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Refrigeration – a low energy process for refrigerating stored grains

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Abstract

The wet bulb temperature of the intergranular air in a bulk of stored grains has a most profound effect on the growth of insect populations and grain quality. In general, grain can be stored for longer if the wet bulb temperature of the intergranular air is lower. In temperate climate aerating the grains with ambient air is usually beneficial, but in tropical climates this is inadequate. An alternative is to refrigerate ambient tropical air before introducing it into the grain store. This is energy-intensive. In this research it is shown that air leaving the upper surface of a bulk of stored grains can be recirculated around the grain store through a grain-chilling unit in a process dubbed refrigaeration. *Ceteris paribus* this results in a lowering of the wet bulb temperature from 16.5 °C using refrigerated ambient air to about 6 °C when refrigeration is implemented.

Key works: refrigeration, stored grains.

Introduction

The need to protect stored grains from attack by biological agents and deterioration has been well documented by several authors. For the last three or four decades, postharvest researchers and practitioners have advocated the desirability of introducing physical, as opposed to chemical, methods of manipulating the stored grain environment so that grains are maintained in good condition. It can be argued that ambient aeration is the only physical method of protecting stored grains that has been widely adopted in Australia. The reasons are its simplicity and low running costs. Wilson and Desmarchelier (1994) advocate the adoption of ambient aeration because it helps to maintain grain quality, and it also forms an integral component of grain management systems that include the use of pesticides and fumigants.

It is important to appreciate that most of the biological and biochemical phenomena that occur in bulks of stored grains are functions of the humidity and temperature of the intergranular air. Ambient aeration attempts to manipulate these variables so that grain can be stored in good condition. Wilson and Desmarchelier (1994) demonstrated that the rates at which insect populations grow, pesticides decay and seeds lose their viability may be expressed as simple functions of the wet bulb temperature of the intergranular air. In general, the lower the wet bulb temperature, the slower are the rate processes.

Thorpe (2005) has proposed an improvement on ambient aeration that makes use of the inherent desiccant properties of a bulk of stored grains. Instead of aerating grain with ambient air Thorpe (2005) suggests that the air leaving the upper surface of a grain bulk be cooled, possibly by means of an air-to-air heat exchanger, before it is readmitted into the grain bulk. The motivation for implementing this strategy, known as recircaeration, is that air leaving a bulk of grain under Australian conditions has a relative humidity

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of about 50 %, whereas the relative humidity of air used for ambient aeration may be about 90 %. By using recircaeration a bulk of grain can ideally attain the dry bulb temperature of the ambient air. This represents an improvement on aeration because wheat initially with a moisture content of 11 % wet basis aerated with ambient air with an average dry bulb temperature of 22 °C and a relative humidity of 90 % will attain a wet bulb temperature of about 19.5 °C. This is a typical condition at harvest time, hence aeration would be of little immediate benefit in cooling grains in this climate. When the air is recirculated, Thorpe's (2005) analysis suggests that the wet bulb temperature of the intergranular air would be about 15 °C. Although this is a small decrease it nonetheless has a significant affect on the rates of increase of insect pests. Recircaeration appears to represent an improvement over ambient aeration the degree to which grains can be cooled is still dictated by the prevailing climatic conditions.

In climates where the minimum dry bulb temperature is in excess of 20 °C, say, for prolonged periods of time it may be necessary to artificially reduce the heat content of the air used to aerate grain.

The refrigeration process proposed in this paper is a hybrid of the refrigeration of thermally uninsulated silos and the recircaeration process described by Thorpe (2005). The underlying idea is that the wet-bulb temperature of chilled grains is lower than ambient temperature and this air can be effectively recirculated around a bulk of grains. A counter argument might be that during the day solar radiation will increase the temperature of the recirculated air to such a high degree that the system will be less effective that refrigerating ambient air.

The objective of this research is to use a simple one-dimensional model of the thermodynamic conditions within a bed of stored grains to investigate the feasibility of refrigaeration. Although the model is highly simplified it is believed that it captures sufficient detail to highlight the limitations of refrigeration and to indicate areas where further research should be conducted.

Physical realisation of refrigeration

The principal components of a refrigeration system are shown in Figure 1. Air from the upper surface of the grain is recirculated through the grain- chilling unit. The system exploits the fact that the wet bulb temperature of recirculated air leaving a bulk of grain may be much lower than ambient air, hence it takes less power to cool it than is required to cool ambient air. Furthermore, the recirculated air has a higher dry bulb temperature than its wet bulb temperature hence heat gains to the air may be less severe than believed *prima facie.*

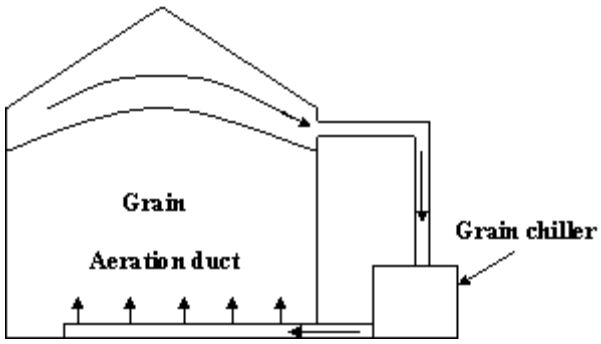

Figure 1. The principal components of a grain refrigeration system.

Heat transfer to the headspace of the grain silo

Figure 2 shows the principal flows of thermal energy around the headspace of a grain store. Heat is transferred to three entities, namely the roof, the air flowing through the headspace; and the grain; thermal energy balances must be written for each one.

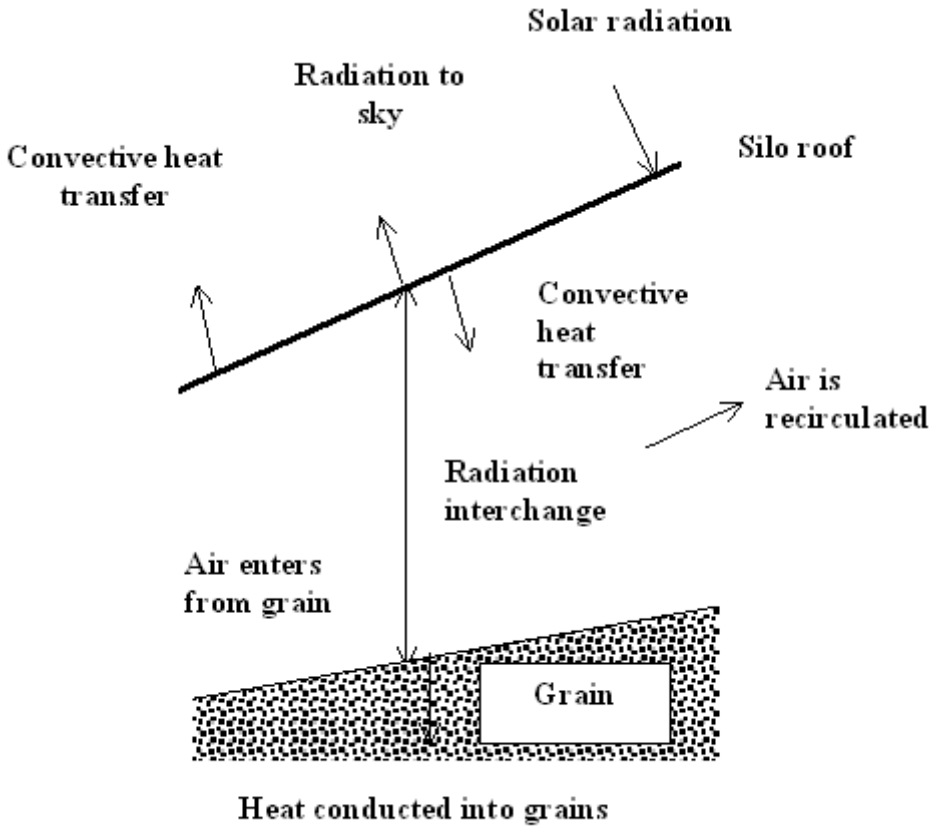

Figure 2. The principal heat flows around the headspace of a grain store.

Heat balance on the silo roof

The thermal energy balance on the roof can be expressed in words as

Solar energy absorbed by the roof	=	*Heat radiated to the sky*	+	*Heat lost by convection*	+	*Net heat radiated to the grain surface*

Algebraically this is written

$$\varepsilon_o Q_{solar} = \varepsilon_o \sigma \left(T_s^4 - T_{sky}^4\right) + h_o \left(T_s - T_{amb}\right) + h_i \left(T_s - T_i\right) + \varepsilon_i \varepsilon_g \sigma \left(T_s^4 - T_g^4\right) \Big/ \left(\varepsilon_i + \varepsilon_g - \varepsilon_i \varepsilon_g\right) \quad (1)$$

in which, $\varepsilon_o, \varepsilon_i$, and ε_g are respectively the emissivities of the outer and inner surfaces of the silo roof and the upper surface of the grain. Q_{solar} is the intensity of the total solar radiation impinging on the outer surface of the silo roof, σ is Stefan-Boltzman's constant, W/(m^2.K^4) and h_o and h_i are the convection coefficients between the upper surface of the roof and the atmosphere and the air in the headspace respectively, W/(m^2.°C). T_5, T_i, T_{sky} and T_g are the absolute temperatures K, of the silo roof, the air in the headspace, the sky, the upper surface of the grain bulk respectively.

Heat balance on the air flowing through the headspace

A heat balance on the air as it enters and leaves the headspace is expressed as

Rate of energy entering the headspace with the air leaving the grain	+	*Rate of energy leaving the headspace in the recirculated air*	+	*Rate of energy leaving the headspace by convection to the roof*	*= 0*

In this model it is assumed that the air in the headspace is perfectly mixed and that its temperature is T_i. This is a gross approximation, but sensitivity analyses on the rates of heat transfer between the headspace and its environment are carried out to assess the need to carry out more rigorous calculations.

Algebraically the heat balance becomes

$$f_a c_a \left(T_g - T_i\right) + h_i \left(T_s - T_i\right) = 0 \qquad (2)$$

in which f_a and c_a are respectively the flow rate of air per square metre cross section of the grain store, kg/(m^2.s), and the specific heat of air, J/(kg.°C).

Heat balance on the upper surface of the grain

The heat balance on the upper surface of the grain may be expressed in words as:

Net heat radiated to the grain surface	=	*Rate at which heat is conducted from the surface*

Algebraically this takes the form

$$\varepsilon_i \varepsilon_g \sigma \left(T_s^4 - T_g^4\right) \Big/ \left(\varepsilon_i + \varepsilon_g - \varepsilon_i \varepsilon_g\right) = k_{eff} \frac{dT}{dx} \qquad (3)$$

in which k_{eff} is the effective thermal conductivity of the bulk of grain.

Heat transfer coefficients

We note that the heat transfer coefficient, h_o, between the outer surface of the roof and the environment is a function of the temperature of the roof, T_5, and ambient temperature, T_{amb}. Kreider et al. (2002) report the following empirical relationship for the heat transfer coefficient, h_o

$$h_o = c_{\pm} \left|T_s - T_{amb}\right|^{1/3} \qquad (4)$$

in which the constants c_+ and c_- (denoted $c_{\pm}$) depend on whether or not the temperature of the roof is lower or higher than that of ambient. If the temperature of the roof is greater than that of ambient air the constant c_+ applies, whereas c_- is used when the roof is colder than ambient air. An analogous expression holds for the heat transfer coefficient, h_i, between the roof and the air in the headspace.

Equations 1, 2 and 3 can be used to define three functions, namely $\phi_1\left(T_s, T_i, T_g\right)$, $\phi_2\left(T_s, T_i, T_g\right)$ and $\phi_3\left(T_s, T_i, T_g\right)$. The three unknowns are the temperature of the surface of the roof, T_s, that of the air in the headspace, T_i, and the upper surface of the grain bulk, T_g. A Newton-Raphson search is an efficient method of solving these three equations and it is implemented by making use of the three functions

$$\phi_1\left(T_s,T_i,T_g\right)=\varepsilon_o Q_{solar}-\varepsilon_o\sigma\left(T_s^4-T_{sky}^4\right)-h_o\left(T_s-T_{amb}\right)-h_i\left(T_s-T_i\right)-e_e\left(T_s^4-T_g^4\right) \tag{5}$$

$$\phi_2\left(T_s,T_i,T_g\right)=f_a c_a\left(T_g-T_i\right)+h_i\left(T_s-T_i\right) \tag{6}$$

and

$$\phi_3\left(T_s,T_i,T_g\right)=\varepsilon_i\varepsilon_g\sigma\left(T_s^4-T_g^4\right)\Big/\left(\varepsilon_i+\varepsilon_g-\varepsilon_i\varepsilon_g\right)-k_{eff}\left(T_g-T_{gdx}\right)\Big/\Delta x \tag{7}$$

where T_{gdx} is the temperature of the grain at a distance $\triangle x$ below the upper surface of the grains. The variable, e_e, accounts for the interchange of radiation between the inner surface of the roof and the upper surface of the grain and bulk it is defined as

$$e_e=\frac{\varepsilon_i\varepsilon_g\sigma}{\varepsilon_i+\varepsilon_g-\varepsilon_i\varepsilon} \tag{8}$$

Equations 5 to 7 are expanded as truncated Taylor series, thus

$$\phi_1\left(T_s+\Delta T_s,T_i+\Delta T_i,T_g+\Delta T_g\right)\approx\phi_1+\Delta T_s\frac{\partial\phi_1}{\partial T_s}+\Delta T_i\frac{\partial\phi_1}{\partial T_i}+\Delta T_g\frac{\partial\phi_1}{\partial T_g} \tag{9}$$

$$\phi_2\left(T_s+\Delta T_s,T_i+\Delta T_i,T_g+\Delta T_g\right)\approx\phi_2+\Delta T_s\frac{\partial\phi_2}{\partial T_s}+\Delta T_i\frac{\partial\phi_2}{\partial T_i}+\Delta T_g\frac{\partial\phi_2}{\partial T_g} \tag{10}$$

and

$$\phi_3\left(T_s+\Delta T_s,T_i+\Delta T_i,T_g+\Delta T_g\right)\approx\phi_3+\Delta T_s\frac{\partial\phi_3}{\partial T_s}+\Delta T_i\frac{\partial\phi_3}{\partial T_i}+\Delta T_g\frac{\partial\phi_3}{\partial T_g} \tag{11}$$

in which we have omitted the arguments of ϕ_i, namely T_5, T_i, T_g, for brevity. When equations 5, 6, and 7 are satisfied ϕ_1, ϕ_2 and ϕ_3 are zero and we have a set of three linear algebraic equations in ΔT_s, ΔT_i and ΔT_g

$$\Delta T_s\frac{\partial\phi_1}{\partial T_s}+\Delta T_i\frac{\partial\phi_1}{\partial T_i}+\Delta T_g\frac{\partial\phi_1}{\partial T_g}=-\phi_1 \tag{12}$$

$$\Delta T_s\frac{\partial\phi_2}{\partial T_s}+\Delta T_i\frac{\partial\phi_2}{\partial T_i}+\Delta T_g\frac{\partial\phi_2}{\partial T_g}=-\phi_2 \tag{13}$$

and

$$\Delta T_s\frac{\partial\phi_3}{\partial T_s}+\Delta T_i\frac{\partial\phi_3}{\partial T_i}+\Delta T_g\frac{\partial\phi_3}{\partial T_g}=-\phi_3 \tag{14}$$

We also note that

$$\frac{\partial h_0}{\partial T_s}=\frac{c-}{3}\left(T_s-T_{amb}\right)^{-2/3}\text{ when }T_s>T_{amb} \tag{15}$$

and

$$\frac{\partial h_0}{\partial T_s} = -\frac{c_-}{3}(T_s - T_{amb})^{-2/3} \text{ when } T_s < T_{amb} \tag{16}$$

As noted above, the value c_+ is used when $T_5 > T_{amb}$ and the differential $\partial h_o / \partial T_s$ is positive because as T_5 increases for a given value of T_{amb} so does the heat transfer coefficient, h_o. However, when $T_5 < T_{amb}$ the heat transfer coefficient increases as T_5 decreases for a given value of T_{amb}. Similar expressions hold for h_i.

$$\frac{\partial \phi_1}{\partial T_s} = -4\sigma\varepsilon_o T_s^5 - 4e_e T_s^3 - h_o + \frac{\partial h_o}{\partial T_s}(T_{amb} - T_s) - h_i + \frac{\partial h_i}{\partial T_s}(T_i - T_s)$$

$$\frac{\partial \phi_1}{\partial T_i} = h_i + \frac{\partial h_i}{\partial T_i}(T_i - T_s); \qquad \frac{\partial \phi_1}{\partial T_g} = 4e_e T_g^3$$

$$\frac{\partial \phi_2}{\partial T_s} = h_i - \frac{\partial h_i}{\partial T_i}(T_i - T_s); \qquad \frac{\partial \phi_2}{\partial T_g} = f_a c_a \tag{17}$$

$$\frac{\partial \phi_2}{\partial T_i} = -h_i - \frac{\partial h_i}{\partial T_i}(T_i - T_s) - f_a c_a \qquad \frac{\partial \phi_3}{\partial T_s} = 4e_e T_s^3$$

$$\frac{\partial \phi_3}{\partial T_i} = 0 \qquad \frac{\partial \phi_2(T_s, T_g)}{\partial T_g} = -4e_e T_g^3 - k_{eff}/\Delta x$$

We are now in a position to solve equations 12, 13 and 14 for the temperatures of the surface of the roof, T_s, the air temperature leaving the headspace, T_i, and the upper surface of the grain bulk, T_g, by noting that improved values, T_s^{new}, T_i^{new} and T_g^{new} are given by

$$\left.\begin{aligned} T_s^{new} &= T_s + \Delta T_s \\ T_i^{new} &= T_i + \Delta T_i \\ T_g^{new} &= T_g + \Delta T_g \end{aligned}\right\} \tag{18}$$

The updated values of the unknowns are inserted into equations 12, 13 and 14 until ΔT_s, ΔT_i and ΔT_g all approach absolute values that are deemed to be sufficiently small, typically 10^{-5} °C.

Refrigeration in tropical climates

In this study the potential benefits of refrigaeration are explored by considering its operation in a tropical climate. The cooling loads on the fabric on a grain silo are quantified with the aim of assessing whether or not they are reasonable.

The essential features of the system investigated are summarised in Table 1.

The wet bulb temperatures of the interstitial air along a bed of grain cooled by refrigerating ambient air are shown in Figure 3. It can be seen that this conventional method of refrigerating grain results in an interstitial wet bulb temperature of about 16.5 °C, too high to achieve

effective control of insects. The difference in the wet bulb temperatures of the air entering and leaving the grain-cooling unit is about 9 °C, and under the specified operating conditions this is equivalent to an enthalpy difference of about 31.6 kJ/kg. On this basis the cooling capacity of the grain-chiller is 19 kW, or 38 W per tonne of grain cooled, and the degree of cooling is inadequate.

It can be seen from Figure 4 that when the air is recirculated around the grain store the interstitial wet bulb temperature is reduced to about 6 °C, which is eminently safe for grain storage. It is found that after 30 days of refrigeration the wet bulb temperature of the air entering the grain chilling unit varies from about 9 °C at night to 22 °C during the day; the higher value results from solar radiation heating the air as it leaves the grain. The air leaving the cooling coil varies between about 4 °C and 12 °C. The corresponding cooling capacities are 6 kW and 18 kW respectively.

During the day, when the maximum rate of cooling capacity is 18 kW, most of the heat enters the system through the roof of the grain store. Since this is a key element in the success or otherwise of the process one must determine whether it is reasonable that the average cooling

Table 1. Key operational details of the system investigated.

Dimensions of the grain bulk	8 m wide x 8 m long x 10 m high
Air flow rate	0.6 m^3/s
Average ambient temperature	30 °C
Amplitude of diurnal variation	5 °C
Ambient humidity	0.019 kg/kg
Maximum solar radiation on the roof	800 W/m^2
Emissivity of outer surface of roof	0.6
Emissivity of inner surface of roof	0.9
Emissivity of upper surface of grain	0.9
Initial grain moisture content	14 % (wet basis)
Initial grain temperature	30 °C

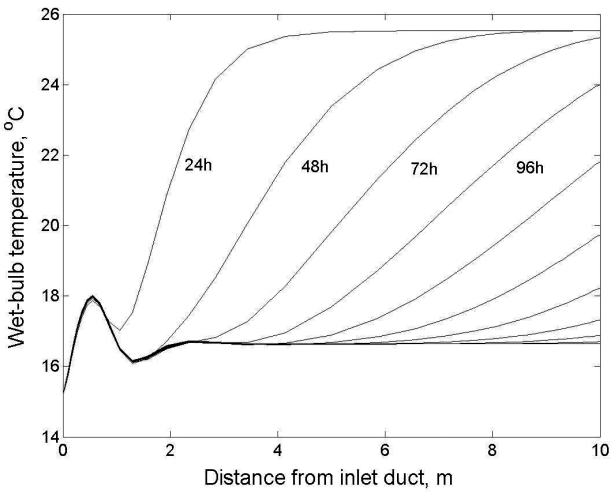

Figure 3. Wet bulb temperature profiles along a bed of aerated with refrigerated ambient air.

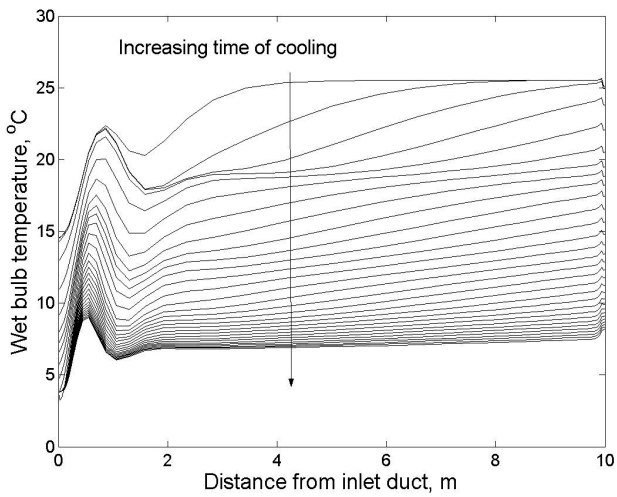

Figure 4. Wet bulb temperature profiles along a bed of aerated with refrigaerated air.

capacity is correct. It is found that the surface of the roof has a maximum temperature of about 64 °C, the temperature of the air in the headspace is 47 °C and the temperature of the air upper surface of the grains is about 44 °C. The heat gained by convective transfer to the headspace air is about 2kW and that gained by thermal radiation to the upper surface of the grain is about 9 kW. During the night, heat transfer through the roof of the silo is about 5 kW. These are of the same order as the cooling capacity of the chilling unit that is still removing heat from the bulk of grain.

The mathematical model was formulated so that the heat transfer coefficients can be readily changed, and it is shown that increasing the heat transfer coefficients by a factor of 5 has little effect on the overall performance of the system. The effect of increasing the absorptivity of the outer surface of the roof from 0.6 to 0.9 increases the wet bulb temperature of the interstitial air by about 1 °C. These results indicate that a grain-chilling unit that is capable of cooling grain in a tropical climate to an interstitial wet bulb temperature of 16.5 °C when ambient air is cooled can be used to refrigaerate grain to a wet bulb temperature of about 6 °C. This is regarded as significant and it is recommended that field trials be carried out to test the concept.

Conclusions

It is hypothesised that air leaving the upper surface of a refrigerated bulk of grain has a low wet bulb temperature. It is further hypothesised that when the air is heated in the headspace by solar radiation falling on the roof of a store the wet bulb temperature remains lower than that of ambient air in tropical climates. These hypotheses have been tested by formulating a mathematical model of the refrigaeration system which is defined as one in which the air is recirculated. It is shown that a grain-chiller that is capable of reducing the average wet bulb temperature of tropical air to 16.5 °C is capable of reducing grain to a wet bulb temperature of about 6 °C in a refrigaeration system.

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