PROJECT REPORT No. 269

OPTIMISING THE PERFORMANCE OF VERTICAL AERATION SYSTEMS

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by

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ABSTRACT

Cooling by aeration is the basis of safe storage but, rather then the conventional systems, blowing air upward from horizontal ducts, many farmers have preferred to use vertical duct systems where air is drawn down through the grain and exhausted from the duct. These vertical aeration systems raise a number of new problems.

Aims- The aims of this study were 1) to establish some principles of vertical airflow based on computerised fluid dynamics modelling, 2) to measure airflow rates, air distribution patterns and cooling speed profiles in several commercial vertical duct installations, 3) to make a systemic study of the effect of perforation size, outlet area and duct diameter on airflow and pressure drop in grain ventilation ducts and 4) to use the measured data to develop and validate a computer model of vertical ventilation systems and to use this to develop practical user guidelines on unit spacing.

Conclusions- Computational modelling showed that if the aerator fan is set to blow rather than suck then the volume of grain cooled is up to 20% greater and better cooling is obtained near the floor. The optimum spacing of vertical aerators was calculated for different grain bed depths. Measurements made in six grain stores with vertical ventilation systems showed that the volume of air moved by the fan was up to 20% more when it was blowing than when it was sucking. Diurnal temperature fluctuations were used to estimate the speed of the cooling zone which was linked to the pressure gradient and this relationship used to estimate cooling times . Using a purpose built test rig, it was shown that horizontal pressure drop was about 50% of that for vertical flow. Ducts with >30% outlet area offered little resistance to flow at working flow rates. Ducts with <10% outlet area restricted the airflow. Duct wall resistance was 16% lower when blowing wheat than when sucking. Seed size affected the duct outlet pressure drop in ducts with round outlets but had little influence on the pressure drop with slot outlets. A simple model, based on cooling front velocities and the pressure and flow characteristics obtained from the farm tests and trials with ducts was developed to enable routine design of new systems in a short time and a user guide has been drafted.

Implications- The development of systematic guidelines for the spacing of vertical ventilation systems will ensure effective cooling in all parts of the bulk and minimise capital investment by recommending the widest practical spacing and minimise fan power requirements.

SUMMARY

INTRODUCTION

The basis of good storage practice is adequate cooling and drying and HGCA have funded research to develop IPM systems for feed grain (Armitage et al., 1992) and malting barley (Armitage and Woods, 1997), based on cooling dry grain to as low a temperature as practical and top-dressing with a pesticide to deal with upward-moving insects and surface mite infestations caused by absorption of moisture by the grain surface layers. These strategies have been based on studies of horizontal aeration.

In order to conform to ACCS standards, there has been a recent upsurge in installation of aeration systems. However, rather then the conventional, systems, blowing air upwards from under-floor or under-grain horizontal ducts, many farmers have preferred to use vertical duct or 'pedestal' systems where air is drawn down through the grain and exhausted from the duct. These are manufactured by several firms, Lishmann, Brice-Baker and Polypipe Civils being the market leaders. The capital costs of vertical ventilation systems, based on current rule-of-thumb, are lower than conventional horizontal duct systems. Vertical systems are less prone to damage during loading out so they can be expected to have a longer service life. Vertical systems do not require holes to be cut into grain support walls and can be used effectively in grain bulks that are not level-loaded.

These vertical aeration systems raise a number of new problems. The air flow, working pressure and the region of effective cooling are difficult to predict. The novel duct system and direction of airflow are likely to affect moisture translocation and insect and mite distributions. Maximum economic benefit from these systems depends on selecting the largest practical spacing for the units and managing the ventilation to ensure effective cooling of the whole grain bulk by sharing fans between a number of duct units. Current practice is to base unit spacings on 'rule-of-thumb' and experience. Often suction is recommended for these systems on the basis of lower resistance and increased airflow, although in reality the resistance to suction or blowing is likely to be the same. In some cases there may be advantages to be gained from blowing.

This study sought to gather the technical data that is required to develop user guidelines for the available systems and to validate and quantify their ability to cool and dry grain. It would identify any advantages that attach to sucking and blowing.

The aims of this study were:-

- To establish some principles of vertical airflow based on computerised fluid dynamics modelling.
- To measure airflow rates, air distribution patterns and cooling speed profiles in several commercial vertical duct installations.
- To make a systematic study of the effect of perforation size, outlet area and duct diameter on airflow and pressure drop in grain ventilation ducts.
- To use the measured data to develop and validate a computer model of vertical ventilation systems and to use this model to develop practical user guidelines on unit spacing.

The development of systematic guidelines for the spacing of vertical ventilation systems can be expected to ensure effective cooling in all parts of the bulk, minimise capital investment by recommending the widest practical spacing and minimise fan power requirements.

DRAFT USER GUIDE TO VERTICAL AERATION SYSTEMS FOR COOLING GRAIN.

Background

The purpose of cooling is to preserve grain quality. The aims are to even out temperature gradients to prevent condensation and sprouting of grain at the surface as well as to reduce biological and biochemical changes in the crop and to control insect and mite infestation. Since mites and moulds both increase slowly at low temperature, cooling is mainly intended for low mc grain, below 14.5% mc. Some short-term protection of grain up to 18% mc may be provided by the use of cooling techniques Wet grain awaiting heated air drying in harvest backlogs should always be ventilated.

Vertical aeration

A relatively new development in aeration design over the last 25 years has been 'the vertical aeration system' sometimes described as 'Pedestals'. Typically, these comprise a 1m upright perforated section of duct spaced at intervals throughout the store and connected by extendible plain ducts to a centrifugal fans located just above the grain surface which usually exhausts the air. Alternatively air may be blown into the duct and conversion between sucking and blowing is easily arranged.

Advantages of vertical aeration

- Low capital costs (£2 /t) for cooling compared to built in under floor or horizontal above floor ducts
- Visibility of vertical ducts mean low likelihood of damage during unloading compared to above-floor horizontal ducts
- Suited to difficult shaped grain bulks and surcharged heaps

Cooling basics

The rate of aeration is critical. This guide outlines recommendations for duct spacing and fan selection that will deliver airflows of around 10 cu m/h/t to all the grain in a bulk. This ventilation rate will provide sufficiently rapid cooling to prevent insects completing their life cycle before the cooling front passes through the grain.

Cooling targets

- Cool to below 15-20°C within three weeks (100-150h aeration). This is easily achievable in the UK as mean daily temperatures rarely exceed 20°C. This prevents the fastest increasing insect, the saw-toothed grain beetle, completing its life cycle and reduces the temperature below its breeding threshold.
- Cool to 10°C as quickly as possible to prevent grain weevils developing.

Continue cooling to as low a temperature as possible (0-5°C by Christmas) to kill existing insects and prevent mites breeding.

Suck or blow?

Most vertical aeration systems allow the fans to be mounted to either blow or suck air through the grain. Each has its own attributes

Table 1. Advantages of blowing or sucking systems

Blowing	Sucking
• Cools up to 20% more grain than sucking	• Avoids condensation of warm moist air on a cold roof and grain
• Aeration can be started during loading without transferring heat from warm to already cooled grain	• Avoids excessive temperature rise in very deep grain beds
• Fans lower relative humidity of ventilating air, reducing re-wetting risk	• Any inlet dampening will be at the surface where it is easily monitored and treated
• Warmest grain, nearest the surface, is more easily monitored	• Cooling conforms to uneven grain surfaces
	• Ducts are unlikely to become blocked with dust

Moisture changes during cooling

The air flows used to cool grain only result in small moisture changes. To achieve drying at recommended rates, the airflow needs to be 10-20 times greater than for cooling. Drying will require the largest ducts and fans and much closer duct spacing than for cooling. Exceptionally, a cooling system may result in some drying where:-

- The grain is initially very hot. Cooling grain reduces mc by about 0.25% for every 5°C drop in temperature, a process known as 'dryaeration'. Cooling from 35-15°C will therefore lower the mc by about 1% mc.
- With a small head space above the grain, solar gain may allow a drying front to be slowly drawn into a bulk in a suction system.

DESIGNING A VERTICAL VENTILATION COOLING SYSTEM

Selecting systems

It is important to install enough aeration ducts in a grain heap to be sure that all parts of the heap can be cooled fast enough to control insect pests and prevent deterioration of grain quality. Commercially available units are available in a number of sizes. The quantity of grain that can safely be cooled by a unit depends on the size of the perforated duct section, the area of the perforations and the fan that is used with it.

Duct wall resistance to air flow

Part of the pressure developed by the fan is used to overcome the resistance of the duct wall/grain interface. Ducts with more than 20% open area offer virtually no resistance to air movement. Ducts with less than 5% open area offer a significant resistance and ducts with less than 1% will offer serious restriction to the air flow.

Increasing the pressure developed by the fan will increase the quantity of grain that a duct can serve. For practical purposes operating pressures are limited to below 1000 Pa. (100 mm wg). Higher pressures require more power and in blown systems the warming of the cooling air reduces the available cooling time. Most commercial systems work at about 700 Pa.

How many units?

The number of units required to serve a bulk of grain can be estimated by choosing a ventilation unit of appropriate scale and dividing the total tonnage by the quantity of grain that the selected unit can cool. In general it is best to use the same size units throughout a store so that one size of fan will fit all ducts. Table 2 shows the tonnage of wheat that can be cooled by 100 hours ventilation in a 3m deep bulk with each of the ducts that were tested.

Table 2. Grain quantity (tonnes) cooled by a range of duct sizes working at 700 Pa.

Duct Size	**Blowing**	**Sucking**
Metal 250mm x 960mm	103	85
Metal 300mm x 960mm	109	91
Plastic 350mm x 1200mm	95	77
Plastic 500mm x 1200mm	116	93

Spacing the units

In a rectangular grain bulk each ventilation unit has to serve a cuboid parcel of grain. The maximum distance between units should be chosen so that the cooling time along the longest path between units is no more than 100 hours.

Table 3. Duct centres (m) for 100 hr cooling time working at 700 Pa.

Depth (m)	**2**	**3**	**4**	**5**	**6**	**7**	**8**
Duct type							
Blowing							
Metal 250 x 960	7.6	6.4	5.6	4.9	4.5	4.0	3.7
Metal 300 x 960	7.8	6.6	5.7	5.1	4.6	4.2	3.8
Plastic 350 x 1200	7.3	6.2	5.4	4.8	4.3	3.9	3.6
Plastic 500 x 1200	8.1	6.8	5.9	5.3	4.8	4.4	4.0
Sucking							
Metal 250 x 960	7.1	5.8	5.1	4.5	4.1	3.7	3.4
Metal 300 x 960	7.1	6.0	5.2	4.7	4.2	3.9	3.5
Plastic 350 x 1200	6.6	5.5	4.9	4.4	4.0	3.6	3.3
Plastic 500 x 1200	7.2	6.1	5.4	4.8	4.4	4.0	3.7

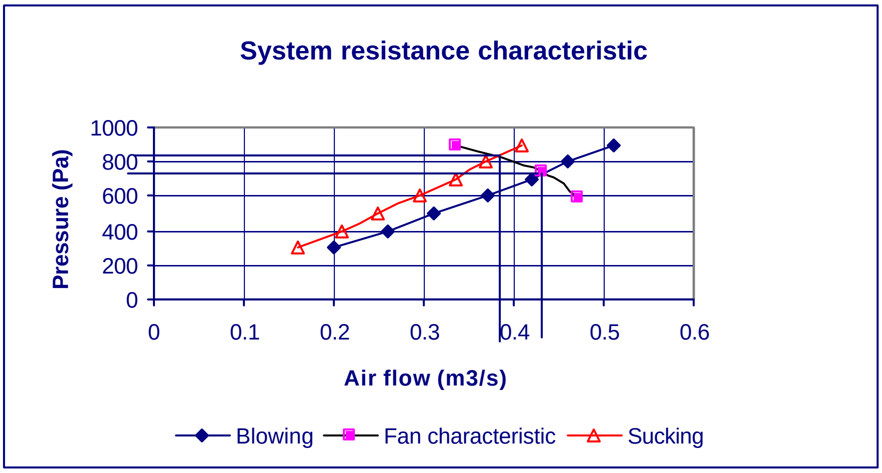
Grain depth has the most effect on duct spacing but the size of the perforated duct section and the amount of free area also has an influence.

If units are spaced further apart the expected cooling time between units will increase, for example a 20% increase in spacing will result in 40% increase in the cooling time Spacings that give rise to cooling times of more than 150 hours are likely to be unreliable because the small pressure gradients in the grain are easily distorted by local variations in grain packing, resulting in some parts of the bulk not being cooled at all.

Selecting a fan

Commercial ventilation systems are supplied complete with a fan that will deliver the correct volume of air at a typical operating pressure for cooling wheat or barley. The volume of air that flows in a cooling unit will depend on the depth and type of crop being ventilated, the diameter and height of the perforated duct and the amount of free area in the perforated duct. The characteristics of the fan against which the duct and crop resistance are balanced will also influence the operating point for an individual duct/fan combination. The air volume, in cubic metres per second, required for a duct can be estimated by multiplying the tonnage ventilated by 0.0027.

Fig 1. Air flow resistance characteristic of a vertical duct.



Tests of commercial installations have shown that it is easier to blow than to suck air through a vertical ventilation system so the direction of air movement will also affect the operating point for a system. Figure 1 shows how the pressure and flow are affected when the direction of air flow is changed. The triangles represent sucking and the diamonds blowing. The fan sees a higher pressure and delivers a lower flow rate when sucking.

The use of an over sized fan on a small duct will result in excessively high working pressures and only a moderate increase in grain tonnage cooled. The use of a fan that is too small will result in a lower working pressure, a smaller flow and extended cooling times at the edge of the grain bulk served.

Number of fans

Cooling recommendations assume one fan will be shared between two or three ducts. Where the grain to be cooled will come from a high temperature drier one fan per two ducts is recommended but where grain comes direct from the field or the store is loaded over several weeks one fan can serve three ducts. If moist grain is being held prior to high temperature drying one fan per duct is recommended.

Hot-spot coolers

Sometimes known as ‘aeration spears’, these are ducts with a threaded tip which enable them to be screwed into the grain and, when a fan is attached, are usually able to cool a core of 16 cu m (12 t) in a day or two. They are designed to cool ‘hot-spots’ rapidly and will often eject large numbers of insects which have been driven towards the duct by the cooling air. When the grain is cool, continued intermittent ventilation is necessary to prevent the insects re-heating the grain.

Using vertical ventilation units for drying

To follow current recommendations, ambient air dryers need to supply up to 20 times the volume of air that is used for cooling. Vertical ventilation systems produce non uniform distribution of air throughout the grain bulk. The grain near the duct receives much more air than that at the edge of the treated zone. For cooling this variation is less important because cooling fronts move through the grain much more quickly than drying fronts. It may be possible to remove up to 2% mc per week if the ducts are spaced close enough, the fans operate continuously and the temperature and humidity of the drying air is favourable. Model results show that even if the average moisture content of the bulk is reduced, some of the grain will remain close to the original moisture content. The provisional duct centres and grain quantities served by each duct to give a maximum of 2% per week moisture removal are shown below. Drying times and final moisture contents can be reduced by small temperature rises of up to 5°C with blowing systems.

Table 4. Suggested duct spacing and tonnage for drying wheat 3 m deep.

Pressure (Pa)	500	600	700	800
grain/unit (tonnes)	36.1	40.0	44.1	50.6
Unit centres (m)	3.8	4.0	4.2	4.5
Air flow (m^3/s)	0.30	0.35	0.41	0.46

Because of the slow and uneven drying that can be expected with vertical ventilation systems they should not be used where the initial grain moisture content is above 18% or there will be a risk of Ochratoxin A formation. Further research is required before definite recommendations can be made on moisture removal by vertical aeration.

Operation of vertical aeration for cooling

It is important that cooling is started as soon as the ducts are covered. Do not delay until a store is full.

Cost-effective cooling depends on operating the fans when ambient is cooler than the maximum grain temperature. This can be achieved manually, based on forecasts of minimum night temperatures or by a simple time switch set to blow during the night or automatically using set point or differential thermostats. The latter switch on the fans automatically when ambient air is cooler than the grain. A differential of 46°C is usually effective. Some temperature monitoring systems can be arranged to provide this control.

100 hours ventilation is required to complete the first cooling to between 15 and 20°C. Depending on system design, one fan can be used with two or three ducts. Fans should blow continuously for 1-2 days before being moved to the next duct but cooling of the grain served by all ducts must still be complete within 2-3 weeks, if insects are not to complete their life-

cycles. When all the grain has been cooled to below 20°C, fans can be moved between ducts at weekly intervals until all the grain has been cooled to below 10°C.

Monitoring

However the fans are operated, it is important to keep records of hours run - either by manual records of when fans are switched on or off or by using an hours meter and recording readings at regular intervals. This will indicate whether or not the aeration system is working properly. It should take no more than 300 hours aeration to achieve minimum temperatures below 10°C.

Temperature records, taken at least weekly, are important for the management of aeration systems. They will indicate the progress of cooling and when the fans can be switched off. In an upwards (blowing) system the warmest grain will be near the surface and between ducts so this is the best place to install a permanent string of temperature sensors or to make regular (weekly) measurements using a hand probe.

Moisture contents will change little during aeration at recommended rates but should be measured at loading when an maximum of 14.5% should be aimed for. Surface mite populations will be inevitable as the surface takes up moisture during the winter and this can be minimised by storing grain at about 13% mc. Above 14.5% mc mites and moulds will slowly increase. Above 18%, there is a threat of Ochratoxin A (OA) formation, even if the grain is cooled.

Even when grain is cool, it pays to operate the fans intermittently, say one night every fortnight, to check for 'off' odours and to prevent the development of hot-spots. These are caused by developing weevil stages hidden inside the grain and are very local and hence difficult to detect. Insect traps can act as an early warning system and are usually laid in pairs in a 5-6m grid at the surface and just below. Weekly examination and records from these will give early warning of any sudden changes in insect activity so that remedial action can be taken.

Part 1.

COMPUTATIONAL FLUID DYNAMICS (CFD) MODELLING

B.HARRAL

SUMMARY

Computational modelling was used to obtain a detailed view of the airflow within the grain bulk under controlled conditions. This approach solves numerically the differential equations that describe heat and mass transfer, and predicts the air velocities, temperatures and pressures throughout the bulk and their variation with time. The dimensions of the store and the shape of the bulk, whether level or heaped, were reproduced in the model. The grain was modelled as a porous medium, with the properties of barley at 16% m.c. (wet basis, w.b). Modelling predicted the airflow paths and enabled the position and shape of the cooling front to be visualised. It has shown that if the aerator fan is set to blow rather than suck then the volume of grain cooled is up to 20% greater and better cooling is obtained near the floor. Modelling has confirmed that the changes in grain moisture content are insignificant when using cooling rates of airflow at ambient temperatures. The optimum spacing of vertical aerators has been calculated for different grain bed depths, particularly those not covered in the farm trials.

METHODOLOGY

Mathematical modelling is a particularly useful technique when a large number of store and ventilation layouts need to be investigated. It is less expensive than experimental trials and is unaffected by changes in weather conditions. Previous work (Bibby and Conyers, 1998) developed an approach using Computational Fluid Dynamics (CFD) techniques for predicting the air velocities, pressures and temperatures inside grain stores. The method solves the differential equations that describe heat and mass transfer within the grain bulk, and produces details of the airflow and temperature distribution throughout the store. Recorded ambient air temperature data can be used as a boundary condition to simulate a realistic storage environment.

Description of the Model

The CFD technique subdivides the grain bulk into cells in which the differential equations that describe heat and mass transfer are solved numerically. The CFD software package CFX 4.3 (AEA Technology, 1999) was used to create the cells and solve the linearised equations. Body-fitted cells were employed so that complex shapes could be reproduced. Figure 1 shows the arrangement of cells in a quarter-model of a grain mass containing a single vertical aerator. The number of cells used depended on the layout being modelled, ranging from 14,500 to 150,000 in a quarter-model and up to 250,000 in a full model. The model did not include the perforations in the vertical duct, or the airflow in the vertical duct and the fan unit. The boundary was defined as the interface between the grain and the vertical duct.

The grain was treated as a porous medium with heat and moisture exchange between the grain and the interstitial air, where the heat transfer coefficient was calculated from a correlation given by Boyce (1965). The floor and retaining walls of the store were assumed to be adiabatic, i.e. no heat transfer.

Moisture transfer has a profound effect on grain cooling because of the large amount of latent heat involved in the evaporation and condensation of water. It is important even when the change in moisture content is a fraction of one percent. To simulate moisture transfer the non-equilibrium model of Sun *et al.* (1995) has been included. However, the inclusion of moisture transfer in a simulation increased the run time from 3h by a factor of nearly ten, so when relative cooling times only were sufficient moisture transfer was omitted.

The model allows time-dependent values to be calculated. In particular, the temperature distribution within the grain bulk enables the volume of cooled grain to be calculated as a function of time. Time steps of up to 5 minutes were used, although steps as short as 1.5 minutes were necessary in some cases to maintain numerical stability.

Physical properties

Modelling the airflow and heat transfer in a grain store requires physical and thermal properties of the solids and gases. The properties used throughout are given in Table 1 for barley at 16% moisture content (w.b) (ASAE D243.3, 1992), although the model can be used with any grain type if the corresponding properties are known. It is assumed that these are uniform and constant throughout the grain within the operating range.

An expression relating the pressure gradient in the grain to the local air velocity was taken from ASAE D272.2 (1992). However, because the natural alignment of grain kernels produces less resistance horizontally, a 70% difference in resistance between horizontal and vertical directions was assumed (Kumar and Muir, 1986).

Grain stores

A series of single aerator simulations has been carried out with grain depths of 2, 3, 4, 5 and 6m, and boundary distances of 2, 3 and 4 times the grain depth. Some simulations have also been run in which additional vertical ducts are present but inactive, which represents a more realistic layout than a single aerator.

RESULTS AND DISCUSSION

Model predictions and discussion

All simulations have assumed a constant ambient air temperature of 12°C, and have started with the grain at a uniform temperature of 25°C. A fixed volume flow rate at the aerator of 0.5 m^3s^{-1} has been used irrespective of the grain depth and mass, and in both sucking and blowing simulations. This was based on results from the farm trials (Part 2) using similar fans, which indicated that the flow rate was almost constant between 585 Pa (flow rate = 0.499 m^3s^{-1} at Site 6) and 762 Pa (flow rate = 0.517 $m^3 s^{-1}$ at Site 4). Strictly speaking, the flow rate should be related to the pressure through the fan characteristic but the data were not available.

At each time step the volume of grain cooled to within 2°C of ambient has been calculated. A proportion of the simulations included the moisture transfer model in order to compare the volumes of grain cooled with and without moisture transfer. In these cases a constant ambient air relative humidity of 70% has been assumed, and an initial grain moisture content of 18% (w.b).

The speed of the air passing through the grain, and therefore the cooling effect, depends on the distance from the aeration unit and the shape of the bulk. Figure 2 shows the airflow paths and the transit times in a 3m deep grain bed with a level top surface – the paths are almost identical whether sucking or blowing. Figure 3 shows the progress of the cooling front through the same bed with the fan set to suck and blow. The cooling front is defined as an imaginary line joining points in the grain 2°C above the ambient air temperature. Figure 3 shows that when the aerator fan is set to suck, the cooling front does not reach the floor and the cooling effect near the floor is relatively small. This is true in many cases even after long periods of ventilation, particularly in bed depths greater then 3m. Conversely, when the fan is set to blow, the grain near the floor is always completely cooled.

Figure 4 shows the predicted volume of cooled grain as a function of time, in 2m and 3m deep beds for various store configurations (the quantity of cooled grain being the volume reduced to 2°C above ambient). Figures 5, 6 and 7 show the results for 4m, 5m and 6m deep beds respectively. The pressure values given in Figures 4-7 are those predicted at the vertical duct. These values will always be lower than experimentally measured pressures because there are insufficient cells in the model to resolve the very steep pressure gradient in the grain near the duct.

There are two trends shown by Figures 4-7. First, if additional ducts are present but inactive they 'short-circuit' the airflow and reduce the quantity of grain cooled. This shows the importance of capping unused ducts. Second, as the boundaries move further away the quantity of grain cooled by a single aerator decreases.

Figure 8 compares the predicted volumes of grain cooled when sucking and blowing. Blowing consistently cools more grain than sucking. For example, when b=9m blowing has cooled 28% more grain than sucking after 4 days, and 20% after 7 days.

Figure 9 shows the increase in predicted quantity of grain cooled when moisture transfer is included in the simulations. Taking b=9m, the model predicts that 140% more grain is cooled after 4 days when moisture transfer is included. At the same time the grain moisture content throughout most of the bulk changes by less than 0.5%. Blowing still cools more grain than sucking but the difference is now 11% after 4 days and 8% after 7 days.

Calculation of Aerator spacing

The predicted volume of grain cooled after 100 hours with the aerator fan set to suck, has been used to calculate aerator spacing. The volumes predicted by simulations in which moisture transfer was omitted, have been multiplied by 2.4 to account for the increase in cooled volume due to evaporation. Table 2 shows the volume at each bed depth. The diameter of the cylinder equivalent to this volume is then calculated, and the optimum aerator spacing is assumed to be the side of the square whose diagonal is equal to the cylinder diameter.

CONCLUSIONS

Modelling has:-

1. enabled the airflow in many different store and aerator configurations to be studied relatively quickly and under controlled conditions.
2. predicted the airflow paths and enabled the position and shape of the cooling front to be visualised. been able to predict the extent to which moisture transfer increases the volume of cooled grain and has also shown that moisture changes in the grain itself are small.
3. enabled the volume of cooled grain to be predicted as a function of time in a variety of layouts.
4. enabled the optimum spacing of vertical aerators to be calculated for different grain bed depths, particularly bed depths not covered in the farm trials.

Table 1. Material properties used in the CFD model.

Barley bulk density (dry)	610.6 $kg.m^{-3}$
Barley specific heat (dry)	1298.0 $J.kg^{-1}.°C^{-1}$
Barley bulk conductivity	0.1317 $J.m^{-1}.°C^{-1}$
Barley volume porosity	0.45
Water vapour in air diffusion coefficient	$2.6x10^{-5}$ $m^2.s^{-1}$

Table 2. Calculated optimum aerator spacing.

Grain bed depth, m	Cooled volume at 100 hrs, m^3	Aerator spacing, m
2	181.1	7.6
3	239.1	7.1
4	223.4	6.0
5	185.9	4.9
6	152.5	4.0

Fig. 1. Example of the cell array used in a quarter-model of a grain store.

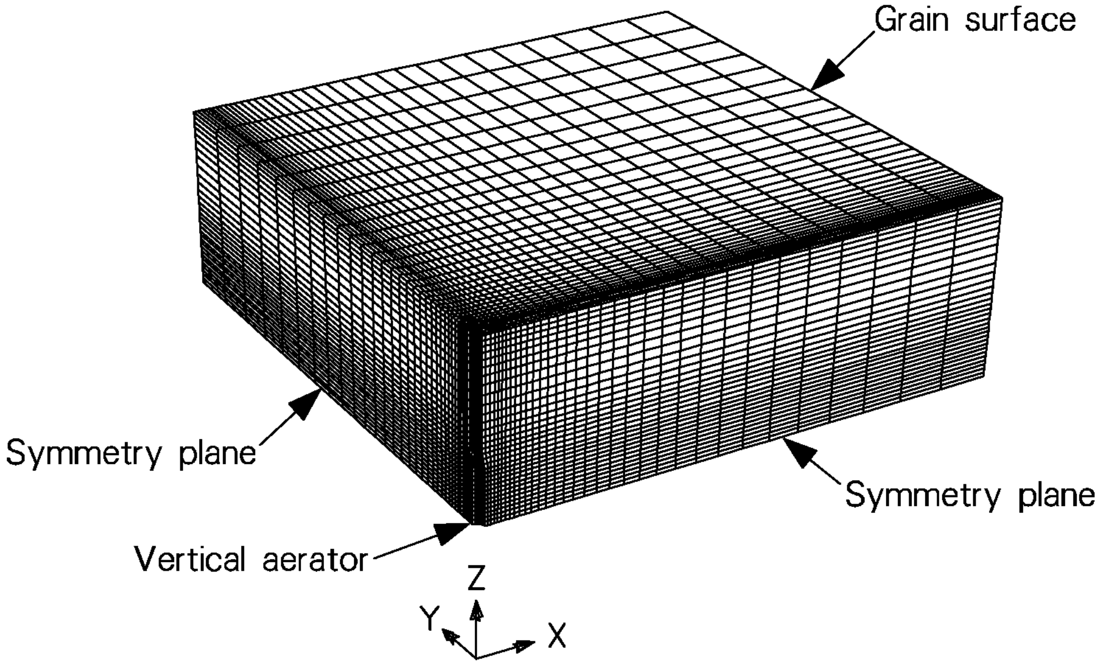


Fig. 2. Predicted airflow paths and transit times (s) on the vertical cross-section through a grain bed 3m deep. The single central aerator is sucking.

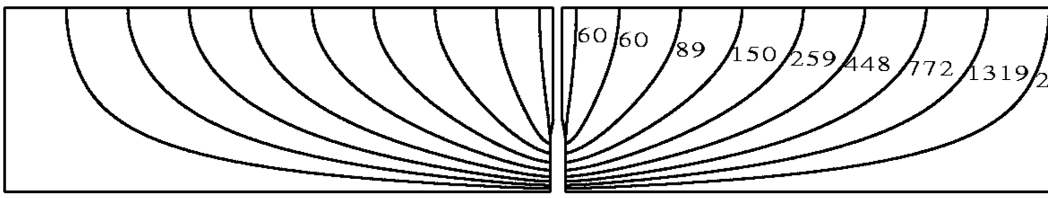


Fig. 3. Predicted position of the cooling front at 12 hour intervals on the vertical cross-section through a grain bed 3m deep when sucking (top) and blowing (bottom) using a single central aerator. Front defined as 2°C above ambient. Moisture transfer is included in the simulations.

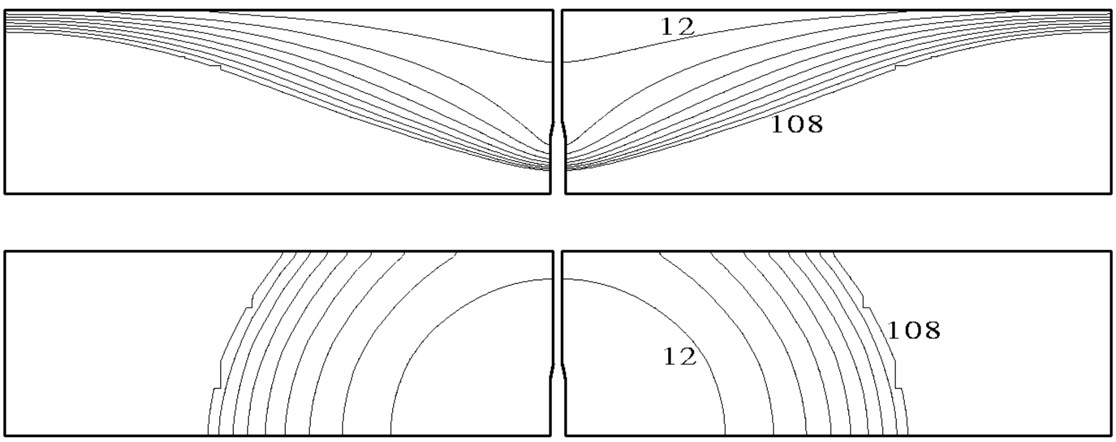


Fig. 4. Predicted volume of grain cooled to within 2°C of ambient by a single vertical aerator. The ambient air temperature is 12°C and the initial grain temperature is 25°C. Moisture transfer is not included in the simulations. s=aerator separation when other aerators are present but inactive(¥=no other aerators present), b=distance to boundary, P=predicted aerator pressure.

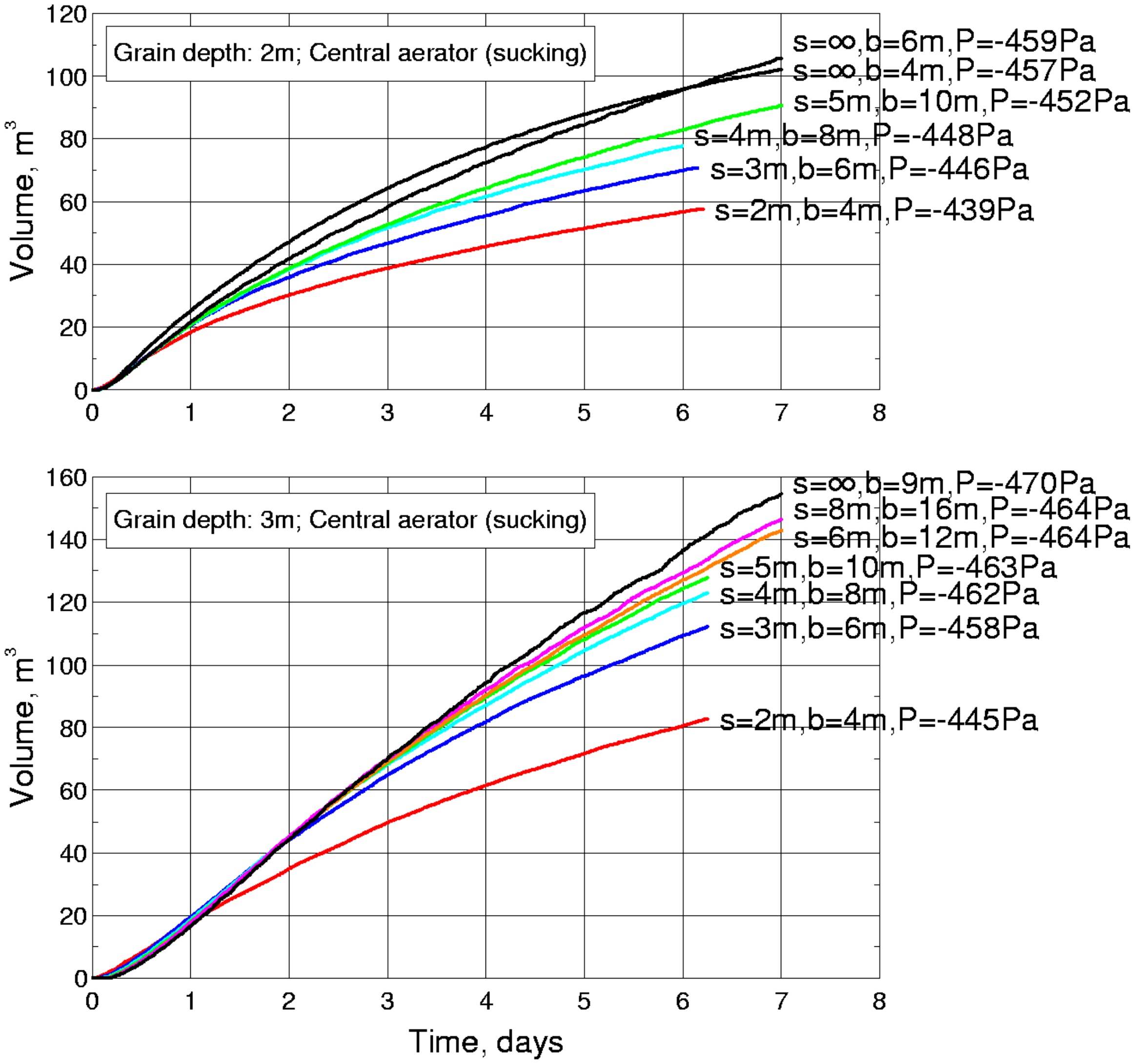


Fig. 5. Predicted volume of grain cooled to within 2°C of ambient by a single vertical aerator in a bed 4m deep with the aerator sucking. The ambient air temperature is 12°C and the initial grain temperature is 25°C. Moisture transfer is not included in the simulations. s=¥=no other aerators present, b=distance to boundary, P=predicted aerator pressure.

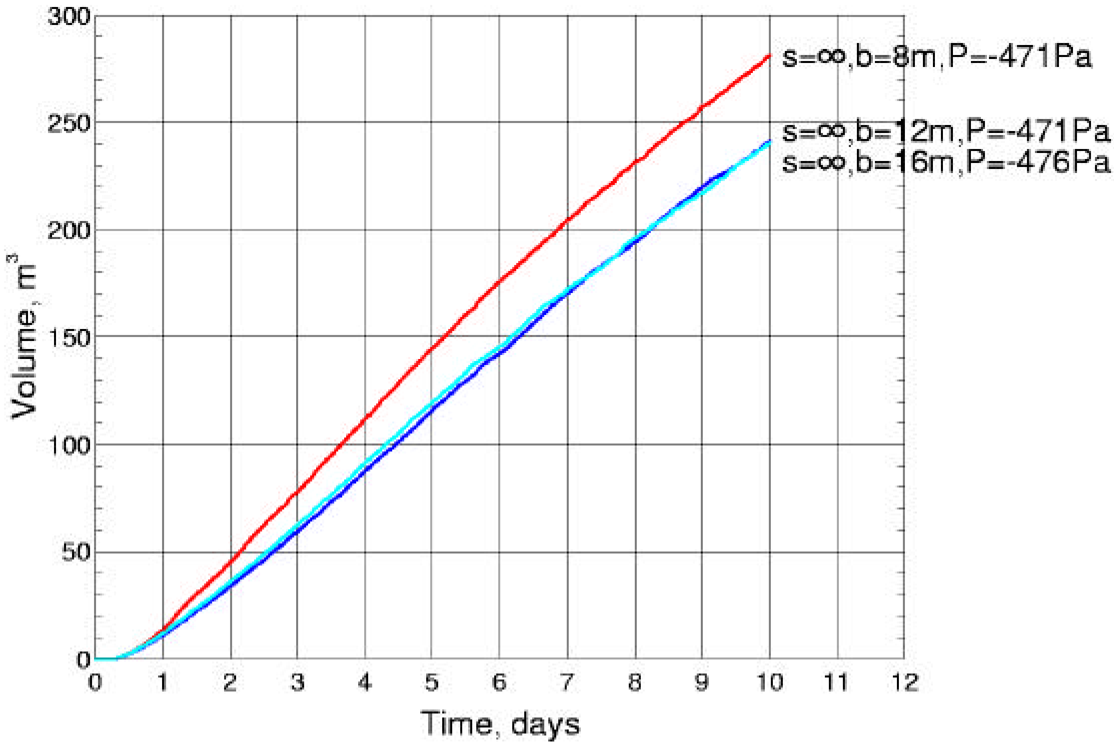


Fig. 6. Predicted volume of grain cooled to within 2°C of ambient by a single vertical aerator in a bed 5m deep with the aerator sucking. The ambient air temperature is 12°C and the initial grain temperature is 25°C. Moisture transfer is not included in the simulations. s=¥=no other aerators present, b=distance to boundary, P=predicted aerator pressure.

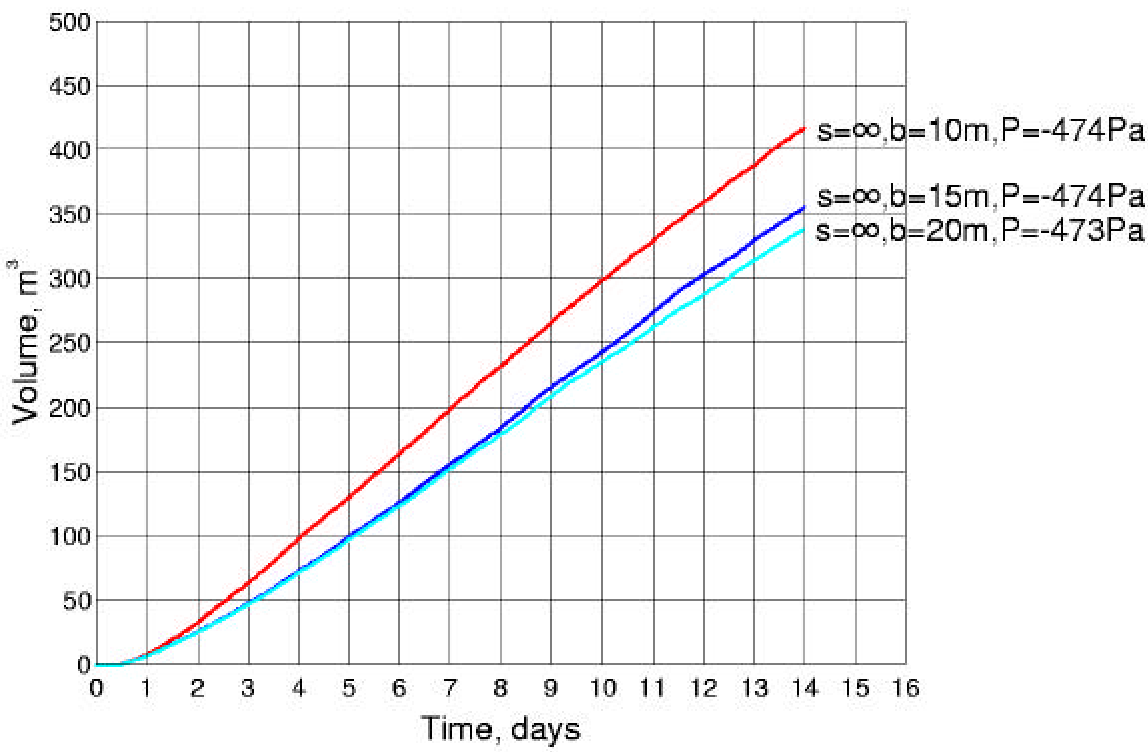


Fig. 7. Predicted volume of grain cooled to within 2°C of ambient by a single vertical aerator in a bed 6m deep with the aerator sucking. The ambient air temperature is 12°C and the initial grain temperature is 25°C. Moisture transfer is not included in the simulations. s=¥=no other aerators present, b=distance to boundary, P=predicted aerator pressure

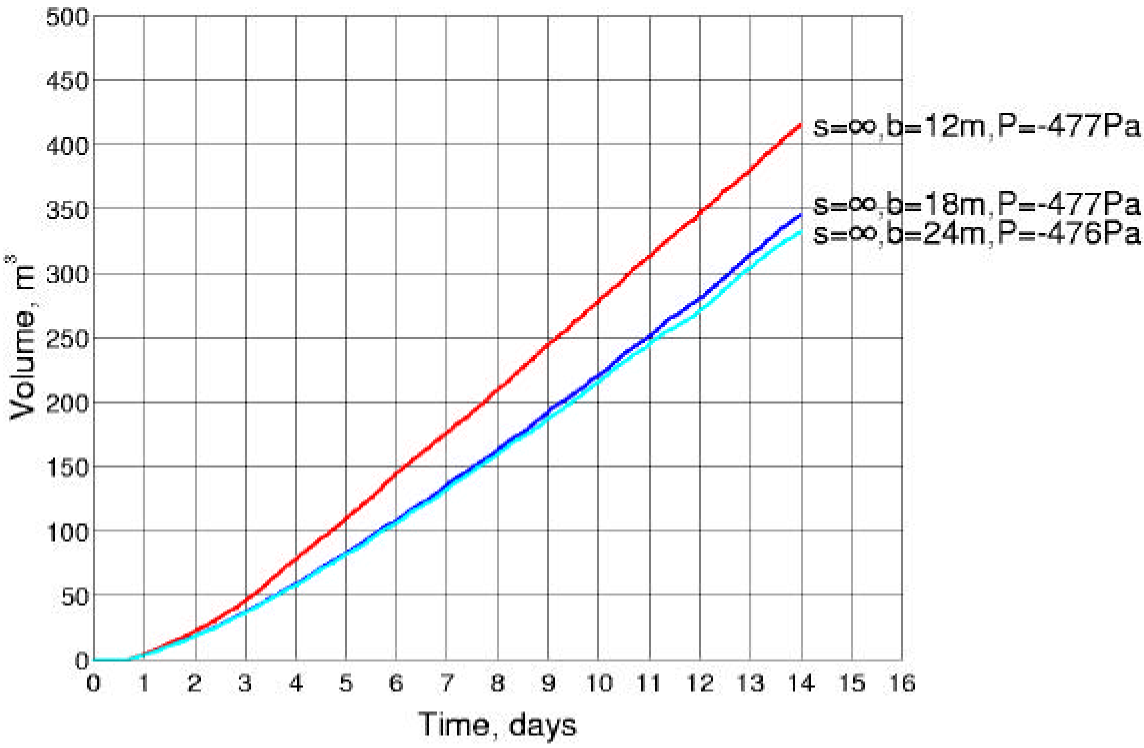


Fig. 8. Comparison between predicted volumes of grain cooled to within 2°C of ambient by a single vertical aerator in a bed 3m deep with the aerator sucking and blowing. The ambient air temperature is 12°C and the initial grain temperature is 25°C. Moisture transfer is not included in the simulations. No other aerators are present. b=distance to boundary, P=predicted aerator pressure.

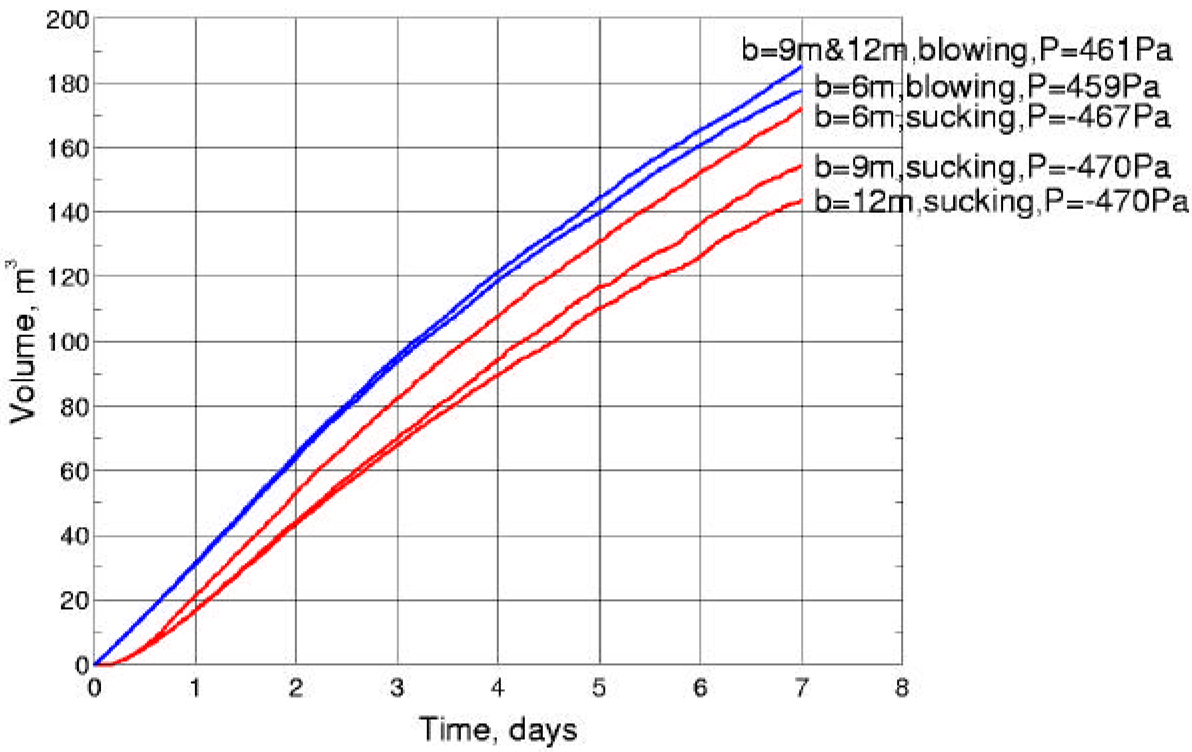


Fig. 9. Comparison between predicted volumes of grain cooled to within 2°C of ambient by a single vertical aerator in a bed 3m deep with the aerator sucking and blowing. The ambient air temperature is 12°C and the initial grain temperature is 25°C. Moisture transfer is included in the simulations. No other aerators are present. b=distance to boundary, P=predicted aerator pressure.

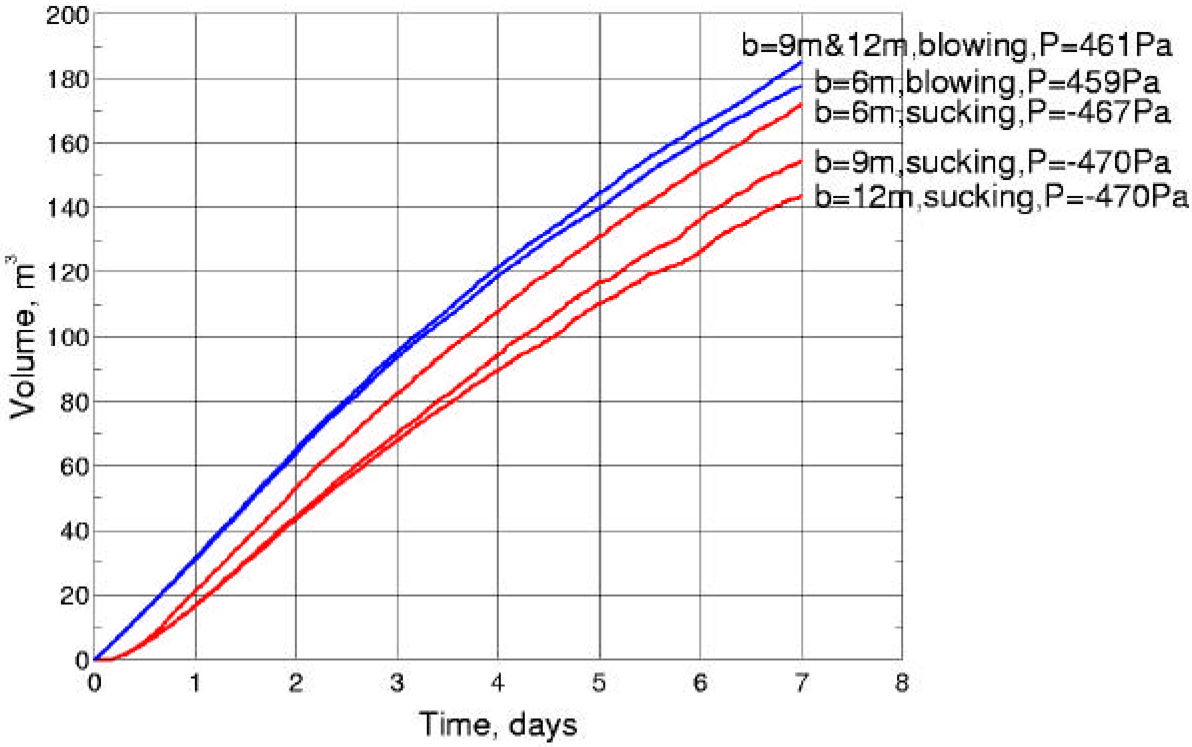
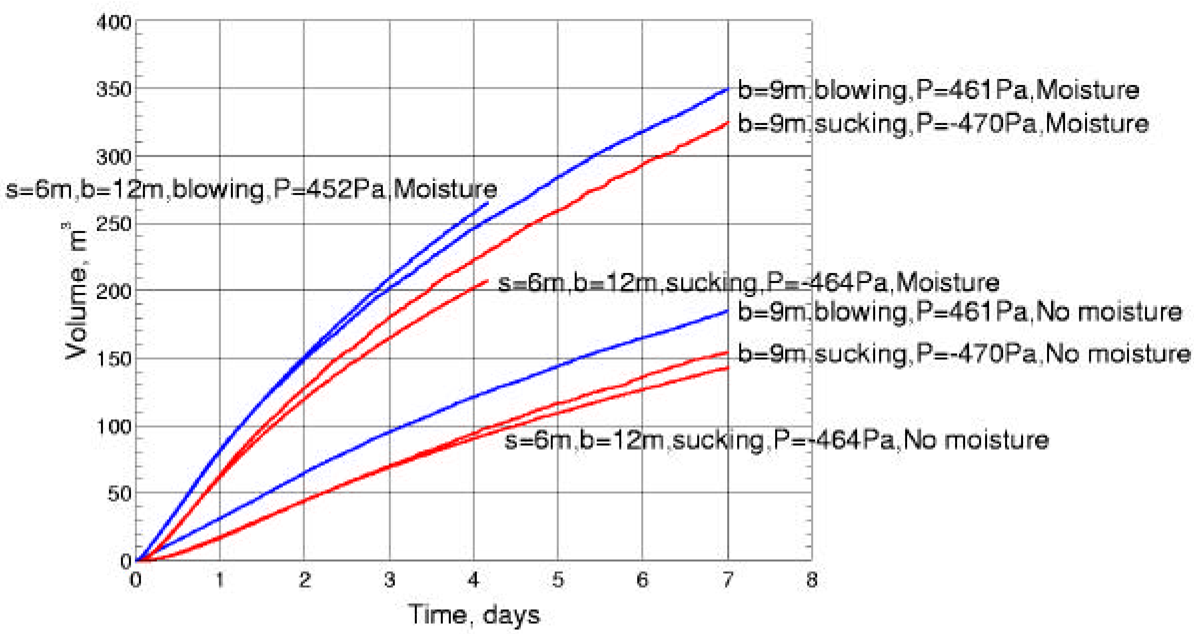


Fig. 10. Comparison between predicted volumes of grain cooled to within 2°C of ambient by a single vertical aerator in a bed 3m deep with the aerator sucking and blowing, and with and without moisture transfer included in the simulation. The ambient air temperature is 12°C, the initial grain temperature is 25°C and the initial grain moisture content is 18% (w.b). s=aerator separation when other aerators are present but inactive, b=distance to boundary, P=predicted aerator pressure.



Part 2.

FARM STUDIES OF VERTICAL AERATION SYSTEM PERFORMANCE

D. BARTLETT AND D. M. ARMITAGE

SUMMARY

Temperature, grain moisture content change and ventilating air pressure were measured over a period of about a month in six commercial grain stores equipped with vertical ventilation systems. At each site the volume of air moved by the fan was up to 20% more when it was blowing than when it was sucking. Correspondingly higher pressures were observed in the grain during blowing. Neither air or grain conditions favoured drying at any of the sites and during the monitoring period only small changes in grain moisture content were observed. During ventilation, clear diurnal temperature fluctuations were seen at most of the monitoring points. These observations were used to estimate the speed of the cooling zone. This cooling zone velocity was linked to the pressure gradient in the grain bulk and this relationship can be used to estimate cooling times where the pressure gradient is known. The data from these observations was used to validate a simplified working model of vertical aeration performance based on the studies outlined in Part 1.

METHODS

Measurements were made on 6 sites. The aims were to measure the velocity of the cooling front as it moved through the grain, to measure the pressure gradient within the grain bulk and to record any changes in moisture content that took place in the grain profile. Observations were concentrated on a single ventilation duct near the centre of a large grain bulk.

Temperature monitoring

Temperatures were measured by thermocouples attached to data loggers. The thermocouples were installed at 3 equidistant depths, depending on the grain depth and in five equidistant columns in a radius between adjacent ducts. Where only one duct was used, the increments of radius were 800mm.

A single temperature probe was suspended in the air space above the grain in suction systems. This probe was placed near the fan inlet in blown systems.

Temperature were logged at 1.5 hour intervals for ca. 4 weeks. These records were used to estimate the cooling front velocity and to provide validation data for the computational fluid dynamics (CFD) model.

Grain moisture content monitoring

Moisture content samples were taken by gravity spear or vacuum sampler from close to each of the temperature probes before ventilation was started and from the same positions at the end of the observations. The moisture content of each sample was determined by oven drying using ISO 712 at 130°C for 2h.

Pressure monitoring

A detailed survey of the pressure profiles in the grain bulk was made at the end of the monitoring period using a micromanometer . With only one ventilation duct operating, the pressures in the grain bulk were measured at 0.5m depth increments in the same columns used for the temperature probes. The pressure inside the ventilation duct was also recorded while the fan was running. All surrounding ventilation ducts were sealed during this test.

Additional sets of vertical pressure measurements were made at larger radii at some sites to establish the radius of influence of the single duct.

Pressure gradients in the grain indicate magnitude and direction of air flow. These observations can be used together with the temperature records to establish the relationship between air flow and the movement of a temperature change front. The pressure observations were also used to validate the CFD model.

Air volume measurement

The volume of air moving through the system, whether blowing or sucking, was measured at the intake or discharge of the fan using a vane anemometer to measure the velocity and multiplying this by the duct cross sectional area.

Cooling performance depends on the volume of air moved through the grain. The pressure and flow generated by the fan depends on the ventilation characteristics of the grain and duct under working conditions.

System performance

A calibrated orifice plate (A flow measuring device that uses the pressure difference created when air flows through a restriction to establish the flow rate) was used with a variable speed fan to measure the flow/resistance characteristics of each system. Airflow rates were established by measuring the pressure drop across the orifice plate. The static pressure at the bottom of the duct was recorded for each of the flow rates.

The relationship between flow and static pressure is the characteristic needed to select a fan to deliver a specified ventilation rate through the grain.

Determination of cooling front velocity

Temperature records taken from cooling grain show a characteristic diurnal oscillation. The ambient temperature record shows the greatest amplitude and when the fan is working the same pattern can be identified at most of the monitoring points within the grain. The amplitude of these temperature waves decreases with distance from the air entry point. There is also a phase shift so the peak temperature at, say, a metre from the entry occurs later than the ambient temperature peak. The time difference between these peaks and the distance between measurement points were used to estimate the velocity of the cooling zone at each point in the bulk.

Site details

The ducts used on sites 1-3 were metal, the perforated sections were of diameter 0.25m and the unperforated 'neck' sections were 0.15m in diameter.

<u>Site 1</u>

The section of store studied, measured 21 x 42m and contained 2000t of wheat peaked to a depth 5m from 2m at the outer walls. There were three rows of vertical ducts; 6 equidistant central ducts with 2m of perforated area and the two outer rows of 4 ducts with 1m perforations.

Thermocouples were inserted at depths of 1.5m, 3m and 4.5m and the distance between the ducts between which they were placed was 7.2m. At the end of the test, the grain had been partially removed from the store which precluded taking final mc samples and necessitated making the airflow measurements between a different pair of ducts in the central row than were used for the temperature measurements.

<u>Site 2</u>

About 300t of wheat were stored in an uneven heap in a store measuring 13.7m x 13.4m. The grain sloped from 1.5m at the walls to 2.15m at the peak. The bulk was cooled by three vertical ducts with 1m perforations which were about 3.7m from the edge of the store.

Thermocouples were placed at depths of 0.6m, 1.1m and 1.6m and the distance between the ducts between which they were placed was 6m.

<u>Site 3</u>

The grain was stored in a 6m peak in a store measuring 23m x 45m. The bulk was cooled by two rows of three vertical duct with 1m perforated section which were positioned about 3m from the walls of the store.

Thermocouples were placed at depths of 1.4m, 2.8m and 4.2m and the distance between the ducts between which they were placed was 6m. Unfortunately, logger failure meant that the only temperature records from this site were initial and final spot readings.

<u>Site 4</u>

A 250 tonne bulk of wheat was retained in a rectangular block 8.4m x 18m. The section used for the trial was 3.0m deep. Six ducts were distributed in two rows of three spaced at 3 to 5 metre centres. Each duct had a 0.92m high x 250mm dia perforated section. Fans were moved between ducts at regular intervals. During the test period the duct was ventilated (sucked) continuously for a period of 6 days.

Six vertical strings of five thermocouples were placed at 600mm intervals on a radius from the duct, the first adjacent to the duct and the last 3m from it. Each string measured the temperature at 0.6, 1.2, 1.7, 2.3 and 2.8m from the floor.

Site 5

A single plastic ventilation duct 1.2m high x 300mm dia was installed in an otherwise unventilated bulk of grain 3m deep. A centrifugal fan was connected to continuously blow cooling air into the duct. Unfortunately, this fan was running backwards during the test so the volume of air delivered was severely restricted.

Five sets of temperature measurements were made at 1 m intervals on a radial line from the duct. At each point the temperature was measured at 0.5, 1.5 and 2.5m from the floor. Because of the low ventilation rate, data from this site was difficult to use effectively.

Site 6

A level bulk of grain 20 x 7m and 3m high was ventilated by 10 ducts placed at 3.3m centres in two rows. The perforated section of the ventilation ducts were 250mm dia x 0.92m high and during the test period air was sucked from the duct by a 1.1kw centrifugal fan.

Five sets of temperature measurements were made at 1 m intervals on a radial line from the duct towards a retaining wall. At each point the temperature was measured at 0.5, 1.5 and 2.5m from the floor

RESULTS AND DISCUSSION

Temperatures and hours of aeration

Temperature changes on the 5 sites where successful measurements were made are shown in Figs 1-5. At site 3, where the data loggers failed, initial temperatures were 12-18°C and at the end of the test they had fallen to 8.4-12.4°C. Sites 1-3 were ventilated for 492, 841 and 384 hours. The other sites received intermittent ventilation but no specific record of blowing time was made.

The temperature records for all the sites show that the grain temperature tracked ambient temperature when the fans were operating and remained static when the fans stopped. A large part of the grain volume served by each duct showed diurnal temperature fluctuations (eg see Figs 6 and 7). These changes were most marked near the air entry point (top surface Fig 1) and became less significant further into the bulk (Fig 7). The selective operation of the fans, choosing only cold air would dramatically improve the cooling efficiency of these systems.

Moisture contents

The change in moisture contents measured at the 6 sites are shown in Tables 1 and 2 Generally, the grain was dry and no significant moisture changes were noted but at site 2, the grain was initially 19% mc and local reductions of more than 1% were noted in the middle of the bulk.

Pressure monitoring

Pressure profiles through the bulks showed consistently greater pressures when the ducts were blown than when they were sucked (Tables 3 to 8). At some sites, observations near the duct were inconsistent because the ventilation duct was not always truly vertical. In site 5, the fan was found to be running backwards and this resulted in much lower pressures and flows than usual. At sites 4 and 6 the pressure profile was extended beyond the planned radius to establish how far the influence of the fan could be detected.

System performance

The operating point (when the pressure and flow produced by the fan match the resistance and flow through the duct and grain) of each fan was established by measuring the volume flow rate and static pressure at the bottom of the duct. These results are given in Table 9

At all sites where valid measurements of flow in blowing and sucking were made, the average increase in air volume moved in blowing was 25%. Some of this difference may be attributed to differences in fan performance as a result of different flow conditions at the inlet. The differences are supported by a general increase in pressure gradients observed in the grain during blowing.

System resistance to airflow.

At each site the ventilation duct was connected to a variable speed fan and orifice plate assembly so that the pressure generated in the duct for a number of different air flows could be measured. The results of these tests are plotted in Figs 8-9 together with the working point flow and pressure measurements obtained with the installed fan.

Cooling front velocity

The observations of pressure within the bulk were used to estimate the pressure gradient between each of the temperature measuring points. Table 9 shows how the speed of the cooling front was distributed through a section of the bulk at site 4 where the cooling air was sucked through the grain

Fig. 10 shows a correlation between these parameters using data from all the farm sites with useable results. This relationship was later used to estimate the cooling time along the air flow paths in the grain bulk and to investigate the effect of ventilator spacing and grain depth (see Part 4).

Table 1. Summary of moisture changes during the test period.

Site	Mean MC (%) Initial	Final	Start date	Duration (days)
2	19.0	18.5	25/9	31
3	13.8	13.9	29/11	50
4	15.5	15.4	20/10	50
5	14.9	14.8	28/8	39
6	15.6	15.5	6/10	43

Table 2. Changes in moisture content (% wet basis) with depth during the test period (Row 1 nearest fan).

Site	Row Depth	1	2	3	4	5
2	top	-0.2	0.6	0.1	-0.2	-0.2
	middle	0.8	1.2	1.1	1.3	0.8
	bottom	0.7	-0.1	0.5	0.2	0.8
3	top	-0.2	0.3	0.5	0.3	0.5
	middle	-0.6	-0.8	-0.6	-0.8	0.1
	bottom	0	0	0	0.2	0.0
4	top	-0.2	-0.2	-0.4	-0.7	-0.8
	middle	0.5	0.6	0.6	0.5	0.9
	bottom	0.5	0.3	0.4	0.2	0.0
5	top	0	0.4	0.2	0.4	-0.2
	middle	0	0.4	0.3	0.2	0.0
	bottom	0.3	0	-0.3	-0.2	-0.4
6	top	0.1	0.1	0.3	0.2	-0.4
	middle	0.8	0.7	0.7	0.5	0.4
	bottom	-0.5	-0.5	-0.4	-0.1	0.1

Table 3. Pressure measurements (Pa) in 5 columns at Site 1.

	Radial distance from duct (m)					
Depth (m)	next to duct	1.2	2.4	3.6	4.8	6.0
Sucking.						
0.5	-9.6	-5.8	-4.1	-2.4	-1.3	-0.4
1.0	-14.7	-11.2	-7.2	-4.5	-2.8	-1.7
1.5	-23.6	-19.5	-11.8	-7.1	-4.4	-2.0
2.0	-42.2	-32.0	-16.7	-8.8	-5.3	-2.6
2.5	-91.5	-53.0	-21.5	-11.7	-6.3	-3.0
3.0	-173.0	-73.1	-25.1	-13.8	-7.2	-3.1
bottom	-171.0	-92.0	-29.0	-15.7	-8.1	-3.4
Blowing.						
0.5	9.1	7.5	4.4	2.8	1.6	0.8
1.0	16.4	14.9	9.2	5.7	3.7	1.8
1.5	27.5	24.5	14.6	8.4	5.4	2.7
2.0	46.5	41.1	21.1	11.2	6.8	3.6
2.5	103.0	70.8	27.4	14.6	8.8	4.1
3.0	195.0	105.7	32.4	16.6	10.0	4.9
bottom	230.0	142.0	38.4	18.4	10.8	5.0

Table 4. Pressure measurements (Pa) in 5 columns at Site 2 *.

	Radial distance from duct (m)					
Depth (m)	next to duct	1.0	2.0	3.0	4.0	5.0
Sucking (Fan B)						
0.5	-21.8	-10.2	-5.2	-2.1	-1.2	-0.2
1.0	-60.0	-21.5	-10.8	-4.3	-2.1	-1.0
1.5	-205.0	-35.0	-16.2	-6.2	-3.0	-1.3
2.0	-297.0	-42.5	-20.0	-6.9	-3.3	-1.3
bottom	-304.0	-44.2	-20.6	-6.9	-3.2	-1.3
Sucking. (Fan A)						
0.5	-24.0	-15.3	-6.5	-3.0	-1.0	-0.2
1.0	-65.1	-35.6	-13.8	-6.4	-2.8	-1.0
1.5	-269.0	-56.9	-20.3	-9.8	-3.7	-1.2
2.0	-346.0	-71.4	-25.6	-11.9	-4.5	-1.8
bottom	-353.0	-75.6	-27.0	-12.4	-4.7	-1.8
Blowing. (Fan A)						
0.5	28.0	17.0	7.0	3.9	1.8	1.0
1	72.0	36.0	15.5	8.3	2.8	1.4
1.5	320.0	57.8	22.0	11.1	4.0	1.8
2	465.0	70.0	27.2	14.0	4.8	2.0
bottom	446.0	73.0	28.5	14.7	4.8	2.3

- Two different fans were used at site 2. The pressure distribution was recorded with fan B sucking and with fan A, both sucking and blowing.

Table 5. Pressure measurements (Pa) in 5 columns at Site 3.

		Radial distance from duct (m)				
Depth (m)	next to duct	1.0	2.0	3.0	4.0	5.0
Sucking.						
0.5	-4.4	-4.1	-2.9	-2.4	-2.0	-1.2
1.0	-7.8	-7.6	-5.7	-5.1	-3.6	-2.6
1.5	-12.3	-12.3	-9.7	-7.7	-5.8	-3.6
2.0	-18.8	-18.1	-13.3	-11.0	-7.7	-5.3
2.5	-27.0	-25.9	-18.1	-15.1	-10.1	-6.5
3.0	-37.8	-38.1	-23.7	-18.9	-12.3	-7.6
3.5	-58.7	-60.6	-29.3	-22.6	-14.0	-8.7
Blowing						
0.5	4.3	4.1	3.5	3.3	2.5	2.0
1.0	8.6	8.8	7.3	6.2	4.4	3.3
1.5	14.1	13.4	10.9	8.9	6.7	4.5
2.0	20.7	18.9	15.7	12.6	8.4	6.1
2.5	28.4	28.3	21.8	16.4	11.5	7.7
3.0	44.1	41.4	30.4	20.8	13.6	8.5
3.5	59.9	60.8	39.4	24.5	15.6	9.7

Table 6. Pressure measurements (Pa) at Site 4.

					Radial distance from duct (m)					
Depth (m)	next to duct	0.6	1.2	1.8	2.4	3.0	3.5	4.0	4.5	5.0
Sucking										
0.5	-11.5	-11.5	-8.3	-6.9	-5.0	-3.8	-2.7	-1.8	-1.0	-0.5
1.0	-25.7	-26.8	-20.6	-14.7	-10.9	-7.6	-5.6	-4.2	-2.7	-1.7
1.5	-48.9	-49.3	-36.6	-25.8	-18.4	-12.0	-8.5	-6.6	-4.8	-3.2
2.0	-101.9	-101.4	-58.8	-36.9	-24.3	-16.0	-11.2	-8.5	-6.0	-3.7
2.5	-188.9	-216.0	-83.4	-47.4	-29.4	-19.1	-12.1	-9.9	-7.0	-4.6
3.0	-198.9	-236.0	-101.1	-53.2	-32.3	-20.5	-13.9	-10.8	-6.9	-5.0
Blowing										
0.5	17.7	13.8	11.9	9.6	7.6	5.7	4.3	3.2	2.5	2.0
1.0	36.3	30.9	25.5	18.1	14.5	10.3	8.0	5.7	4.5	3.4
1.5	69.2	57.2	44.8	30.3	23.3	15.8	11.6	8.1	6.4	4.8
2.0	166.0	113.9	74.0	41.8	31.5	20.7	14.9	10.4	7.7	5.8
2.5	336.0	218.0	115.4	52.7	38.5	24.6	17.9	12.2	9.0	6.3
3.0	323.0	260.0	143.5	57.8	42.7	27.0	19.7	13.3	9.6	6.4

Table 7. Pressure measurements (Pa) at site 5.

	Radial distance from duct (m)					
Depth (m)	next to duct	2	3	4	5	6
Sucking.						
0.5	2.6	1.8	1	0.9	0.8	0.9
1.0	4.8	3.4	2.1	1.3	0.8	0.8
1.5	8.8	5.3	2.9	1.4	1.1	0.6
2.0	15.6	6.9	3.3	1.8	1.1	0.8
2.5	20.3	8.2	3.9	1.9	1.2	0.8
2.9	21.5	8.1	4	1.8	1.3	
Blowing						
0.5	2.6	1.8	1	0.9	0.8	0.9
1.0	4.8	3.4	2.1	1.3	0.8	0.8
1.5	8.8	5.3	2.9	1.4	1.1	0.6
2.0	15.6	6.9	3.3	1.8	1.1	0.8
2.5	20.3	8.2	3.9	1.9	1.2	0.8
2.9	21.5	8.1	4	1.8	1.3	

Table 8. Pressure measurements (Pa) at site 6*.

	Radial distance from duct (m)										
Radius (m)	duct	0.4	0.8	1.3	1.7	2.1	3.0	4.0	5.0	6.0	7.0
Depth (m)											
Sucking											
0.5	-6.9	-5.9	-5.3	-4.5	-3.1	-3.4	-3.4	-2.1	-1.1	-1.0	0.0
1.0	-14.7	-13.4	-11.2	-9.6	-8.1	-7.6	-6.8	-4.1	-2.5	-2.0	-0.6
1.5	-28.4	-24.7	-20.0	-15.7	-12.9	-11.8	-10.7	-6.6	-3.5	-2.9	-1.1
2.0	-53.0	-42.3	-29.1	-21.5	-17.6	-15.1	-14.6	-8.7	-4.7	-3.4	-1.7
2.5	-153.0	-70.6	-37.8	-27.4	-20.5	-18.0	-17.5	-10.4	-5.7	-3.8	-2.0
3.0	-181.5	-86.0	-42.3	-29.7	-21.0	-18.7	-19.2	-11.4	-6.7	-4.2	-2.3

*No blowing measurements were made at this site

Table 9. Operating point for ventilation systems.

Site	Fan size	Flow direction	Volume (m^3/s)	Static Pressure (Pa)
1	152.5mm dia	Suck	0.67	370.8
	152.5mm dia	Blow	0.82	505.2
2	B 127mm square	Suck	0.65	636.0
	A 152.5mm dia	Suck	0.55	600.0
	A 152.5mm dia	Blow	0.70	590.0
3	152.5mm dia	Suck	0.37	868.2
	152.5mm dia	Blow	0.54	540.0
4	138mm dia	Suck	0.52	762.0
	138mm dia	Blow	0.55	715.0
5	138mm dia *	Suck	0.12	16.8
	138mm dia *	Blow	0.08	23.2
6	138mm dia	Suck	0.50	585.7

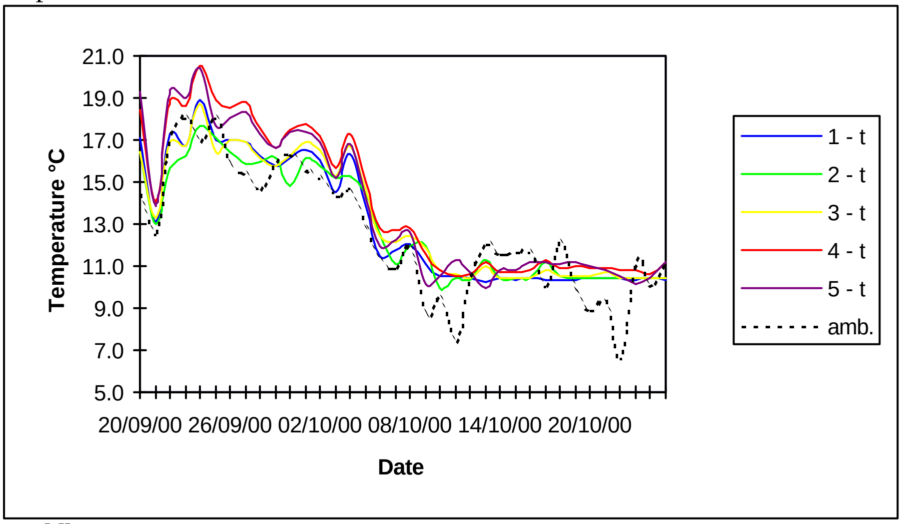
*Fan running backwards

Table 10. Cooling front velocity (mm/h) measured from the surface.

Distance (m)	Radial distance from the duct					
from surface	0.6	1.2	1.8	2.4	3.0	3.6
0.2	166	117	133	111	66	105
0.7	162	122	116	71	66	64
1.3	132	119	104	80	61	54
1.9	154	130	108	70	58	
2.5	124	136	95			

Fig 1. Temperatures at 3 depths and 5 columns in store 1.

Top



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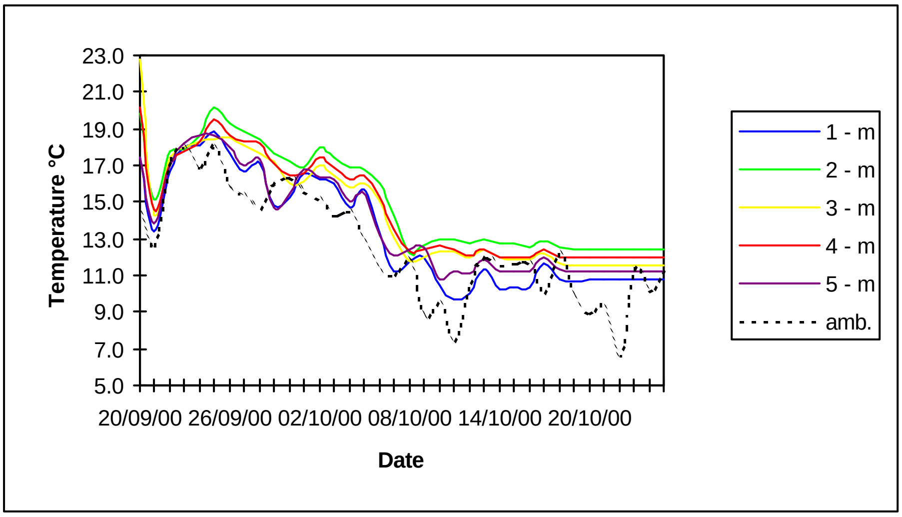

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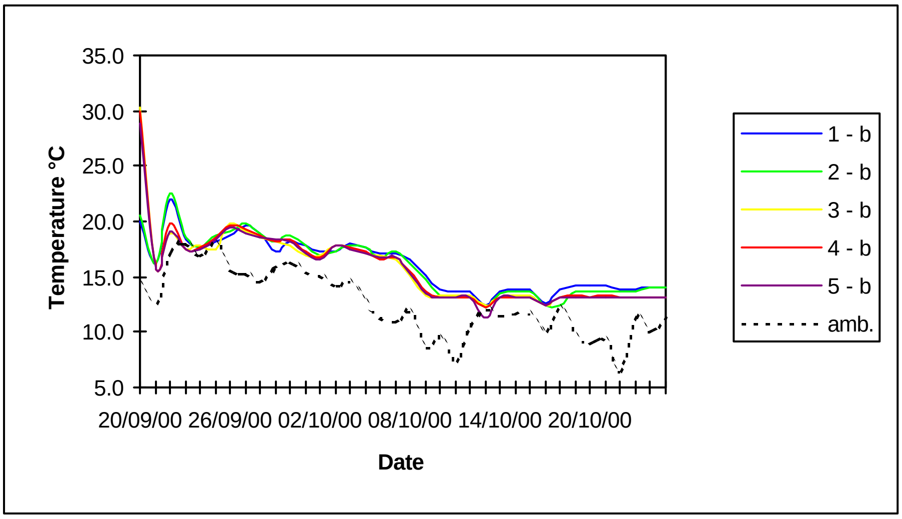

Fig 2. Temperatures at 3 depths and 5 columns in store 2.

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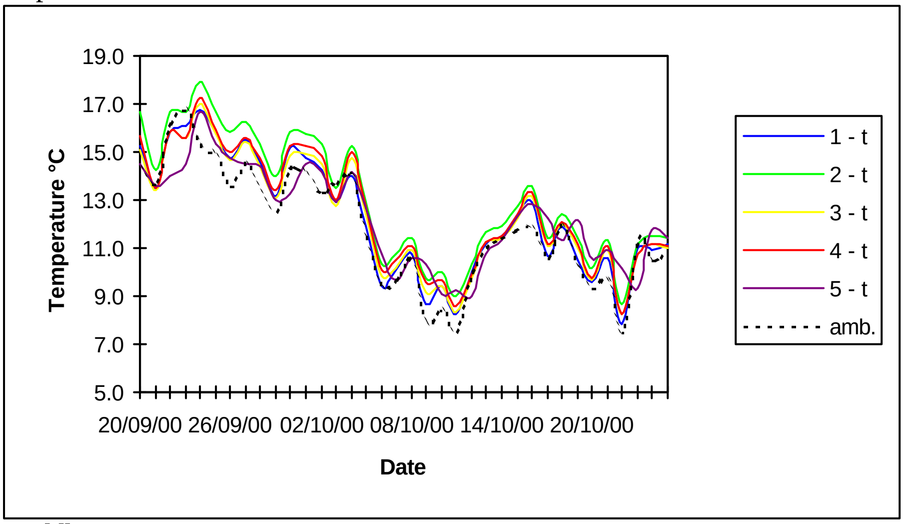

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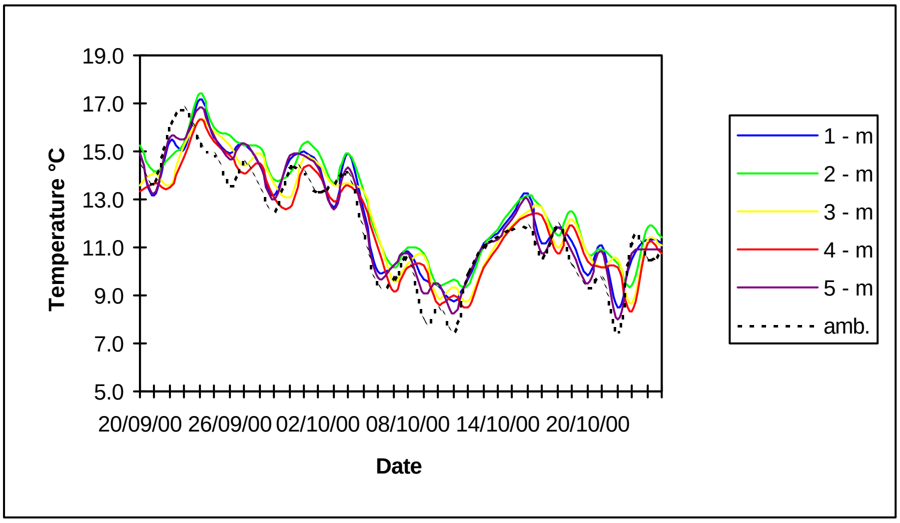

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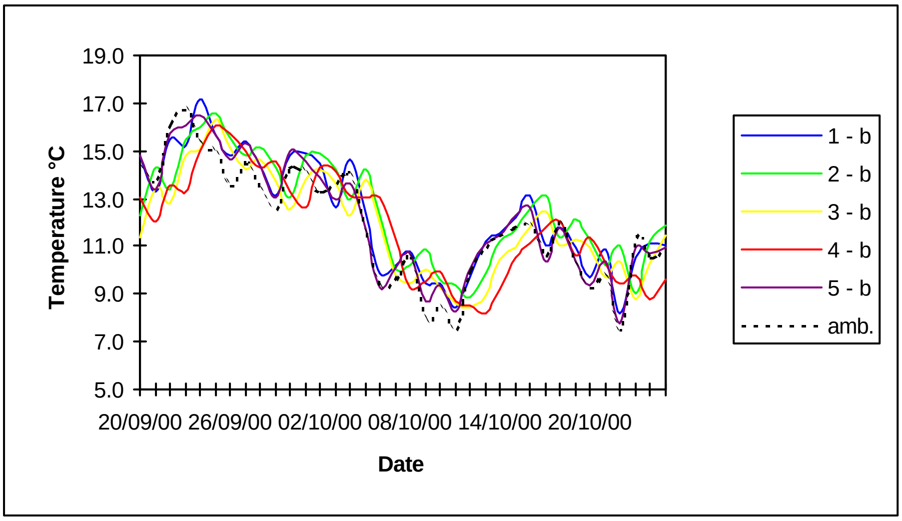

Fig 3. Temperatures at 3 depths and 5 columns in store 4.

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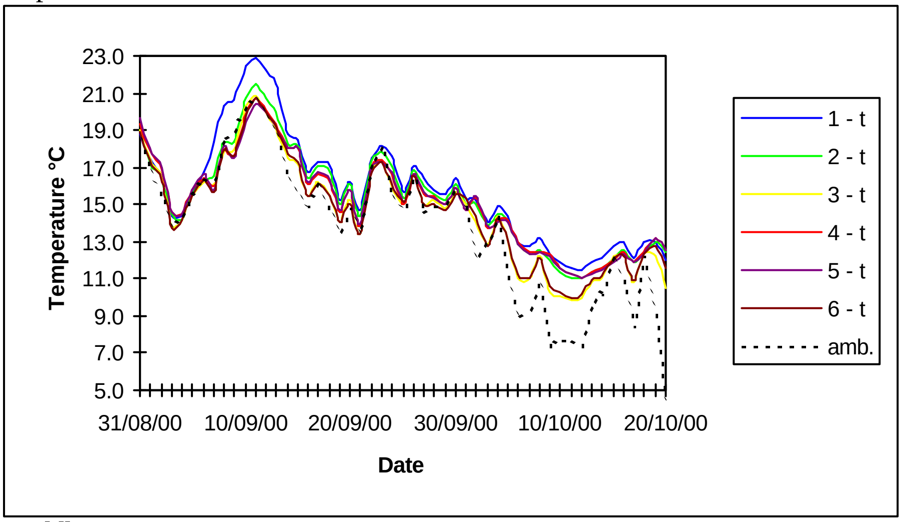

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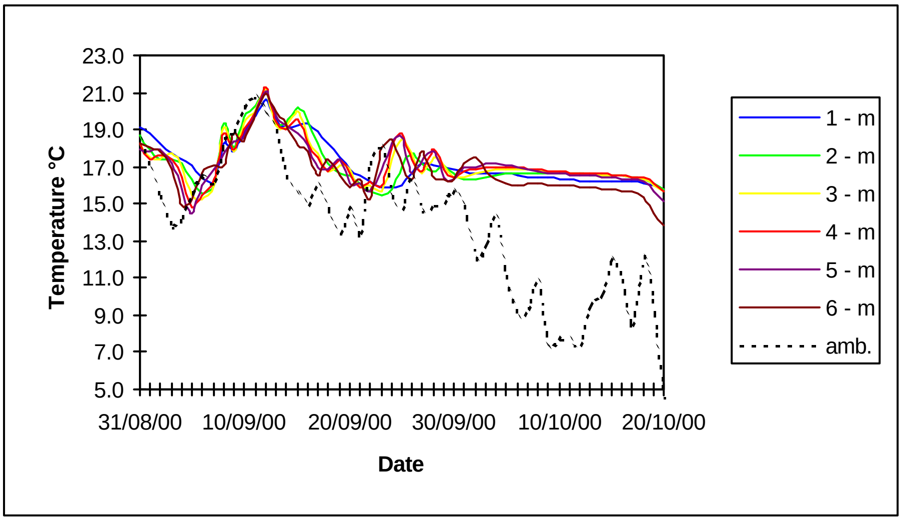

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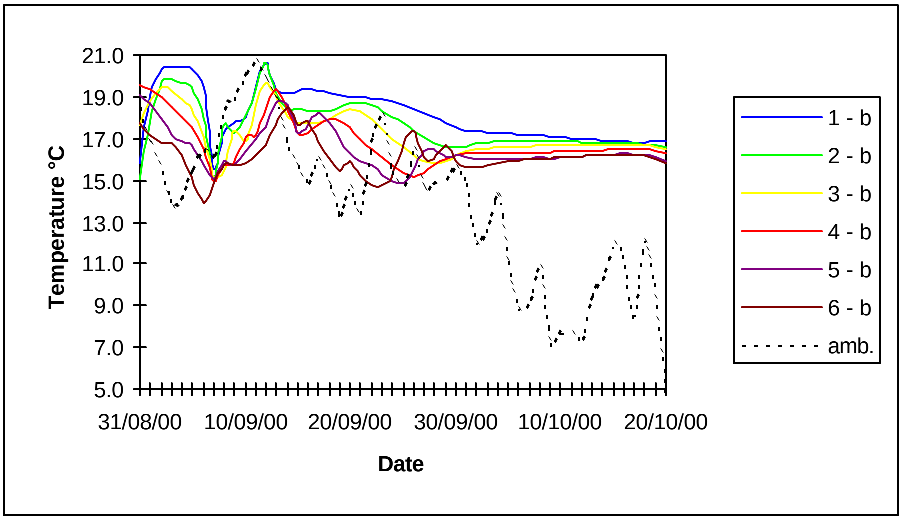

Fig 4.Temperatures at 3 depths and 5 columns in store 5.

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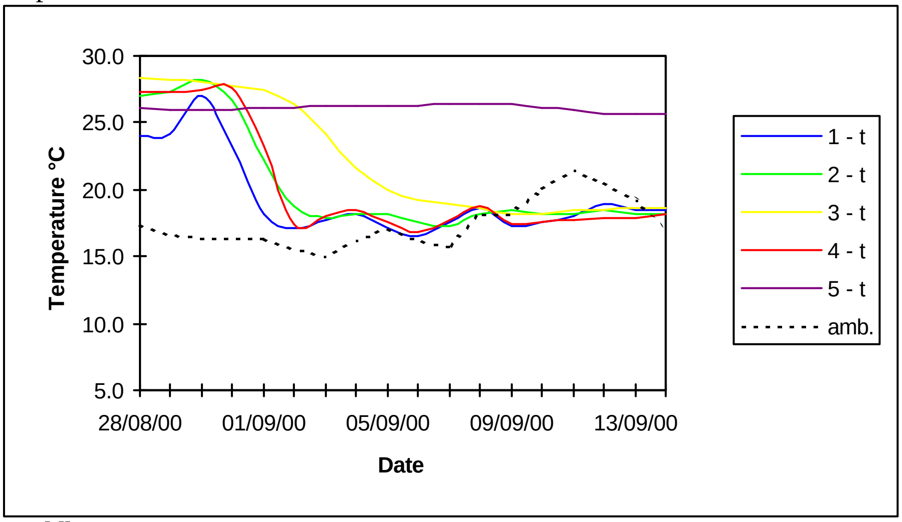

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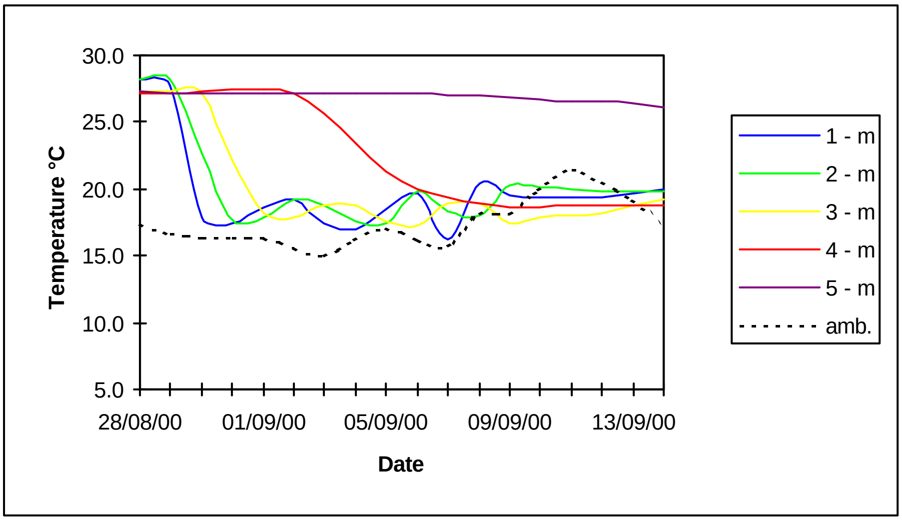

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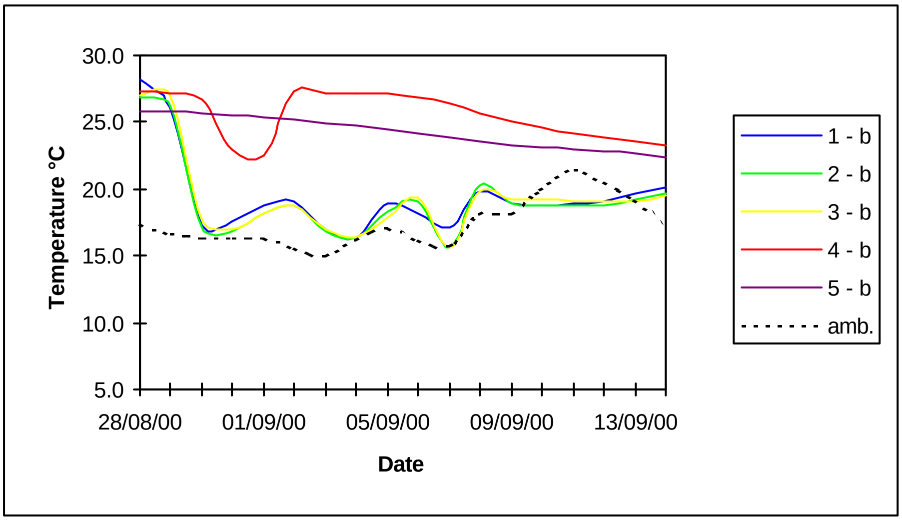

Fig 5. Temperatures at 3 depths and 5 columns in store 6.

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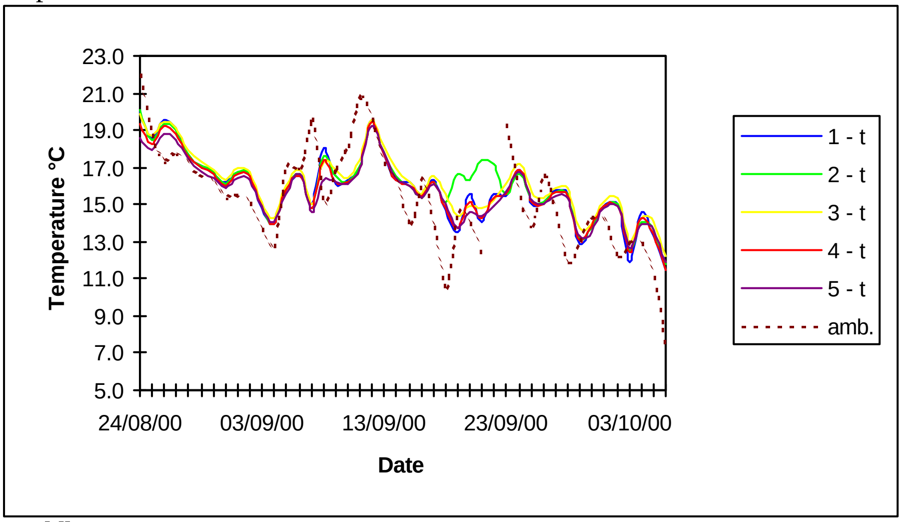

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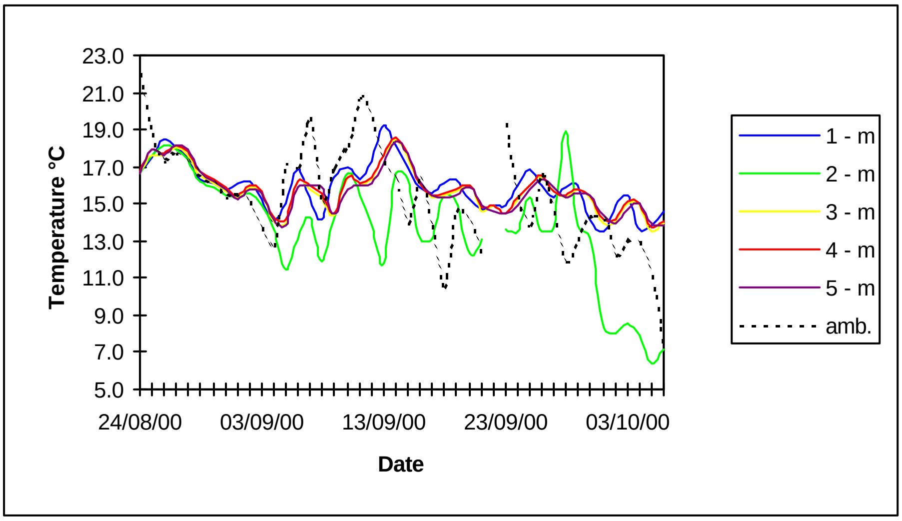

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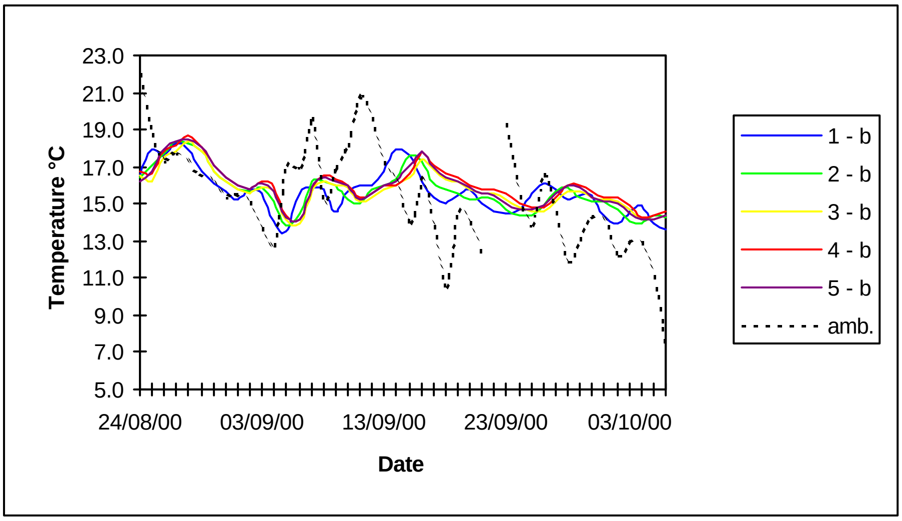


Fig 6. Diurnal temperature fluctuations at site 4 - near grain surface.

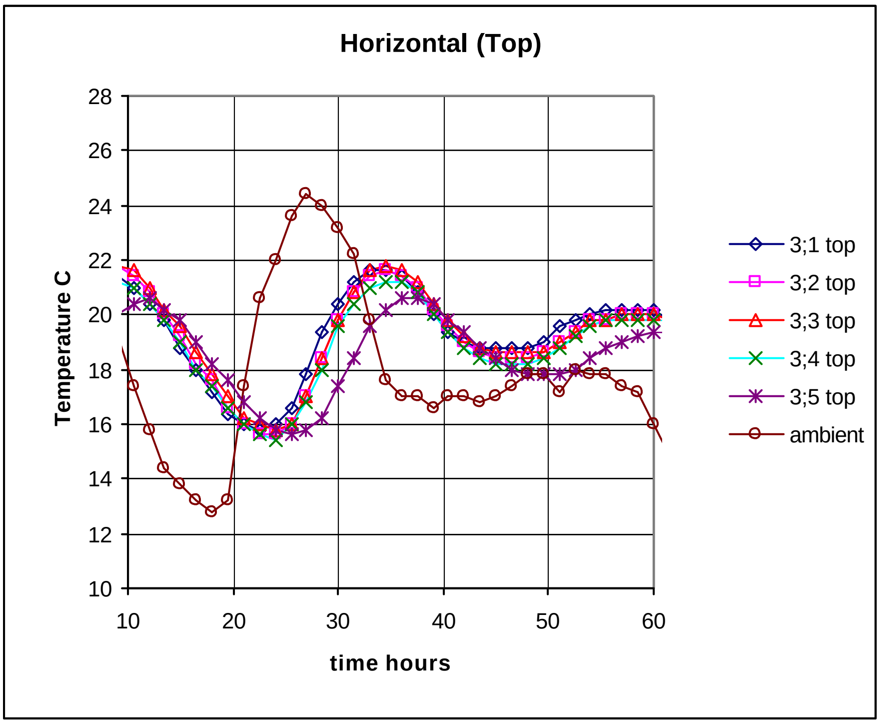


Fig 7. Diurnal temperature fluctuations at site 4 - near bottom of grain bulk.

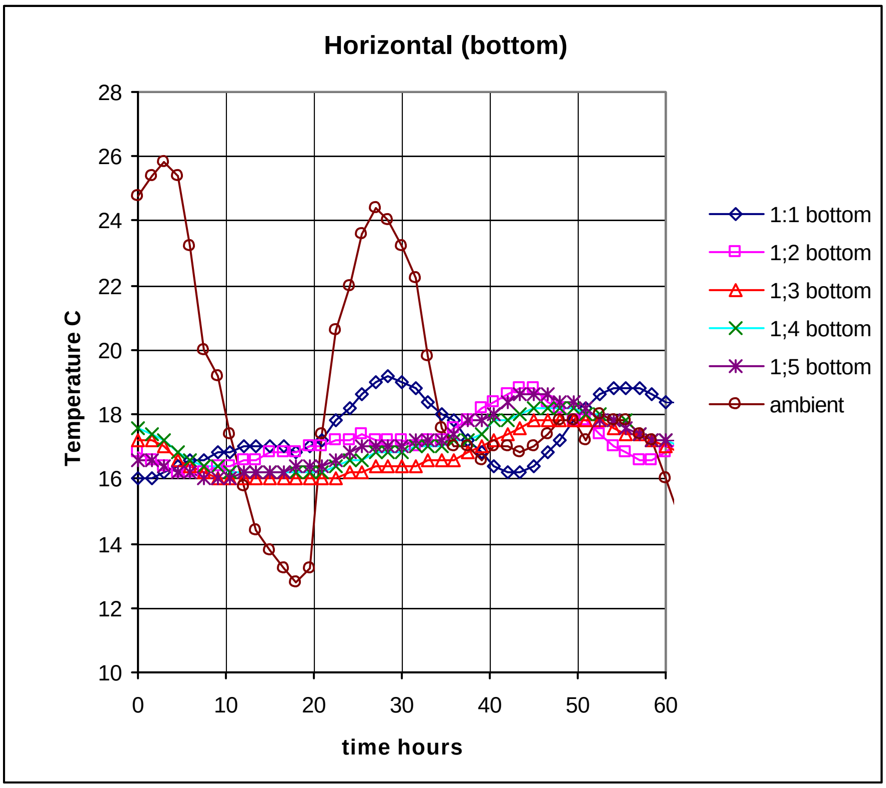


Fig 8 . System Characteristics and working point of the systems.

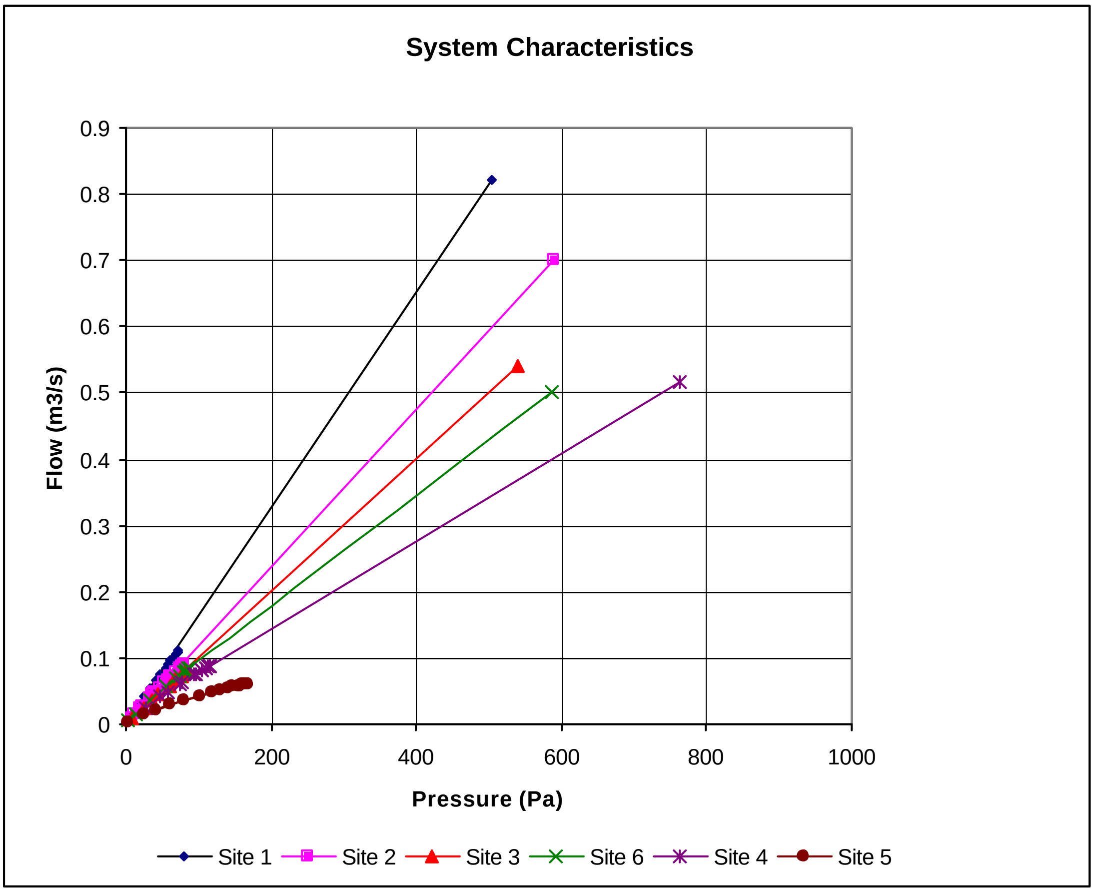


Fig 9. Detail of the individual air flow.

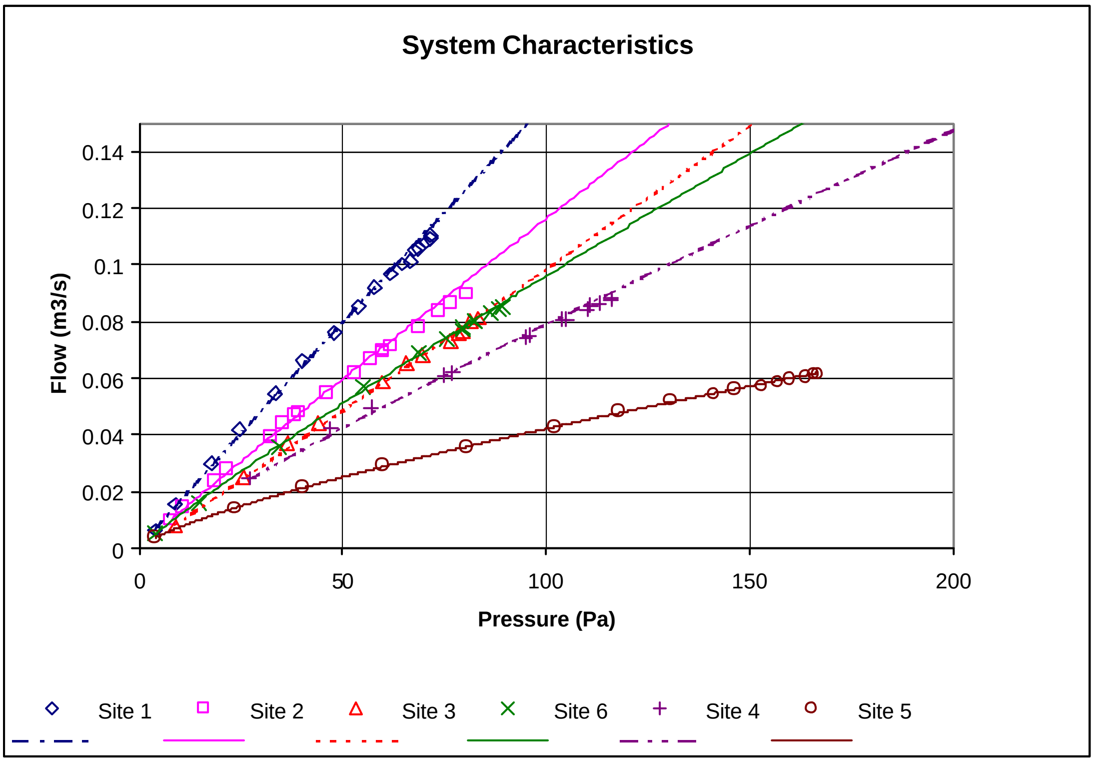
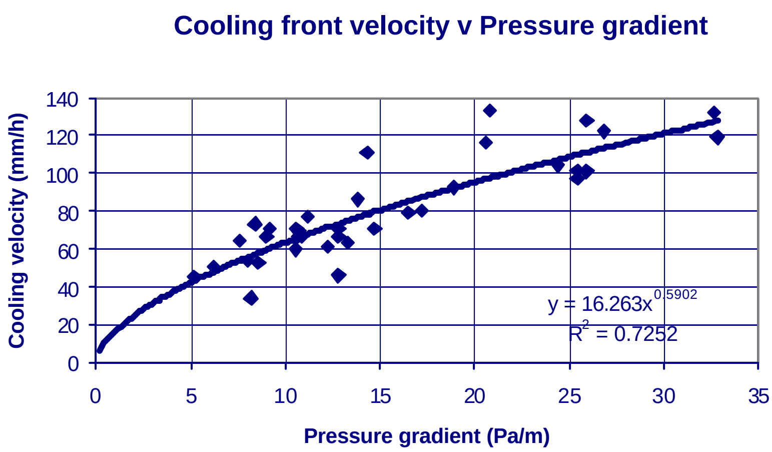


Fig 10. Correlation of cooling front velocity with pressure gradients based on observations on 6 farm sites.



Part 3

VENTILATION COMPONENT PERFORMANCE

D. BARTLETT

SUMMARY

Five different perforated duct sections were tested with wheat, oilseed rape (OSR) and beans to establish their air pressure/flow characteristics in sucking and blowing. The horizontal and vertical air flow resistance characteristics for each of the crops was measured.

- Vertical air flow pressure drop results agreed with published data.
- Horizontal pressure drop was about 50% of that for vertical flow.
- There was no significant difference in resistance to air flow between sucking and blowing for either vertical or horizontal flow.
- Ducts with >30% outlet area offered little resistance to flow at working flow rates.
- Ducts with <10% outlet area restricted the airflow.
- Duct wall resistance was 16% lower when blowing wheat than when sucking.
- Seed size affected the duct outlet pressure drop in ducts with round outlets but have little influence on the pressure drop with slot outlets.

INTRODUCTION

Pressure measurements in the field (Part 2) have shown that 50% of the fan pressure is used to drive the ventilating air through the duct wall and the first 20 - 30 cm of grain. A detailed study of air flow and pressure drop in this part of the ventilation system was therefore made using a purpose-built test rig. Field work also indicated higher air flows during blowing than sucking so the detailed test program also considered both positive and negative flows through the test system.

The pressure drop through the duct wall is influenced by the quantity, size and distribution of air outlets and by the diameter of the duct. To measure this, a number of commercially available ducts were used and tests were repeated with oilseed rape (OSR), wheat and beans to establish the effect of seed size on duct air outlet performance.

Model studies (Part 1) have shown that in order to reproduce the observed temperature distributions in grain bulks the resistance to cooling air flows must be less in a horizontal direction than in a vertical one. Data collected in this study has been used to establish both vertical and horizontal air flow resistance coefficients so that reliable predictions about grain cooling rates with different duct spacings can be made.

METHODS

The test equipment

The test equipment is illustrated in Fig.1. Air was blown or sucked to/from the test duct section located co-axially in a small cylindrical test bin. The test bin walls were formed from steel mesh, lined with hessian cloth, to allow air to escape radially. Air was prevented from escaping vertically by a polythene sheet laid on the grain surface.

Air was moved through the rig by a variable speed centrifugal fan. The airflow rate was measured by a non-standard orifice plate, upstream of the fan. Air pressure measurements were made 1cm and 21cm from the duct wall in the grain in a radial direction away from the duct. A third static pressure measurement was made inside the duct.

These pressure measurements were repeated for each test at 14 different airflow rates. The physical details of all the test ducts and the properties of the seeds used are shown in Tables 1 and 2.

Vertical air flow resistance tests were made using the same test bin. Air was introduced into the bulk through a hessian cloth at the bottom of the bin and allowed to escape through the top surface. Radial flow was prevented by lining the bin walls with polythene. The static pressure just above the hessian floor was measured at a number of air flow rates.

Horizontal air flow resistance coefficients were obtained by correlating the radial air speed at the duct wall with the pressure drop between 1cm and 21cm from the duct surface. Allowance was made for the decreasing radial velocity by calculating a parallel flow path equivalent length between the pressure measurement points.

The fan capacity in the test rig limited the maximum air speed at the duct wall to less than 0.3m/s. Normal working air speeds for commercial units range up to 0.75m/s Some pressure differences could not be reliably measured at this low flow rate so some extra measurements were made using a small section of duct wall and air speeds up to 1.6m/s.

RESULTS

Test program

Two metal (tube1 & tube2) and three plastic (tube3, tube4 & tube5) ducts were tested. Three sucking and three blowing tests were made with each . A power law function, $dP/dh = k\,v^{n}$ was fitted to the pressure and velocity measurements. dP/dh is the pressure gradient in the grain (Pa/m), v is the air approach velocity (m/s) and both k and n are constants. The resulting values of k and n are tabulated. Fig. 2 shows an example of the radial resistance to flow results for wheat, both sucking and blowing, with each of the ducts. The differences between individual sets of results are attributed to variation in packing density of the seed in the test chamber.

Duct wall air flow resistance (Tables 3 and 4)

Fig. 3 illustrates the form of the results obtained for pressure loss across the duct wall. In this example the small difference between sucking and blowing is clearly shown. Table 3 shows the coefficients for air flow resistance at the duct wall in wheat for the test replicates. The values in bold type summarise all the comparable data sets for the specified tube and flow direction. The horizontal and vertical air flow resistance coefficients for wheat are shown in Table 4.

Results for oilseed rape (Tables 5 and 6)

Tables 5 and 6 show the duct wall coefficients and horizontal and vertical flow coefficients for oil seed rape. Only tubes 1 and 3 were tested but both were tried with and without hessian sleeves to limit seed leakage.

Results for beans (Tables 7 and 8)

Table 7 shows the duct wall resistance coefficients for tubes 1-4 with beans and table 8 shows the horizontal and vertical resistance coefficients for bulk beans.

Supplementary duct wall resistance tests

The metal ducts offered very little resistance to air flow so additional tests at higher air speeds were made to supplement the results obtained from the test rig using a section of flattened metal duct. The air speeds used were high enough to fluidise the seed layer so only suction measurements could be made. The results of these tests are shown in Fig. 4. These results supplement those obtained in the replicated tests where the pressure differences were very at the available test air flow rates.

Radial pressure profiles in Wheat

In order to confirm the predicted pressure gradients near to the duct wall a series of pressure measurements were made at 1cm intervals from 1cm to 15 cm from the duct wall (Fig. 5). These observations were only made at one air flow rate in wheat both sucking and blowing. They show that the measured radial pressure gradient is in good agreement with that calculated by the simple model.

DISCUSSION

Pressure drop across duct walls

All results for duct wall pressure drop (tables 3,5,7 and figs 3 and 4) show that blowing is associated with a smaller pressure drop than sucking. Metal ducts, with their much higher perforation rate (Table 1), showed only a very small pressure drop at working flow rates. The pressure drop through plastic duct outlets was nearly 30 times greater than that for metal ones.

Radial pressure profile

Both types of duct showed an increasing pressure gradient nearer to the duct both when blowing and sucking (Fig. 5). This pressure gradient was heavily dependent on the distance from the centre of the duct. The pressure gradient for the plastic duct started to increase more steeply near the duct surface than that for the metal one. This may be explained by the distribution of openings in the plastic duct and the much smaller total outlet area available.

The results of these measurements are shown in Fig 5. Also included in this figure is a plot of the predicted pressure gradient at the average of the blowing flow rates observed for the two ducts. This prediction is based on a radial air flow model using the k and n values from table 2. Static pressures in the duct were not measured but, using the correlations for pressure drop across the duct wall (Table 3), the duct pressure has been calculated and is also plotted in the figure.

Both ducts operated at similar pressures and flows in the test rig. The distribution of pressure drop in the two cases was significantly different. The metal duct had a smaller diameter but much lower resistance to air discharge than the plastic duct. Only 8% of the pressure drop was across the duct wall. The plastic duct, because of its larger diameter discharged air into the grain further from the centre line of the duct. The air speed in the grain was lower than for the metal duct but because it had much higher resistance to discharge, 48% of the system pressure was used to overcome the duct wall resistance. The pressure distribution predicted in the grain demonstrates the importance of duct diameter and air outlet area in arriving at a practical working pressure for the flow required.

These tests could only be made at 20% of working flow rate so, using the test results, a comparison of the two ducts was made at normal flow rates. At a flow rate of 0.55 m^3/s the metal duct required a fan pressure of 854 Pa and the predicted pressure in the grain next to the duct was 782 Pa. The plastic duct required 847 Pa to maintain a flow rate of 0.43 m^3/s and the predicted pressure in the grain next to the duct was 403 Pa. This 22% reduction in air flow with the plastic duct will be reflected in longer cooling times or the need to use closer duct spacings.

In general, ducts with less open area will need to be larger in diameter to match the performance of smaller ducts with greater open areas.

Vertical air flow coefficients

The repeat tests show separate sets of results which can be attributed to differences in packing the test bin. These differences are most noticeable in the OSR data (Table 6).

Effect of particle size on pressure drop at the duct wall

The pressure loss coefficients for the duct walls show that, as the seed size increases, there is a decrease in the interference between the seed and the perforations. For large seeds (beans) the duct outlets behave as unobstructed holes while for small seeds they behave more like the bulk seed resistance. The metal duct resistance was so low that the presence or absence of hessian cloth made little difference to the air flow characteristics with oil seeds. The same comparison with the plastic duct showed a 60% reduction in resistance when the outlets were covered with hessian cloth.

Table 1. Properties of seeds used.

	Wheat	Oil seed rape	Beans
Moisture content (% wb)	13.6	9.5	17.1
Porosity	0.3424	0.3025	0.3357
Specific weight (kg/m^3)	840.5	682.0	793.0

Table 2. Physical dimensions of the test ducts.

Tube No	1 (Metal)	2 (Metal)	3 (Plastic)	4 (Plastic)	5 (Plastic)
Outside diameter (mm)	250	300	350	350	500
Inside diameter (mm)			310	310	452
Discharging length (mm)	861	858	1005	1005	980
Outlet size Length or Dia (mm)	3	3	72	75.7	129
Width (mm)			2.37	1.5	4.71
Number of holes per m^2	67000	50000	202	196	107
Open area % of duct wall	47.4	35.4	3.45	2.23	6.55

Table 3. Duct wall resistance in wheat.

Duct Sample	Blowing		Sucking	
	k	**n**	**k**	**n**
Tube 1	133	1.49	115	1.43
	41	1.73	108	1.45
	170	1.63	90	1.67
			155	1.83
	115	***1.62***	***117***	***1.60***
Tube 2	***170***	***1.63***	***155***	***1.83***
Tube 3	3039	1.72	3531	1.75
	3093	1.76	2736	1.78
	2402	1.80	3517	1.71
	2845	***1.76***	***3261***	***1.75***
Tube 4	***2954***	***1.73***	***3748***	***1.75***
Tube 5	***2597***	***1.61***	***2891***	***1.62***

Table 4. Horizontal and vertical air flow coefficients in wheat.

	k	n
Horizontal flow	3398	1.15
Vertical flow	7106	1.27

Table 5. Duct wall air flow resistance in oilseed rape.

Duct Sample	Blowing		Sucking	
	k	**n**	**k**	**n**
Tube 1 + hessian*			191	1.77
	97	***1.38***	***54***	***1.23***
Tube 1	***114***	***1.41***	***65***	***1.31***
Tube 3 + hessian #	***906***	***1.96***	***906***	***1.96***
Tube 3 #	***2759***	***1.39***	***2759***	***1.39***

* These measurements were made at air speeds up to 1.2 m/s.

There was no difference between sucking and blowing for tube 3 with or without the hessian cover

Table 6. Horizontal and vertical air flow resistance coefficients in oilseed rape.

	k	n
Horizontal flow	***5970***	***1.05***
Vertical flow*	10893	1.12
	8404	1.11
	9649	***1.12***

*Variation in vertical resistance due to different packing density.

Table 7. Duct wall air flow resistance in beans.

Duct Sample	Blowing		Sucking	
	k	n	k	n
Tube 1			104	2.42
	102	***2.01***	***93***	***1.92***
Tube 2			***44***	***1.84***
Tube 3	***1521***	***1.95***	***1887***	***1.96***
Tube 4	***2852***	***2.03***	***3011***	***1.97***

Table 8. Horizontal and vertical air flow coefficients in beans.

	k	n
Horizontal flow	1703	1.47
Vertical flow	3303	1.56

Fig 1. Test equipment.

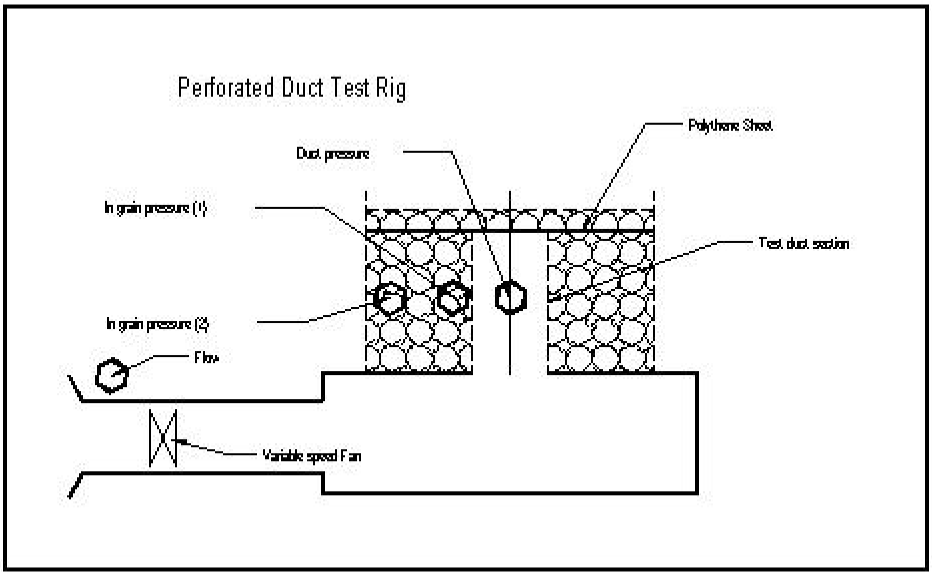


Fig 2. Resistance to horizontal airflow of five ducts under wheat.

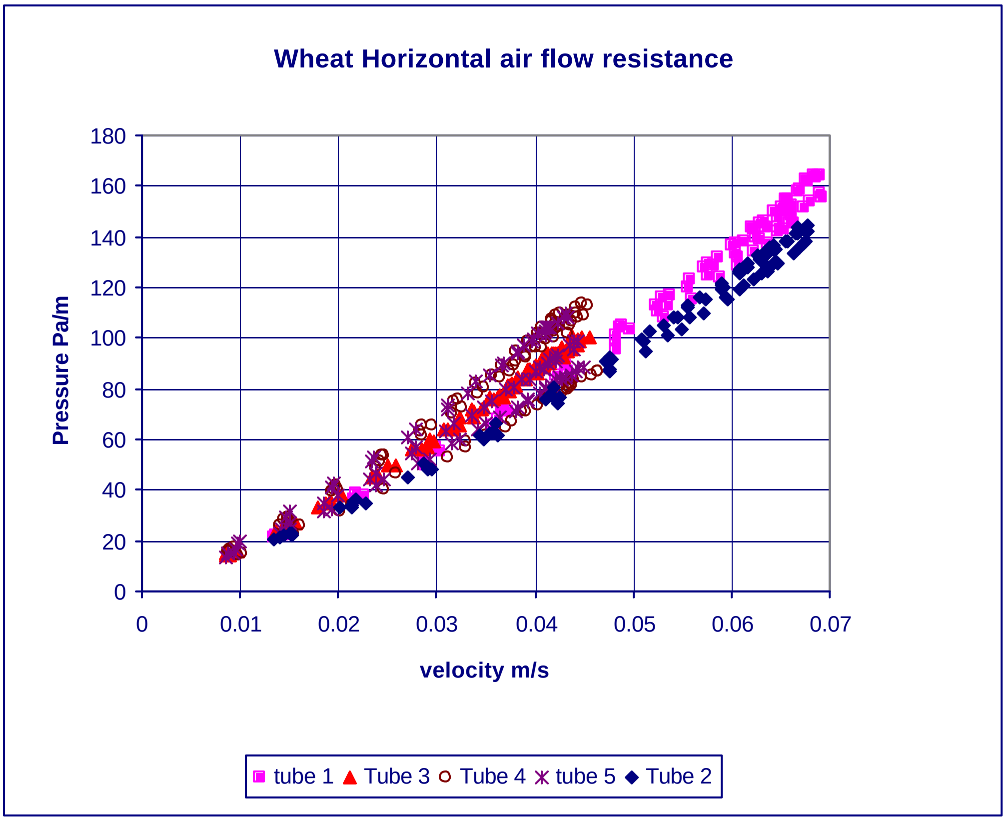


Fig 3. Example of pressure loss across a duct wall.

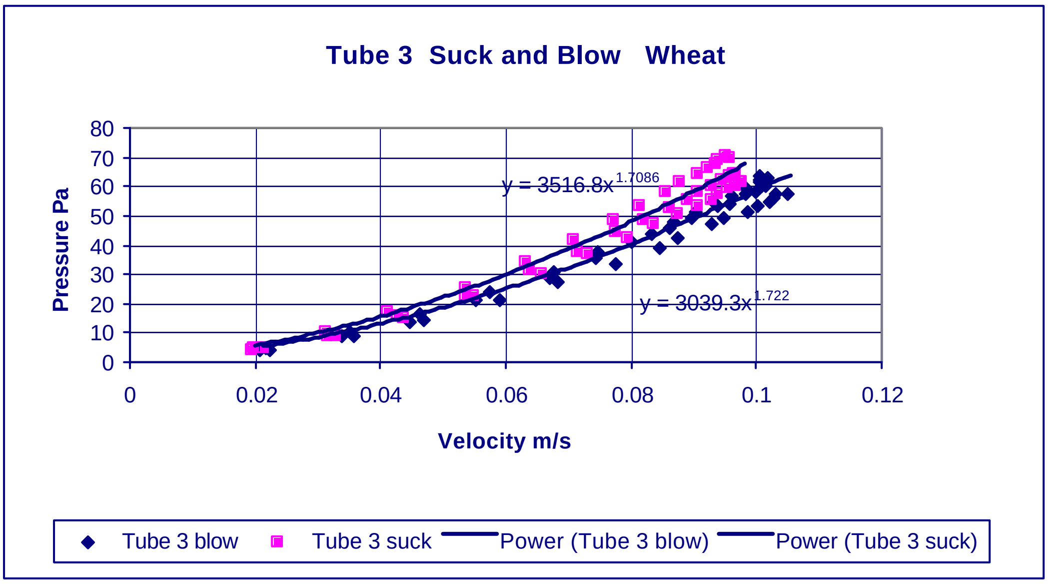


Fig 4. Pressure drop across the duct wall at higher air speeds (Duct 1/2).

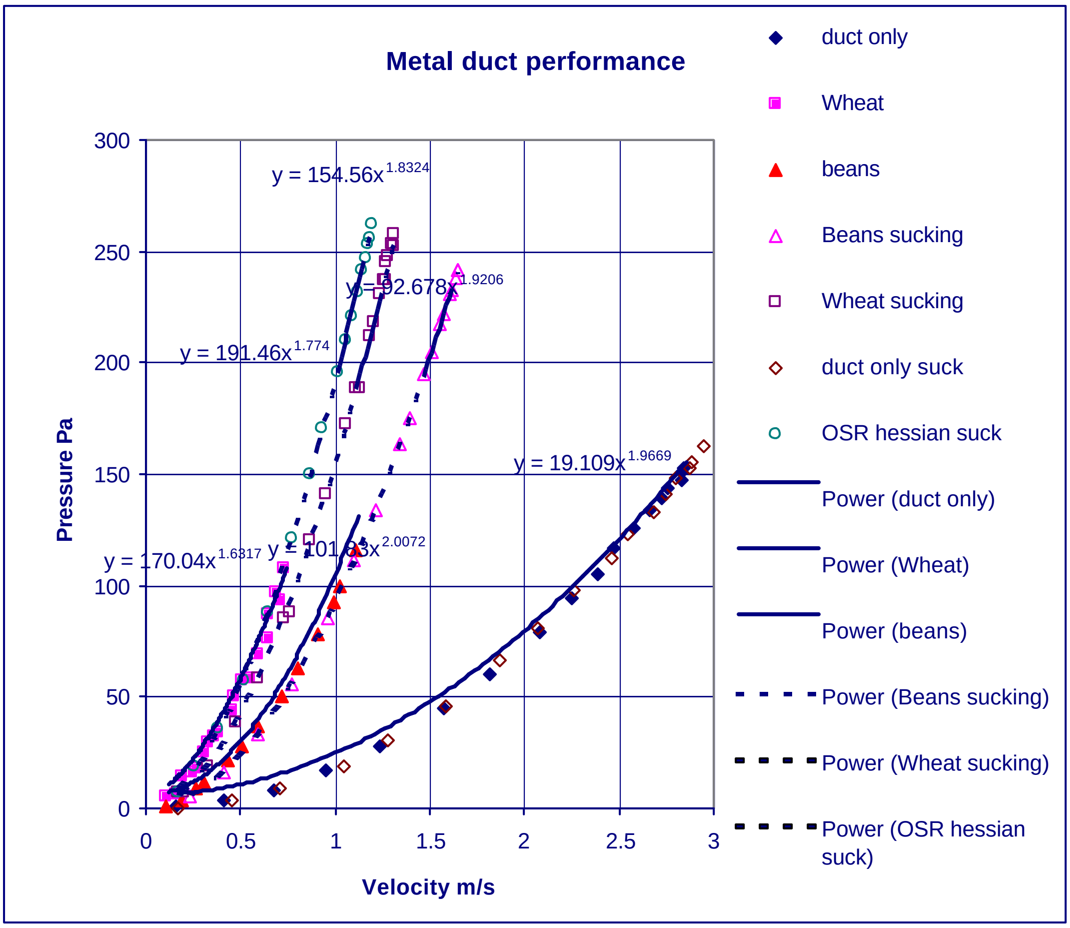
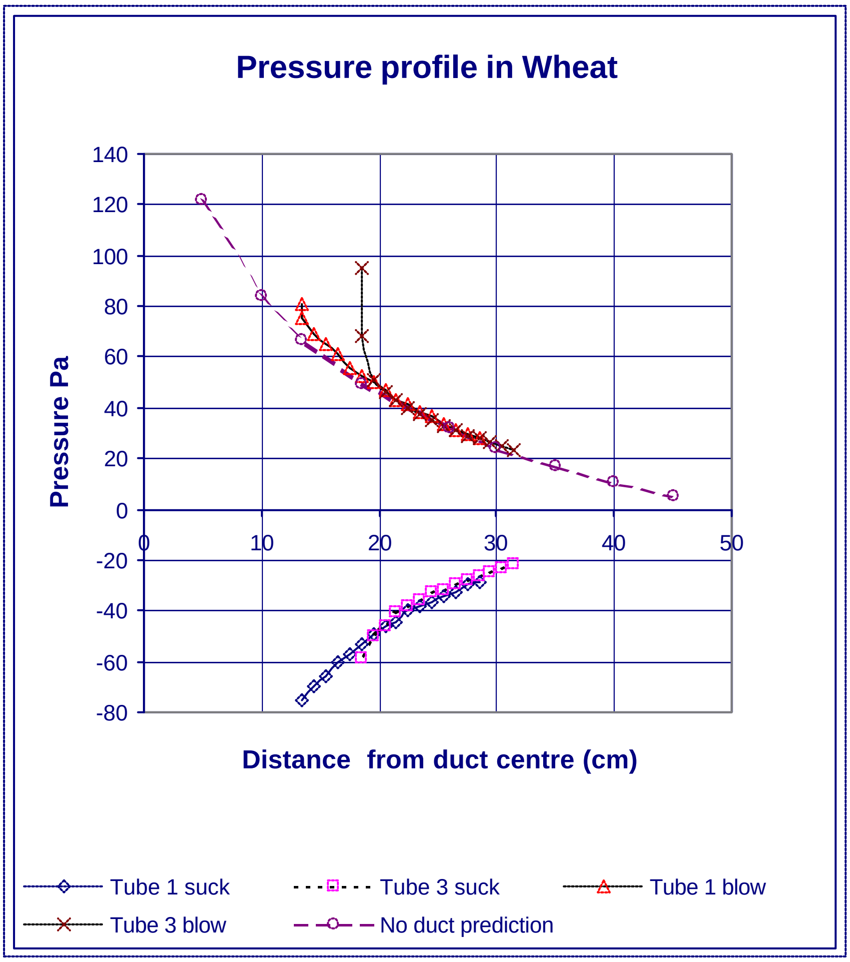


Fig 5. Radial pressure profile in wheat.



Part 4

OPTIMISATION USING THE COMPUTER MODELS

D. BARTLETT

INTRODUCTION - DESIGN METHODS

The practical design of ventilation systems to cool grain is based on the biological characteristics of the grain and pests that attack it. Warm, dry grain is at risk from insect attack, warm damp grain is at greater risk of attack by fungi but is also attractive to insects. It has been well established that in the UK grain pests can be effectively controlled by cooling with ambient air. Design ventilation rates of 0.0025 m^3/s tonne have been shown to be effective (Burges and Burrell, 1964). This ventilation rate can cool the grain to below 20°C within 14 days using about 100 h ventilation, so preventing any insects present completing their life cycle (Armitage et al., 1990).

Cooling warm grain with ambient air is most effective if only air at below average temperature is used. In 14 days (336 h) only about 200 h will be available for cooling. If one fan is shared between two ducts the cooling process for each duct must be completed in 100 h.

Vertical ventilation systems show a wide variation in air flow rate throughout the bulk that they serve. The natural distribution pattern from a vertical ventilator is radial; in a flat grain bulk the natural shape served is a cylinder. Most flat stores are rectangular so the natural distribution pattern must be distorted into a square by locating a number of ventilators on a square grid so that the whole bulk is served. The longest cooling path is between the ventilator and the corner of the square it serves.

The design problem is therefore to establish the ventilation unit spacing that will ensure a cooling time of 100 hours along the longest path.

The computer model

The computer model developed in Part 1 is based on a rigorous application of physics to the complex flow path created by vertical aeration systems in grain bulks. It has provided a valuable insight into the distribution of air and the movement of cooling fronts. It has also demonstrated a clear difference between sucking and blowing. This model repeatedly solves the air flow and heat and mass transfer equations in many thousands of elements to produce a result. This requires considerable computing power and takes many hours to execute. Modelling changes to the geometry of the ventilation duct and the grain heap are possible but can increase already lengthy modelling times by a factor of 10. This limits its usefulness for evaluating all of the combinations of variables in an aeration system design.

For these reasons, a much simpler model, based on cooling front velocities and the pressure and flow characteristics obtained from the farm tests (Part 2) and the vertical component trials (Part 3) was required to enable new and existing systems to be developed and tested quickly. This model needed modest computing power and executed very quickly so a large number of potential duct configurations could be compared. The simple model was compared with the complex one and good agreement demonstrated (Table 1).

Table 1. Comparison of duct spacing from precision and approximate models (blowing, tube 2).

Grain depth (m)	Duct spacing (m)	
	Complex model	Simple model
2	7.59	7.64
3	7.12	6.44
4	5.96	5.66
5	4.87	5.09
6	4.02	4.5

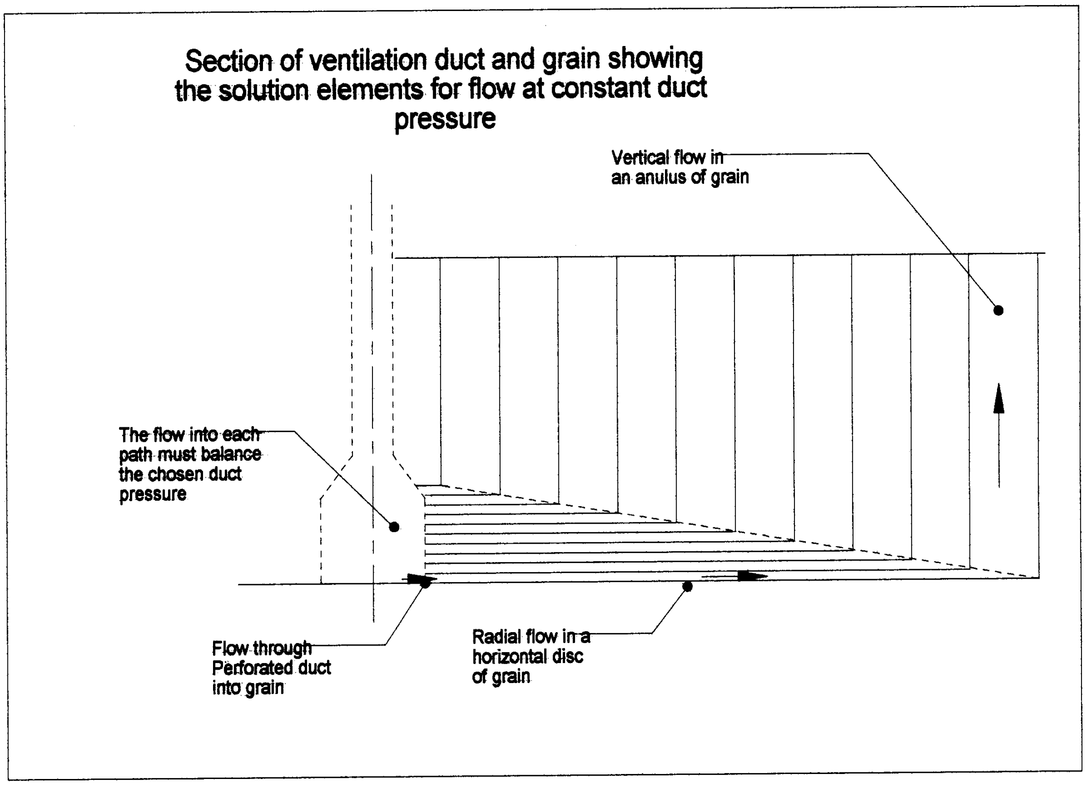
METHODS

The simple model

The simple model calculates the pressure gradient in small steps along the longest air path between the duct and the grain surface. By using the relationship derived from the farm studies (Part 2), the time taken for the cooling zone to move along this path can be predicted for a specified fan pressure. This model also calculates the total flow that the fan must deliver to maintain this pressure gradient in the grain.

Figure 1 shows how a cylindrical bulk of grain is divided to allow the calculation of pressure gradient along the air flow paths. Flow is assumed to take place radially in a shallow disk of grain and then parallel flow takes place in a vertical annulus of grain. These elements are nested so that together they represent the whole bulk of grain.

Fig 1. Section of ventilation duct and grain showing the suction elements for flow at constant duct pressure.



The flow is different into each of the nested paths. The one at the top, nearest to the duct, will have a high flow rate and the one at floor level and furthest from the duct will have a lower flow rate. The model balances the flows on these different paths to maintain a constant pressure inside the duct. The pressure gradient along each path also includes the pressure difference across the wall of the duct. The sum of the flow into all the paths is the total flow that the fan must deliver at the specified duct pressure.

Once the pressure gradient in the grain has been established, the time for a cooling front to travel the length of any path can be estimated by integrating the relationship for pressure gradient and cooling front velocity along the chosen path.

The model was used to calculate the cooling time for a range of grain bulk diameters at practical working air pressures and, by inspection, the diameter that gave the design cooling time could be selected.

The model can be used to investigate the effect of grain depth, the height and diameter of the perforated section of the ventilation duct and changes in air pressure. The pressure drop across the duct wall into the grain can be adjusted to reflect different outlet areas and the observed differences between sucking and blowing.

The model has been used to investigate the influence of design parameters on the capacity and performance of the cooling system.

RESULTS AND DISCUSSION

Duct diameter

Increasing the diameter of the perforated duct increases the open area into the grain. This larger area results in lower air speeds and lower pressure drop through the duct wall. Because of the increased diameter, the air speed in the grain is also lower so the reduced working pressure should result in an increased flow rate so that ducts can be spaced further apart.

Changes in duct diameter have much more effect at small diameters (150 mm) than at large ones. In cooling applications, there is little to be gained by using diameters greater than 500 mm.

Duct height

Increasing the height of the perforated section of duct also increases the open area. The fan pressure will fall and a greater volume of air will be moved through the grain. If the grain depth remains constant, increasing the perforated height will increase the air flow in the grain near the duct but will have only a small effect on the flow in the bulk of the grain away from the duct. Only modest increases in duct spacing will result from this change. Duct outlet height should not exceed 45% of the grain depth.

Duct perforation rate

The perforations restrict the flow of air through the duct wall. The size of the pressure drop depends on the air speed through the holes. If less than 5% of the duct wall is open, the fan will have to work against a higher pressure and the flow will be restricted. Closer duct spacing will be needed. Above 10% open area, there is very little effect on fan performance and duct spacing will be little affected.

Working air pressure

The air pressure developed by the fan will depend on the design of the ventilation system as outlined above. Depth of grain has a relatively small effect on total pressure. To cool deep bulks of grain, it is best to select a fan that will work against pressures up to 1000 Pa. More powerful fans cost more to operate and, when blowing, will add heat to the cooling air. If a small fan is used with a large duct the operating pressure and air flow rate will be low and extended cooling times can be expected.

Practical performance guides

The simple computer model has been used to calculate the duct spacing that will give 100 hour cooling times with a fan that is capable of generating 700 Pa for both sucking and blowing with ducts 1,2,3 & 5 (Detailed features of each duct can be found in Part 3, Table 2). The duct spacing for suction systems is based on 80% of the flow predicted for the blowing systems at the working pressure. Duct 4 is not reported separately because it behaved in the same way as duct 3.

Cooling grain in round bins

The natural air distribution pattern round a single duct takes the form of a cylinder of grain with the duct at its centre. This flow pattern fits a round bin effectively and Table 2 shows the relationship between bin diameter, bin depth and cooling time with a single central duct.

Table 2. Bin diameter cooled by a central single type 1 duct (blowing at 700 Pa).

Depth (m)	50 h	100 h	150 h
	Bin Diameter (m)		
2	7.2	10.7	13.6
3	6.0	9.0	11.5
4	5.2	7.9	10.0
5	4.4	7.0	9.0
6	3.8	6.3	8.2
7	3.4	5.7	7.5
8	2.8	5.2	6.9

Cooling grain in rectangular bulks

Practical grain bulks are rectangular so the duct spacings shown in tables 3 and 4 assume a rectangular grain bulk divided into square blocks each served by a duct. The distance between ducts ensures that the cooling criteria are met for the corners of the square served. Grain quantities only include that which is inside the square served by a duct.

Table 3. Metal ducts - Spacings and tonnage cooled in 100 hr at 700 Pa.

	Tube 1				Tube 2			
	Blowing		Sucking		Blowing		Sucking	
Depth (m)	Centres	Tonnes	Centres	Tonnes	Centres	Tonnes	Centres	Tonnes
2	7.6	96.2	7.1	84.0	7.8	101.6	7.1	85.7
3	6.4	103.0	5.8	84.7	6.6	109.0	6.0	91.0
4	5.6	104.9	5.1	87.1	5.7	110.2	5.2	92.0
5	4.9	102.9	4.5	86.0	5.1	108.9	4.7	91.5
6	4.5	100.0	4.1	84.8	4.6	106.5	4.2	90.7
7	4.0	96.2	3.7	79.5	4.2	102.4	3.9	87.7
8	3.7	90.9	3.4	77.4	3.8	98.0	3.5	84.0

Table 4. Plastic ducts - Spacings and tonnage cooled in 100 h at 700 Pa.

	Tube 3				Tube 5			
	Blowing		Sucking		Blowing		Sucking	
Depth (m)	Centres	Tonnes	Centres	Tonnes	Centres	Tonnes	Centres	Tonnes
2	7.3	89.1	6.6	73.6	8.1	109.2	7.2	87.4
3	6.2	95.4	5.5	76.7	6.8	116.1	6.1	93.2
4	5.4	97.0	4.9	80.0	5.9	118.6	5.4	97.0
5	4.8	97.1	4.4	80.7	5.3	118.1	4.8	97.1
6	4.3	93.2	4.0	79.0	4.8	116.5	4.4	96.9
7	3.9	89.6	3.6	74.7	4.4	113.0	4.0	92.2
8	3.6	85.4	3.3	71.1	4.0	105.4	3.7	90.9

The influence of cooling time on duct spacing

The simple model can be used to show the effect of design cooling time on the quantity of grain that can be cooled. Table 5 shows how cooled quantity of grain cooled by a single tube 1 type duct varies with depth and chosen cooling time. This table is based on the fan delivering a constant pressure of 700 Pa at all depths and grain quantity is based on a cylindrical column of grain. The quantity of grain cooled decreases as depth increases because the chosen pressure will move a decreasing quantity of cooling air through the increased depth of grain. The quantity of grain cooled in shallow layers is limited by the range of horizontal air flow in the bulk and an increased vertical flow rate near the duct. The optimum depth of crop for vertical aeration systems with a 0.9 - 1.2 m perforated section in wheat is 4.0 metres

Table 5. The influence of cooling time on tonnage and duct spacing at 700 Pa.

	Cooling time (h)					
	50		100		150	
Depth (m)	Tonnes	Centres	Tonnes	Centres	Tonnes	Centres
2	51.8	5.1	114.5	7.6	184.9	9.6
3	54.0	4.2	122.5	6.4	198.3	8.1
4	54.1	3.7	124.8	5.6	199.9	7.1
5	48.4	3.1	122.5	4.9	202.4	6.4
6	43.3	2.7	119.0	4.5	201.7	5.8
7	40.4	2.4	114.5	4.0	196.8	5.3
8	31.4	2.0	108.1	3.7	190.4	4.9

System resistance model

The simple model calculates the air flow that will balance any specified air pressure inside the perforated duct section. This information can be used to match a suitable fan to a ventilation duct to deliver the required cooling performance. Tables 6 and 7 illustrates the relationship between air flow and pressure when blowing tube 2 and tube 4 in wheat.

Table 6. Airflow and pressure in Tube 2 while blowing into wheat.

	Pressure Pa				
Grain depth	400	500	600	700	800
(m)	Air volume (m^3/s)				
2	0.28	0.33	0.38	0.43	0.48
3	0.26	0.32	0.37	0.42	0.47
4	0.25	0.30	0.35	0.40	0.45
5	0.22	0.27	0.33	0.38	0.42
6	0.20	0.25	0.30	0.35	0.40
7	0.17	0.23	0.28	0.32	0.37
8	0.15	0.20	0.24	0.29	0.33

Table 7. Airflow and pressure in Tube 4 blowing in wheat.

Grain depth (m)	400	500	600	700	800
	Pressure Pa				
	Air volume (m^3/s)				
2	0.27	0.31	0.35	0.39	0.42
3	0.26	0.30	0.34	0.38	0.42
4	0.24	0.29	0.33	0.37	0.40
5	0.22	0.27	0.31	0.35	0.39
6	0.20	0.25	0.29	0.33	0.37
7	0.18	0.23	0.26	0.31	0.34
8	0.15	0.20	0.24	0.28	0.32

Cooling of other crops

The model demonstrated here can be used with other crops (oilseed rape and beans) and can be used to approximate the ventilation performance of other products where the air flow resistance and bulk density are known. Cooling time predictions for products other than wheat should be treated with caution since the cooling front velocity model is based on wheat.

The tabulated performance data in this section are based on the geometry of the metal and plastic ducts tested. The model can be used to investigate the effects of alternative geometries and duct perforation rates and will provide air flow resistance data upon which fan selections can be based.

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