A TECHNIQUE FOR MEASURING PRODUCT HEAT LOAD DURING BEEF CHILLING

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ABSTRACT

The refrigeration of beef carcasses is an important user of energy. Due to the carcass's complicated shape and composition, it is difficult to calculate the rate of heat release as it evolves with time during the cooling process. A method has therefore been devised to measure the latent, sensible and radiative heat loads from beef sides. The latent heat load is measured within 3% error by suspending the beef side on a load cell measuring to ± 0.15 ppm. The sensible and radiative heat load is measured within 5% error using a flow-calorimeter type movable wind tunnel constructed to hold one beef side. Due to the small magnitudes of the weight losses, relatively large fluctuations in the weight loss curve occurred and it was necessary to smooth the curve by curve-fitting a three-parameter equation, using an evolutionary algorithm. Typical results are shown. These results are much more precise than results obtained earlier with compensatory heating methods.

NOMENCLATURE

a, b, c Parameters in the weight loss equation
$c_p$ Specific heat capacity of air (J/kgK)
$\Delta T$ Temperature change over product (K)
$\Delta V$ Voltage difference across thermopile (mV)
$h_{fg}$ Latent heat of vaporisation (J/kg)
$M$ Mass flow rate (kg/s)
$q$ Product heat load (W)
$\text{RH}$ Relative humidity (%) 
t Time (s)
$W$ Cumulative weight loss (kg)

Subscripts

air Property of air
Lat Latent
Rad Radiative
Sens Sensible

1. INTRODUCTION

The beef processing industry is one of the major users of refrigeration in Australia. The heat load imposed on beef refrigeration systems is comprised of several sources including the product, heat from fans and lights, infiltration through insulation, air exchange through open doors, and cooling of room structures. For chilling and freezing processes the product heat load is usually the largest load.

The rate of product heat release is variable in batch systems, exhibiting a peak heat load during the initial stages of cooling. If a refrigeration system is not designed to remove the peak load, the product may fail to meet quality and health specifications. It is difficult to predict the change in product heat load with time during beef chilling, due to the complicated shape and composition of beef sides, hence an accurate method of measuring the product heat load is required to verify prediction methods, and for use in empirical models.

A number of methods have been published for measuring heat load, including calibration of the compression cycles to the heat load (Cooper, 1969), and the use of clip-on power meters connected to the compressor electrical supply (Collett and Gigiel, 1986; Gigiel and Collett, 1989; Drumm et al., 1992). James and Bailey (1981) used a refrigerated room as a form of differential calorimeter, applying a constant cooling capacity to the room, and maintaining the air temperature using a heater. The product heat load was back calculated from the change in the energy supply to the heater during product cooling. Lovatt et al. (1992) encountered a number of difficulties with this approach which were overcome by adapting the principle of a flow calorimeter to heat load measurement.
Methods using calorimetry reported the most accurate results due to their direct measurement of the product heat load, rather than measurement of the total system. However, as they were based on changes in the air temperature, the calorimetry methods only measured the sensible heat load.

It is important to account for the latent heat load as well as the sensible heat load, as weight loss from beef sides during chilling has been reported to range between 1.2-2.3% (Herbert et al., 1978; Collett and Gigiel, 1986; Allen et al., 1987; Gigiel and Collett, 1989; Drumm et al., 1992).

The objective of this paper is to devise an accurate method to measure the total heat load evolved from beef sides during chilling, including sensible, latent and radiative heats.

2. HEAT LOAD MEASUREMENT SYSTEM

2.1 Sensible Heat Load

The sensible heat load is the heat lost from the product which contributes to a rise in the air temperature. Lovatt et al. (1992) applied the principle of a flow calorimeter to sensible heat load measurement from sheep and lamb carcasses and cartons of methylcellulose gel during freezing. They measured the air temperature upstream and downstream of the product, which was situated in a controlled environment tunnel, as well as the air flow rate. This allowed calculation of the sensible heat load from eq. (1).

$$q_{\text{sens}} = c_{\rho \text{,air}} M_{\text{air}} \Delta T_{\text{air}}$$  (1)

The proposed measurement system is based on the principle of Lovatt et al.'s system, with several modifications to improve heat load measurement accuracy and application.

The main modification was the use of a mobile wind tunnel inside a chiller room. This kept the application of the system flexible, as the tunnel could be used in existing chillers, and the construction costs minimal.

Another advantage was that the system was essentially adiabatic as the temperature inside the tunnel was nearly the same as that in the main chiller room. This removed a major source of error reported by Lovatt et al.: the infiltration of heat through the tunnel walls.
Furthermore, heat load error due to air infiltration was minimised, because the pressure difference between the working section and the main chiller room was negligible, and the temperature difference was also small.

The wind tunnel was designed to hold one beef side, with the main tunnel section dimensions: 650 mm × 1100 mm × 2775 mm. A schematic drawing of the wind tunnel is given in fig. 1. The tunnel walls were constructed of 50 mm thick polystyrene sandwich panel to minimise heat loss to the main chiller room.

2.1.1 Temperature Array

The temperature difference over the product was measured using a thermopile, consisting of 18 pairs of copper-constantan thermocouple junctions (KO258, 28 gauge), spaced evenly across the tunnel in an array of 3 × 6 junctions, located before and after the product. The temperature of the incoming air was also measured.

Every precaution was taken to avoid spurious signals in the thermopile circuit. No extra junctions were allowed, and care was taken to avoid mechanical stress in the wires. The thermocouples were shielded from radiative and evaporative effects by strips of PVC angle. These allowed good circulation of the air around the thermocouples, but protected the tip from the view of external radiative sources such as the lights, and from any condensation dripping from the rails.

The voltage output was logged with a Datataker DT500 datalogger with a resolution of 1 μV.

2.1.2 Air Flow

Good air flow over the beef side was achieved using an extraction fan, located at the base of the tunnel. A Zea FC 050-4EQ axial plate fan was installed, with a theoretical maximum volumetric air flow of 2.25 m³/s. The practical maximum flow rate was 0.861 m³/s due to the high pressure drop between the main tunnel section and the fan duct. A variable speed controller allowed fan speed adjustment.

A set of eight turning vanes were installed to improve the air flow between the bottom of the tunnel and the fan duct. The turning vanes were mounted on a removable frame to allow easy cleaning of the tunnel.

Air velocity was measured, and averaged, over an evenly spaced grid of 15 points across the tunnel cross-section, using a calibrated Omega HH-30 rotary vane anemometer (± 1%) for velocities above 0.3 m/s, and a TSI Velocical 8355-GB hot wire anemometer for lower velocities. The air flow rate was calculated as the product of the average air velocity and the tunnel cross-sectional area.

2.1.3 Tunnel Flexibility

The tunnel was built in two pieces to allow flexibility for use in different applications, as well as portability. The base section contained the turning vanes, fan duct and fan, while the top section contained the main tunnel section and door. A steel flange system, with rubber gaskets, was used to join the two sections together.

The front side of the tunnel (above the flange arrangement) was hinged as a door to allow carcass entry and sealed with rubber to prevent heat loss.

The tunnel was mounted on castors for mobility.

2.1.4 Air Relative Humidity

The relative humidity of the air was measured using a Vaisala HMD 20YB relative humidity probe (± 3 %RH). The relative humidity and air temperature were used to calculate the specific heat capacity and density of the humid air using the equations of Wong and Embleton (1984) and Giacomo (1982) respectively.

2.2 Latent Heat Load

The latent heat load is the heat lost due to evaporation of water at the product surface. The evaporative loss was measured by suspending the beef side on a load cell, and continually logging the weight change of the side throughout the cooling process. The latent heat load could then be calculated from eq. (2).
\[ q_{Lat} = h_{fg} \frac{\Delta W}{\Delta t} \quad (2) \]

2.3 Radiative Heat Load

As well as sensible and latent heat, the beef side also loses heat due to radiation to the tunnel walls. From the walls, some of the heat is transferred back to the air stream within the tunnel by convection, the rest is transferred through the sandwich panel to the main chiller room by conduction.

The temperature in the main chiller room is approximately the same as the air temperature within the tunnel so the apportioning of the amount of heat to each mode of heat transfer depends solely on the relative size of the heat transfer coefficients.

For 5°C air travelling at 2 m/s over the flat tunnel walls, the convective heat transfer coefficient is 4.75 W/m²K (Holman, 1992). The sandwich panel consisted of 0.05 m thick polystyrene, which at 5°C has a thermal conductivity of 0.027 W/mK (Durand, 1985), giving a conductive heat transfer coefficient 0.54 W/m²K. Hence more than 90% of the radiative heat is transferred by convection to the tunnel air stream, which means it is measured simultaneously with the sensible heat load, so eq. (1) becomes

\[ q_{Sens} + q_{Rad} = c_{p,air} M_{air} \Delta T_{air} \quad (3) \]

3. METHOD TESTING AND CALIBRATION

3.1 Sensible Heat Load

The wind tunnel was tested for calibration using a fan heater with 1200 W and 2400 W settings to provide a constant load. The heater did not impose any radiative or latent heat load.

The tunnel fan speed was set to give air velocities within the tunnel of 0.251 m/s, 0.435 m/s, 0.714 m/s, 0.890 m/s, and 1.139 m/s. At each velocity, the system, as described in section 2.1, was run without the heater load for about 25 minutes to establish a baseline voltage difference across the tunnel thermopile. The baseline voltage difference varied between 0.032 and 0.071 mV, and decreased with increasing air velocity. The heater was then set to the 2400 W setting, and the change in voltage across the thermopile, as well as the power drawn by the fan heater, logged for 30 minutes. The procedure was repeated using the 1200 W setting. The power was measured using a clip-on ammeter and voltmeter. A power factor of 1 was used since the effect of the heater fan on the air velocity and heat load was negligible.

The heat load measured by the thermocouple array system closely matched the power drawn by the fan heater. A theoretical value of the equivalent voltage difference was back calculated for the measurement of the power drawn by the fan heater. This value was used to check the calibration of the tunnel system.

Fig. 2 shows a comparison between the thermopile and theoretical voltage difference versus heat load curve for each velocity, with 5% error bars. All of the measured thermopile voltage differences were within 5% of the equivalent theoretical values, except for the 1200 W voltage difference at the 0.251 m/s air velocity.

A calibration curve for the tunnel was derived from the slope of the curves in fig. 2 plotted against the air velocity, as shown in fig. 3. The tunnel calibration curve was within 5% error of the theoretical curve for velocities above 0.251 m/s. Hence the tunnel apparatus was calibrated to a reasonable level.

Fig. 2 - Comparison of thermopile and
theoretical voltage difference vs heat load curves, for tunnel velocities of: a) 0.251 m/s, b) 0.435 m/s, c) 0.714 m/s, d) 0.890 m/s, and e) 1.139 m/s; with 5% error bars.

Fig. 3 - Calibration curve for the tunnel system, of the calibration slope (slope of the voltage difference vs heat load plot) plotted against the operating range of tunnel air velocities (5% error bars).

3.2 Latent heat load

The load cell was tested using a 20 kg load over a 70 hour period at air temperatures of 0°C and 5°C. The results showed less than ± 0.15 ppm variation at 0°C and ± 0.05 ppm at 5°C. Both showed no drift over the 70 hour period. The error in the load cell led to a worst case error of ± 3% in the latent heat load.

The load cell was also successful in monitoring small changes in weight, such as those due to evaporation.

4. BEEF CHILLING TRIALS

The wind tunnel and load cell set-up was moved to an experimental chiller room in a beef processing plant. A series of 70 trials were carried out using beef sides, measuring the heat load as described in section 2.

Chiller conditions were varied from air temperatures of -1°C to 10°C, with air velocities over the leg ranging from 0.6 m/s to 2.3 m/s. Product weight ranged from 80 kg to 190 kg, with AUS-MEAT P8 site fat scores between 1 and 5 (0 - 32 mm). Both central composite and one-factor-at-a-time experimental designs were employed.

4.1 Experimental Procedure

For each design condition the chiller room was equilibrated to the desired temperature. A Datataker DT500 data logger was used to log the voltage difference across the thermopile, the load cell output, and the temperature and relative humidity of the incoming air. Surface and centre temperatures for the leg, loin and shoulder of the beef side were also measured. Each measurement was averaged over a 5 minute interval.

The tunnel fan speed was set and the air velocity in the tunnel was measured. The air velocity measured was adjusted for use in the heat load calculations to account for the change in the tunnel cross-section due to the beef leg. It was assumed the air flow rate would be constant before and after product loading.

Fig. 4 - Heat load vs Time profile for an 88 kg side with a fat score of 3, cooled with air at 1.8°C, with a velocity of 1.276 m/s over the leg.
The system was logged with no product for a period of time before each beef trial to allow measurement of the baseline thermopile and load cell readings.

Each beef side was taken directly off the chain after hot weighing and washing, which was usually about one hour after slaughter. Once inside the chiller, thermocouples were inserted into the leg, loin and shoulder, and the side was moved into the tunnel where an initial weight was recorded. The side was usually in the chiller for 5-7 minutes before being loaded into the tunnel. Each side was chilled for 20-25 hours before unloading.

5. RESULTS AND DISCUSSION

Typical product heat load - time profiles are shown in figs. 4 to 7 for varying beef side types, and air conditions. General trends in the results were that the peak heat load increased with decreasing air temperature and side fatness, and increasing side weight and air velocity. The peak latent heat load was usually similar in magnitude to the peak sensible and radiative heat load, but the latent heat load dropped away quickly after the initial peak.

The sensible and radiative heat load curves were particularly sensitive to fluctuations in the entry air temperature. When the air temperature was reasonably constant very smooth curves were obtained, however, results were very noisy under poor temperature control, which was frequently the case with the available experimental chiller. The apparatus needed to be sensitive to detect small changes in the load, hence to obtain good results using this method it is imperative to achieve good chiller air temperature control.

The latent heat load was determined from the rate of weight loss curve. Direct differentiation of the load cell readings was impracticable, due to the random noise which was much larger than the weight change over the time interval used. It was therefore determined indirectly by fitting a smooth curve to the cumulative weight loss data and differentiating.

Several methods were attempted to fit the cumulative weight loss curve. The most accurate and robust method was to use a curve as
given in eq. (4). The form of this equation ensures that the curve has a finite slope at t = 0 and tends to a finite asymptotic value as t \to \infty.

Eq. (5) gives the differentiated form of eq. (4), which is the equation for the rate of weight loss.

\[
W = \frac{a}{1 + (b + t)} - \frac{a}{1 + b^c}
\]  
(4)

\[
\frac{dW}{dt} = -\frac{ac(b+b+t)^c}{1+(b+t)^2}
\]  
(5)

The parameters \(a\), \(b\) and \(c\) describe the shape of the half-sigmoid, and were found using the evolutionary error minimisation method of Pham (1994). The method provided good fit to the cumulative weight loss curve, as is shown in fig. 8, and allowed extrapolation back to time zero, which was important for the latent heat load calculation.

Another reason for the improved accuracy is that the proposed system takes the latent heat load into account, which is important during the initial stages of chilling. Latent heat load is measured within 3% error.

6. CONCLUSIONS

A method has been devised to measure the sensible, latent and radiative heat loads evolving from beef sides during chilling. The method uses a portable wind tunnel as a flow-calorimeter to measure the sensible and radiative heat components. The latent heat load is found by suspending the beef side on a load cell inside the tunnel. The method is more accurate than previous methods, and is capable of providing results within 5% error for most air flow rates for the sensible and radiative load, and within 3% for the latent heat load. However, the method is very sensitive to fluctuations in the air temperature, hence it needs to be used in situations with good temperature control.

7. ACKNOWLEDGMENTS

The authors would like to acknowledge the Meat Research Corporation of Australia for funding this work, Mr. Mark Davey and Mr. Neil McPhail for their assistance with the beef chilling trials, and IMTP and REMSERV Australia for their co-operation in the use of their beef sides.

8. REFERENCES


