Heat transfer from black-steel pipe in a functionally similar barn environment

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Licsko, Z.J., Feddes, J.J.R., Leonard, J.J. and Darby, D.E. 1990. Heat transfer from black-steel pipe in a simulated barn environment. Can. Agric. Eng. 33:341-345. Heat transfer from black-steel pipe was measured in an environmentally-controlled chamber that simulated typical air patterns in a barn environment. Two commonly-used sizes of pipe (38-mm and 51-mm ID) were tested at three wall orientations and one ceiling orientation. Design heat loss equations for a single pipe were used as a reference for comparing heat losses from a bank of pipes. Heat losses from the wall-mounted 51-mm ID pipe were overestimated (9-19%) while heat losses from the ceiling-mounted 38-mm ID pipe were underestimated (5-7%). The design heat loss equation accurately predicts heat loss from 51-mm pipe for a ceiling-mounted orientation since this orientation acts most similarly to a single pipe. Dust deposits on pipes and minimal air circulation were found to cause heat loss reductions of up to 10%.

Le transfert de la chaleur du tuyau en acier noir était mesuré dans une chambre ou les mouvements de l'air sont typiques de ceux dans l'environnement d'une grange. Deux grands tuyaux (38-mm et 51-mm diamètre intérieur) communément utilisés ont été soumis à des essais aux trois orientations sur le mur et une sur le plafond. Des équations de transfert de chaleur pour un tuyau étaient utilisées comme référence pour comparer les pertes calorifiques d'un groupe de tuyaux. Les pertes calorifiques du tuyau 51-mm diamètre intérieur monté sur le mur étaient surestimées (9-19%) et les pertes calorifiques du tuyau 38-mm diamètre intérieur étaient sous-estimées (5-7%). Le design equation de la perte calorifique prédit exactement la perte calorifique du tuyau 51-mm pour une orientation sur le plafond puisque cette orientation agit le plus comme un seul tuyau. Les dépôts de poussière sur les tuyaux et le circulation de l'air minimal ont contribué aux réductions de 10% en la perte calorifique.

INTRODUCTION

Black-steel pipe is used for heating purposes in a variety of livestock barns. Some important advantages in its use are:

1. Few maintenance requirements,
2. Even heat distribution throughout the barn,
3. Ruggedness and ease of installation, and
4. Low cost.

Total heat loss from black-steel pipe currently is determined by utilizing equations that calculate, for idealized conditions, both the radiant and convective heat losses for a particular heated surface (IHVE 1976; ASHRAE 1989). These data are primarily intended for industrial and residential applications and deal with simplified cases such as single pipes or vertical banks of horizontal pipes. The performance of these heat transfer equations in barn environments with different arrangements of pipes and high air exchange rates is not well understood. Actual heat transfer from black-steel pipe in a barn will be affected by the pipe-wall clearance (Feddes et al. 1984), by the proximity of other heated pipes or heated surfaces, by air movement and by other environmental factors, such as drafts from forced air circulation fans which may cause substantial forced convective heat losses from heating pipe. Radiant heat losses, which can account for approximately half of the total heat loss from black-steel pipe, can be affected by the pipe's surface emissivity, and the heat transferred to the room can be affected by poor insulation of outside walls.

The objective of this paper was to determine how closely published values for total heat loss from a single black-steel pipe (ASHRAE 1989) compared with heat losses from a bank of pipes, under simulated animal housing conditions, when considering the effects of pipe orientation, pipe surface emissivity and settled dust.

EXPERIMENTAL FACILITIES

Experimental heat loss values for black-steel pipe must be obtained from a facility that replicates the environmental conditions found in a typical livestock barn. An environmental chamber at the Ellerslie Research Station was used for this purpose (Leonard and McQuitty 1986) (Fig. 1). The chamber was of insulated wood frame construction and was housed within the confines of a larger room where air temperature could be controlled. A 50-mm, continuous slot inlet and a model 562-2 Zehl-Abegg variable speed fan provided air exchange to the chamber. Since no animals were housed in the chamber, the effect of animals on room air movement was simulated by covering a representative floor area with heating pads maintained at temperatures comparable to livestock surface temperatures.

A mobile laboratory trailer and the instrumentation used in whole house calorimetry studies (Feddes and McQuitty 1981) were used for monitoring and data acquisition.

INSTRUMENTATION AND DATA COLLECTION

The hot water heating system for the experimental facility consisted of an electrically-heated boiler, control valves, a circulating pump and a test section of black-steel pipe (Fig. 2). The boiler was heated using four 1500 W electrical heating elements. The operation of the four elements was computer-controlled to maintain the desired set point temperature of the water entering the pipe test section.

The black-steel pipe used in this experiment was manufactured by Stelco Inc. according to specifications outlined in the American Society for Testing and Materials (ASTM) standards A120 & A53. Specifically, the pipe used was schedule
NOTE: All components except Pipe Test Section are wrapped in insulation.

Fig. 2 Schematic of heating system.

40 black-steel pipe with inside diameters of 51 mm and 38 mm. Emissivity of the pipe was 0.95 as measured with an infrared pyrometer (Megascope, Model 200, Omega Engineering, Stanford, CT). The test section of pipe was approximately 18 m in length. A turbine type flowmeter (ITT Barton, Edmonton, AB) was used to measure flow through the test section of black-steel pipe and had an accuracy of 0.01 L/s. The flowmeter was located in accordance with the manufacturer’s specifications.

To measure temperature, curve matched thermistors (UUB311J, Fenwal Electronics Inc., Framingham, MA) with a manufacturer-specified accuracy of 0.2°C were used. Thermistors were located in the environmental chamber as follows:

1. two in the air inlet of the environmental chamber,
2. one at the exhaust duct,
3. one on the pipe surface at the start and one at the end of the test section of black-steel pipe to measure water temperature. These thermistors were covered by 100 mm of fiberglass insulation wrapped around the pipe.
4. 36 in the environmental chamber.

Thermistors in the environmental chamber were located along the center line of the chamber to provide a cross-sectional representation of the room’s temperature. Thermistors were also used to measure the surface temperature of each of the heating pads. The wall, ceiling, floor and heating pad surface temperatures were measured with the infrared pyrometer. The mean radiant temperature was determined from the surface temperatures and their respective areas. This temperature was found to be similar to that of the room (20°C) and hence the measured room temperature was used as the mean radiant temperature.

EXPERIMENTAL PROCEDURES

To determine the heat output from black-steel pipe for a variety of pipe orientations typical of livestock building heating systems, test sections of 51-mm and 38-mm pipe were mounted below or adjacent to the air inlet slot as follows: (A) a vertical bank of four pipes, (B) four pipes each offset one diameter and (C) four pipes each offset two diameters (Fig. 3). Within the bank, pipes were vertically spaced at 200 mm o.c. The heat loss from a ceiling-mounted orientation (H) also was determined. In addition to determining the heat loss from bare black-steel pipe, heat output was determined for one orientation (38-mm pipe, orientation A) in which the pipe was covered with as much grain dust as would stay on the pipes: approximately 5-mm. This was done to simulate the dust accumulation that occurs when heating pipes are not regularly cleaned.

Data were gathered during experimental runs, which were replicated three times. In each experimental run the heat loss was measured from a particular pipe size and orientation at a particular temperature. The air inlet speed was maintained at about 5 m/s to simulate typical barn conditions. In addition, to determine the effect of inlet air speed on total heat output for orientation A, the ventilation fan was shut off and the air inlet was covered to create nearly stagnant air conditions. Smoke pencils were used to obtain qualitative data on airflow around the pipes.

In calculating the total heat loss from the test section, the average heat loss over a 500-second sampling period was assumed to accurately represent steady state conditions. The
The equation used for this is:

\[ Q = m C_w (T_i - T_0) \]  

where:

- \( Q \) = total heat loss from pipe test section (W),
- \( m \) = mass flow rate of water (kg/s),
- \( C_w \) = specific heat capacity of water (J/kg°C),
- \( T_i \) = inlet water temperature (K), and
- \( T_0 \) = outlet water temperature (K).

Several factors were considered in calculating heat loss. Since the flowmeter measured water volume, corrections were made for the change in mass flow with measured water temperature. Similarly, corrections were made for the change in specific heat capacity of water with temperature.

### Table 1

<table>
<thead>
<tr>
<th>Hot Water Temp.</th>
<th>51°C</th>
<th>50°C</th>
<th>49°C</th>
<th>48°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>135°C</td>
<td>0.81</td>
<td>0.83</td>
<td>0.83</td>
<td>1.00</td>
</tr>
<tr>
<td>140°C</td>
<td>0.82</td>
<td>0.83</td>
<td>0.83</td>
<td>1.00</td>
</tr>
<tr>
<td>145°C</td>
<td>0.83</td>
<td>0.83</td>
<td>0.83</td>
<td>1.00</td>
</tr>
</tbody>
</table>

Theoretical Heat Loss

The specific heat capacity of water with temperature can be calculated as:

\[ C_w = \frac{m}{\Delta T} \]

where:

- \( C_w \) = specific heat capacity of water (J/kg°C),
- \( m \) = mass flow rate of water (kg/s),
- \( \Delta T \) = change in temperature (°C).

### THEORETICAL HEAT LOSS

The theoretical heat loss from black-steel pipe has both radiant and convective components. Design equations for predicting convective heat transfer coefficients for pipes are based on carefully controlled experiments where the apparatus ensured that the fluid properties surrounding the test section of pipe were uniform (Rand et al. 1977). The contribution of each mode of heat transfer is difficult to calculate accurately if only measured heat losses were considered. Therefore, a direct comparison of total heat loss values for different orientations, pipe sizes, and temperatures was not possible because of the effect of different chamber temperatures on the measured heat loss.

### 38-mm Pipe

<table>
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<tr>
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</tr>
</thead>
<tbody>
<tr>
<td>135°C</td>
<td>0.79</td>
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</tr>
<tr>
<td>140°C</td>
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<td>0.83</td>
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</tr>
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<td>0.83</td>
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### 35-mm Pipe

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<th>49°C</th>
<th>48°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>135°C</td>
<td>0.84</td>
<td>0.83</td>
<td>0.83</td>
<td>1.00</td>
</tr>
<tr>
<td>140°C</td>
<td>0.83</td>
<td>0.83</td>
<td>0.83</td>
<td>1.00</td>
</tr>
<tr>
<td>145°C</td>
<td>0.83</td>
<td>0.83</td>
<td>0.83</td>
<td>1.00</td>
</tr>
</tbody>
</table>

### Table 1: Ratio of total measured to theoretical single pipe heat loss and total measured heat loss

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<td>1.00</td>
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### Table 2

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total heat loss from the black-steel pipe. To facilitate the comparison of different runs and orientations, total heat loss from the black-steel pipe was standardized in Table I as a ratio of the actual measured total heat loss to the calculated theoretical heat loss for a single pipe. The theoretical total heat loss for a single pipe provided a convenient standard against which other configurations could be compared. It was determined using temperature data collected during the experimental runs and the following equations for convective and radiant heat losses.

For free convective heat loss from a single pipe, ASHRAE (1981) and IHVE (1970) recommend the use of Eq. 2.

\[ q_c = 1.32 \frac{[T_{ps} - T_a]^{1.15}}{D_o} (T_{ps} - T_a) \]  

(2)

where:

- \( q_c \) = heat transfer by natural convection (W/m\(^2\)),
- \( T_{ps} \) = temperature of pipe surface (K),
- \( T_a \) = temperature of ambient air (K), and
- \( D_o \) = outside diameter of pipe (m).

Several variations of Eq. 2 have been derived experimentally and follow the same general format (Rand et al. 1977). Radiant heat loss from a single pipe, based on the Stefan-Boltzmann Law, was calculated using Eq. 3 (IHVE 1970):

\[ q_r = \varepsilon k (T_{ps}^4 - T_{mr}^4) \]  

(3)

where:

- \( q_r \) = heat transfer by radiation (W/m\(^2\)),
- \( \varepsilon \) = pipe emissivity,
- \( k \) = Stefan-Boltzmann constant (5.67 x 10\(^{-8}\) W/m\(^2\)K\(^4\)), and
- \( T_{mr} \) = mean radiant temperature of surrounding surfaces (K).

**EXPERIMENTAL RESULTS AND DISCUSSION**

The effect of increasing water temperature on total heat loss for orientation A was determined before the main part of the experiment was undertaken. This was done to ensure that the change in total heat loss was linear within the normal operating range of 50 to 100°C. The linearity of total heat loss from the pipe test section over this range is shown in Fig. 4. Having established this, subsequent tests of the total heat loss from the different orientations and pipe sizes were conducted at only two water temperatures: 50°C and 90°C. Means and standard deviations of heat loss measurements for the three replicates of each run are summarized in Table I. For each pipe size, orientation, and water temperature, heat loss is expressed as the ratio of measured heat loss over theoretical heat loss for a single pipe. The contribution of \( q_r \) and \( q_c \) was found to be similar when using measured data in Eqs. 2 and 3. With 51-mm pipe, the effect of pipe orientation on total heat output between the three wall-mounted orientations (A, B and C) at the same temperature was found to be insignificant. Orientations B and C were not tested with the 38-mm pipe since no difference could be measured between these two orientations and the A orientation for the 51-mm pipe. However, there was a difference between the three wall-mounted orientations of the pipe and the ceiling-mounted pipe for both water temperatures (Table I). For the wall-mounted orientations, the reduction in total heat loss was approximately 10% at 50°C and 17% at 90°C. The total heat output for the ceiling-mounted orientation (H) agreed closely with that predicted by theory for a single pipe suggesting there was little heat transfer interference between the two pipes. Also, ratio differences occurred between the high and low temperatures for the A, B and C orientations.

The similarity in heat loss between the three wall-mounted orientations is attributable to insufficient offset between pipes at different levels to prevent interference in the convective air flow from adjacent pipes. Figure 3 shows the air flow patterns, as observed during smoke tests, for these three orientations of pipe. For orientations B and C, the convective air flow tended to follow the angle of the pipe orientation rather than moving straight up. This pattern of convective air flow was stronger for the 51-mm pipe than for the 38-mm pipe.

The measured heat loss from the smaller diameter pipe for the ceiling-mounted (H) orientation at both temperatures was somewhat higher than that predicted from heat loss equations. A similar result was obtained for orientation A at the lower temperature (Table I). Draught-free surroundings are unlikely to occur and any air velocity has a larger effect on heat emission from smaller pipes (IHVE 1976). For orientation A at the higher temperature, some inter-pipe effect resulted in a lower value for heat loss than that of a single pipe. A larger vertical spacing for the 51-mm pipe may reduce this inter-pipe effect and result in a larger total heat loss.

The effect of dust on the heat loss of a wall-mounted orientation (A) of the 38-mm pipe also was investigated. When the maximum amount of grain dust was deposited on the pipes, the total heat loss was reduced by about 10% (Table I). Although data for the 51-mm pipe were not collected, a similar reduction in total heat loss would be expected.

The effect of near-stagnant air conditions on total heat loss from black-steel pipe was also determined. However, it was impossible to achieve steady state conditions at the higher test temperature since the environmental chamber’s ambient temperature rose too quickly. Consequently, only data collection at the lower water temperature was possible. Results showed a large difference in ratios between total heat loss in ventilated versus unventilated conditions (Table I). Under unventilated air conditions the total heat output was about 10% less than...
under ventilated conditions. Conditions in the unventilated case were not perfectly stagnant and so it would be misleading to explain the difference simply in terms of free and forced convection. Nevertheless, the reduction in total heat loss under unventilated air conditions is attributed to a reduction in airflow over the pipes leading to reduced convective heat losses.

SUMMARY AND CONCLUSIONS

The following conclusions were drawn from this study:

(1) Current design equations for a single pipe accurately predict heat loss from 51-mm ID pipe for a ceiling-mounted orientation (B).

(2) Current heat loss equations underestimate measured heat loss for the 38-mm ID pipe (5-7%), except for orientation A at 90°C.

(3) Current heat loss equations overestimate heat loss for the wall-mounted orientations of 51-mm ID pipe at water temperatures of 50 and 90°C by 9 to 19%.

(4) Dust reduced total heat loss by about 10% from 38-mm ID pipe.

(5) An inlet air velocity of 5 m/s, typical of barn conditions, results in approximately 10% higher total heat transfer than that of stagnant air.

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REFERENCES


