN1 (coarse grid); the results of simulations AHN2 (fine grid) is shown by the thin lines; x and z coordinates in m

air velocity beneath the ceiling (at \(x = 4.5\) and \(y = 2.98\) m) and near the floor in the back part of room was reduced with about 25% in both measurement and simulation. Both measurements and simulations showed a recirculating flow in the occupied zone between the transverse pen partition and the outlet slot, and there was a good agreement between measured and simulated air velocity profiles at \(x = 3.45\) where the return flow has passed the pen partition.

Figure 8 shows that there are only insignificant differences between heat release at floor level and from the simulated pigs and this indicates that a detailed geometric modelling of the animals in some cases might be unnecessary for simulation of airflow in livestock rooms,
Fig. 6. Comparison of measured and simulated air speed in arrangement A at low airflow rate. The arrows show the measured results and the lines show the simulation results; $x$ and $z$ coordinates in m

such as lying pigs near the pen partition wall. This is an important result because detailed geometric modelling of animals is very time consuming.

At high flow rate both measured and simulated airflow pattern was nearly unaffected by the heat sources.

3.3.2. Low flow rate

For non-isothermal conditions the airflow rate has a significant influence on the flow pattern in the experiment, see Fig. 3. The reduction of airflow rate from 0.36 to 0.14 m$^3$/s resulted in a considerable growth of the recirculating zone in the back end of the room. In the simulation the reduction of the airflow rate caused minor changes in the flow pattern near the rear wall, but the predicted recirculating zone was very small compared with measurement, and this is significant because it influences the possibility to predict the condition in the occupied zone.
The airflow rate had no significant influence on the airflow pattern at isothermal conditions, and this shows that it is the combined effect of heat load and the reductions of efficient room height by the introduction of pen partition that caused the observed growth of the recirculating zone.

### 4. Conclusions

The effects of pen partitions and thermal pig simulators on airflow in a livestock test room was investigated by measurements and computer fluid dynamic (CFD) simulations. To achieve a reasonable interpretation of the results it was decided that the reference case for the investigation should be a two-dimensional airflow near the floor in the empty room. Both measurements and simulations showed that guiding plates beneath the ceiling were efficient to create the two-dimensional airflow in the occupied zone. But the introduction of guiding plates increased the differences between measured and simulated air velocity close to the ceiling and close to the floor. The simulations showed no significant differences between the relatively coarse rectangular grid and the finer non-rectangular grid.

Both measurements and simulations showed that the introduction of pen partition and thermal pig simulators reduced the air velocities in the occupied zone of the test room.

Detailed geometric modelling of the animals might in some cases be unnecessary for simulation of airflow in livestock rooms. This will especially be the case when the animals are located close to pen partitions or other large-scale obstacles in the occupied zone.

The poor ability to predict recirculating zones sets limits to the expected precision of calculations with the \( k-\varepsilon \) turbulence model in livestock rooms where recirculating zones often occurs. Further on the poor ability to predict recirculating zones makes it also difficult to investigate the boundary conditions of heat supply in a ventilated room, because it is impossible to separate errors caused by these boundary conditions from errors caused by the poor ability to predict recirculating zones.
Fig. 8. Comparison of measured and simulated air speed in arrangement C at high airflow rate (none isothermal); the arrows show the measured results, the bold lines shows the simulation results of simulation CHY1 (heat release from pig simulators) and the thin lines shows the results of simulation CHY2 (heat release from floor); all dimensions and x and z coordinates in m

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