Potatoes - Storage
Gt Britain

POTATO MARKETING BOARD

SUTTON BRIDGE EXPERIMENTAL STATION

REPORT No. 6

STORAGE DESIGN AND EQUIPMENT FOR ENVIRONMENTAL CONTROL IN POTATO STORES

Part 2: Control of Environment

APRIL, 1973

POTATO MARKETING BOARD

50 HANS CRESCENT
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CONTROL OF ENVIRONMENT

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Part 2—Control of Environment

INTRODUCTION

As the title of this Report indicates, it comprises the Second part of a two part publication, aimed at providing practical data which the interested producer or his adviser may draw upon when designing new potato stores or modifying existing ones. Part 1: Storage Buildings, was published in May 1971, and is concerned with the fabric of potato stores and, in particular, with the effect of the building upon the storage environment.

Part 2

Potatoes are living organisms which survive only between definite environmental limits; within these they respond to their environment in many ways, for instance, the effect of temperature on sugar content and the effect of ambient water vapour pressure deficit (V.P.D.) on rate of moisture loss. It is sufficient at this point in the text to say that optimum storage conditions will require at the least some elementary control of the environment within the store.

Any given environment consists of a combination of a number of things—those of importance in a potato store are:

(1) Temperature

(2) Rate of air movement

(3) Water V.P.D.

(4) Illumination.

This report is concerned with illustrating the mechanics of providing artificial control for these four constituent factors. The extent of control for any one of a combination of these factors will depend upon specific circumstances with length of storage life and quality of out-turn required being dominant.

The normal storage life of the potato is terminated by, or through, a combination of:

Rotting—Wilting—Sprout Growth

but given suitable conditions, storage for up to 9 months is feasible. If sprout growth is prevented either chemically or by controlling the temperature, then total wastage at the end of the season would be approximately 7%, provided there are no losses from disease and a water V.P.D. of 1 millibar is maintained.

The average appreciation in the value of the crop at the end of the storage season was of the order of £4 during the period 1964-1970 (see Sutton Bridge Report No. 5, "Economics of Storage") which must cover all standing and running charges for the store. During the last two years of exceptionally large surpluses (1970/71, 1971/72) the increased value of the stored crop was less.

This appreciation is based on an overall ware loss of 10% for a crop sold in March. If the potatoes were marketed throughout the season the average increased return would be lower than £4. Conversely, a quality sample marketed late in the season could expect a greater return than £4. Any *excess* loss due to sprouting, disease and water loss will erode the margin and therefore the designer of a potato store and its environmental control equipment must be confident that the conditions provided will minimise the risk of such losses occurring whilst at the same time being extremely cost conscious!

SECTION 1

PRINCIPLES OF VENTILATION

The constituents of a potato store's environment which it is both practical and economic to control have been enumerated. The tuber temperature and indirectly the store air temperature will be the principal consideration throughout this report.

Temperature

Potatoes in store are living organisms which continue to respire giving off moisture and heat. The production of heat is significant and can be gauged from Table 1.

Table 1

Heat Output of Potatoes during Storage kJ/tonne/h (Btu/ton/h)

CONDITION OF POTATOES	kJ/tonne/h	Btu/ton/h
Immature potatoes at store loading	260	250
Potatoes stored at 10 °C (50 °F) Nov-March	63	60
Potatoes stored at 4·5 °C (40 °F) Nov-March	32-53	30-50
Well sprouted potatoes	105	100
Senescent sweetening June/July	211	200
Average for King Edwards at 4·5 °C (40 °F) to end of April	42	40

Note: Add 5KJ/tonne/h for i) Sprouted potatoes (0·5% by weight)
ii) Senescent sweetening (January)

In a 500-tonne store with potatoes stored at 4·5°C (40°F) producing 42 kJ/tonne/h (40 Btu/ton/h) 7kW of heat per hour is being produced. This heat must be removed by a combination of ventilation with air at a lower temperature and dissipation by conduction through the fabric of the store to ambient air which is at a lower temperature. If this heat were not removed, then the temperature of the potatoes would rise.

Example 1.1: 1 tonne of potatoes (1000 kg) producing 42 kJ/tonne/h (40 Btu/ton/h) has its temperature increased in 24 hours by

$$\frac{0{\cdot}238 \times 42 \times 1000 \times 24}{1000 \times 1000 \times 0{\cdot}86} = 0{\cdot}279°\text{C}$$

$$\left(\frac{40 \times 24}{2240 \times 0{\cdot}86} = 0{\cdot}5°\text{F}\right)$$

where 0·86 = specific heat of potatoes, 1 joule = 0·238 calories.

In practice, in a bulk store such a marked temperature rise will not occur for convection air currents are induced which will provide some cooling. However, if temperatures are to be controlled some ventilation is necessary. Ventilation in this context implies the introduction of cooler ambient air to remove the products of respiration principally heat. It does not refer to air treatment processes (air conditioning). In potato storage, what is primarily a ventilation system can be combined with a conditioning process to warm, cool and humidify the air. This, and closed air conditioning treatments, are discussed in later sections.

The optimum storage temperatures for potatoes are:

Ware market (long term)	4-5°C (39-41°F)
Ware market (short term)	5-8°C (41-46°F)
Processing market (long term)	7-8°C (45-46°F)
Processing market (short term)	10°C (50°F)

Although these figures may be considered optima, a wider range, 2°C-15°C (35°F-60°F) is often found in ware storage, depending on circumstances such as the length of storage period, incidence of latent disease, quality and condition of the sample, etc.

Footnote:-
The tonne (2204 lb) and the ton (2240 lb) are considered to be so close in value, that for the purpose of this report a distinction has not always been drawn between them in the text.

The removal of heat produced as a product of respiration will require continued if necessarily intermittent ventilation throughout the storage of the crop if the optimum temperatures are to be maintained. There are two exceptions to this:

(1) Immediately after storage loading when "curing" to heal wound damage requires a temperature of c. 15°C (60°F) for 10 days.

(2) Just prior to store unloading when the tuber temperature should be 10°C (50°F) to reduce susceptibility to damage.

It has already been shown (example 1.1) that in an unventilated stack of potatoes, the temperature may rise by 0·279°C (0·5°F) per 24 hours. As the surrounding air temperature rises, so convection currents will be established which will remove the heat. This state of thermodynamic equilibrium will apply as long as the air flow is not restricted by the presence of sprouts, or soil. Although convective air currents can prevent the stack as a whole from overheating, a stack temperature differential (T.D.) will be formed which may be of the order of 2°C increase in temperature for every 1 m (1°F per 1 ft) depth of storage.

The following equation enables the average potato temperature to be determined.

$$(Tp\text{-}Ta)^{2\cdot 8} = \frac{K}{8{\cdot}94 \times 10^{-4}} \times \frac{Qh \times 1{\cdot}8}{D}$$

where

Tp = average temperature of potatoes (°C)
Ta = average temperature of ambient air (°C)
K = coefficient of resistance to air flow
Q = metabolic heat (Cal/tonne/h) (1kJ = 0·238 Cal)
h = height of stack (m)
D = storage density (m^3/tonne)

The value of K ranges from $7{\cdot}7 \times 10^{-5}$ for clean potatoes, to 2×10^{-4} for a sample with 20% earth. (Sprouting increases resistance up to fourfold).

1000 calories = 1 Calorie.

By the use of forced draught ventilation (FD), greater storage depths are possible without the risk of overheating either in the general mass or in localised parts (due to the stack temperature differential).

Ventilation

In estimating the amount of ventilation required in a potato store, account must be taken of:

(1) average outside air temperature and range of temperatures;

(2) heat gains from potato respiration;

(3) solar radiation and thermal transmittance of the building (see Sutton Bridge Report No. 6, Part I);

(4) heat gains from equipment.

The ventilation required for heat removal can be calculated from the total heat gain in joules (Btu)/per hour by the following formula:

$$\frac{\text{kJ per hour}}{\text{Tem diff °C} \times 1{\cdot}3} = m^3/h \qquad \left(\frac{\text{Btu per hour}}{\text{Tem diff °F} \times 1{\cdot}14} = \text{cfm}\right)$$

Example 1.2: If a 500 tonne potato store is producing heat at the rate of 40kJ/tonne/h and has a fabric heat gain of 5500 kJ/h, it will require the following ventilation rate to dissipate the heat with an ambient ventilation air temperature 3°C lower than the potato stack temperature:

$$\frac{500 \times 40 + 5500}{3 \times 1{\cdot}3} = 6550 \ m^3/h$$

On a per tonne basis this is equivalent to a constant ventilation rate of 13 m^3/tonne/h (8 cfm/ton)

Because there are other factors which are more difficult to quantify but nevertheless significant, a recommendation for a ventilation rate of 13 m^3/tonne/h (8cfm/ton) would not be given. The removal of field heat, higher respiration rates due to higher storage temperatures and sprouting;

limited periods when ambient air temperatures are suitable and the need to limit stack temperature differentials all indicate that a greater ventilation capacity be provided.

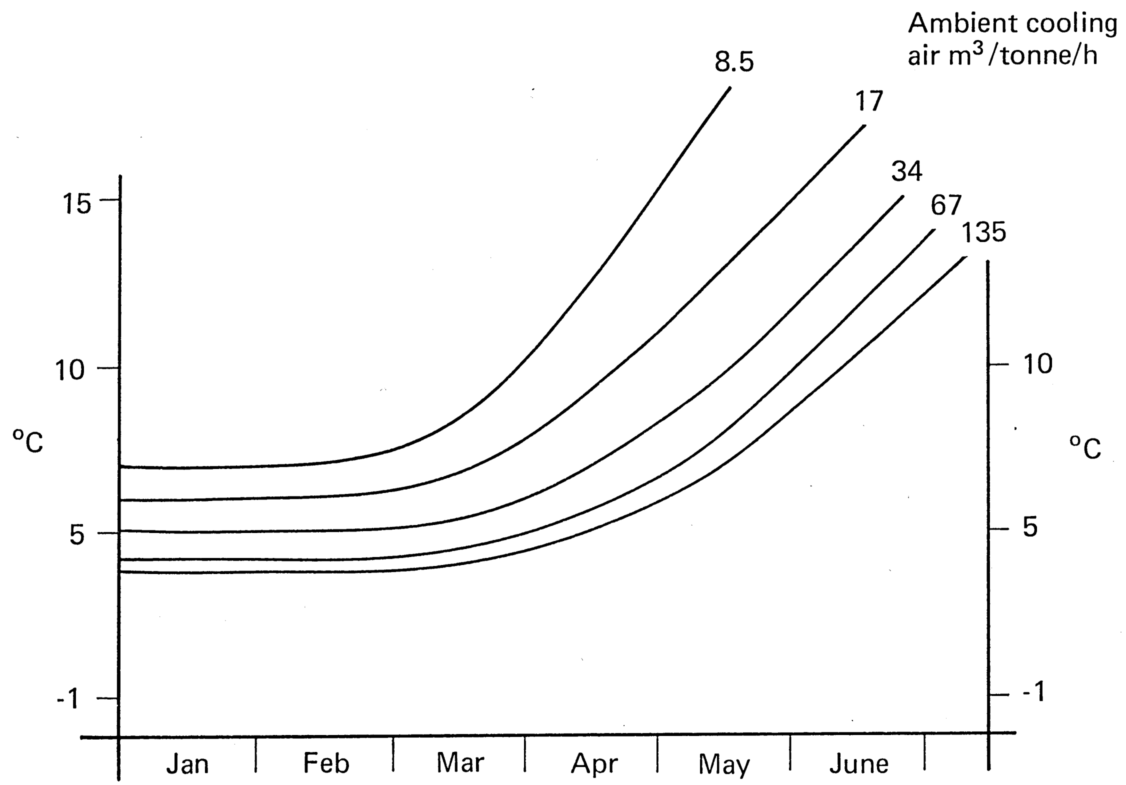
If the control system (see Section 10) is able to provide selective ventilation with ambient air it is possible to determine a relationship between product temperature and fan capacity using meteorological data for the locality in question.

Figure 1 estimates the ventilation rate necessary for the range of potato store temperatures normally required and dependent upon the length of storage time.

Fig. 1

Estimated minimum achievable storage temperature using ambient air cooling

ASSUMPTIONS:
1. Potato Store
2. Perfect store insulation
3. Store situation - East Anglia
4. Fan heating effect not considered
5. Automatic control of ambient air

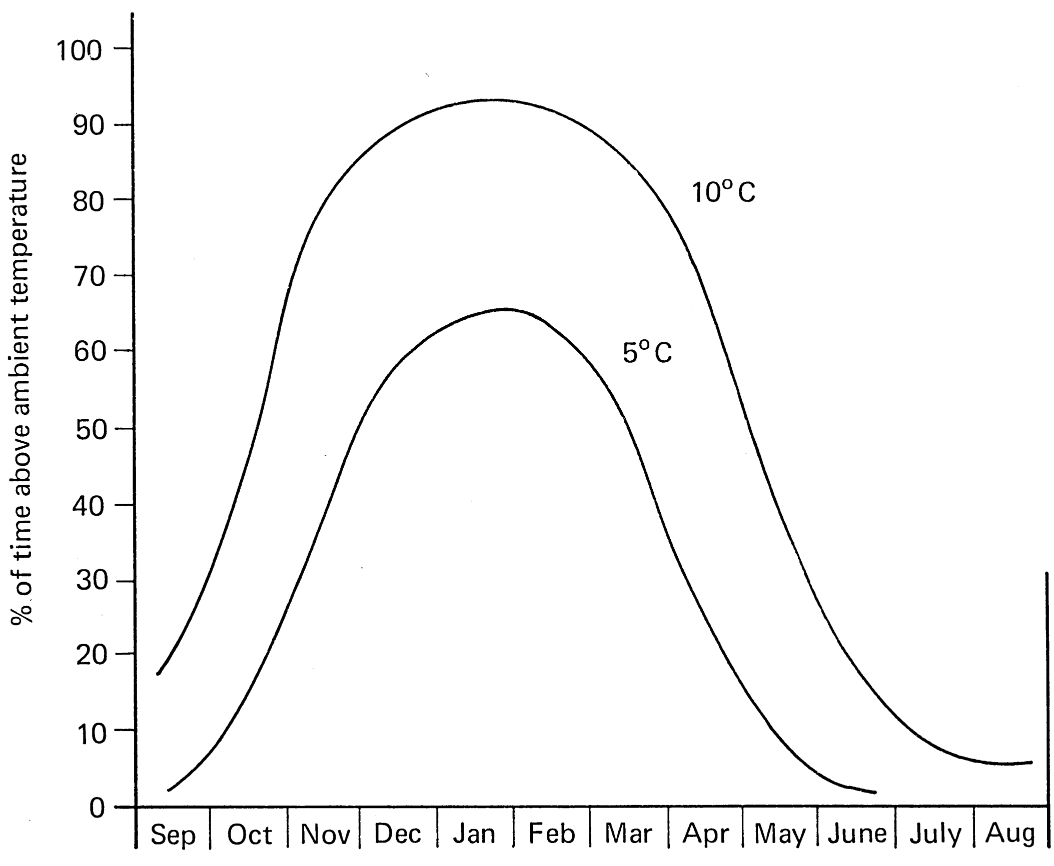


It is a general rule that the ambient air temperature must be below the desired holding temperature for at least 30% of the time if that average holding temperature is to be maintained (see Figure 2).

Example 1.3: If the desired store temperature were 5°C then this could be achieved in most seasons in the U.K. from the early part of November until the middle of April. Should the desired store temperature be 10°C then the period during which this could be maintained would lengthen from October until the early part of June. (From Figure 2).

Fig. 2

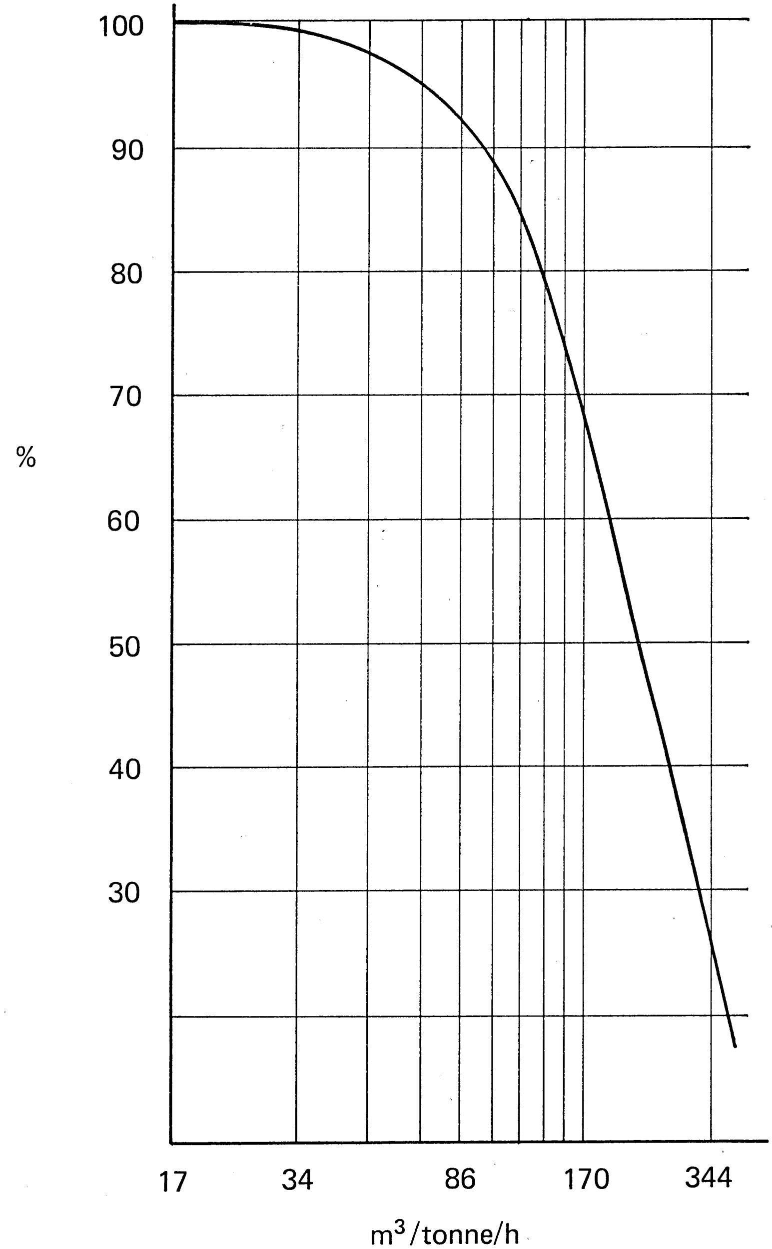
Percentage of time that store temperature is above ambient temperature. (Plotted for East Anglia)



A decision to use very high air flow rates (>135 m³/tonne/h (80 cfm/ton)) to obtain very low product storage temperatures and/or to extend the length of the storage season can be self defeating. This is because there is a rise in air velocity within the duct system (see Section 2) which in turn increases the system resistance. The static and velocity pressure created will be largely converted to heat within the stack and the "cooling efficiency" falls (see Figure 3).

Fig. 3

Cooling Efficiency



The "cooling efficiency" curve has been calculated for potatoes and assumes a log mean temperature difference of 2°C (3·5°F) between the crop and air.

In addition to the theoretical considerations in selecting fan capacity there are practical considerations which will modify the decision. They are:

(1) The practice of extended marketing of the crop which will usually mean a reducing tonnage in store at a time when the ambient air temperature is rising, in effect increasing the ventilation rate per tonne.

(2) A similar effect at the time of store loading when sequential cooling after curing may be possible.

(3) The thermal insulating properties of the store. Although it is not a linear relationship the higher the U value, the higher the ventilation rate needed (see Sutton Bridge Report No. 6, Part I).

(4) The acceptable stack temperature differential. It has been stated that with no F.D. the temperature gradient will be balanced by convective cooling currents (approximately 10 m^3/tonne/h (6 cfm/ton)) but the temperature difference (T.D.) between bottom and top of the stack could be 5°C (9°F). A ventilation rate of 67 m^3/tonne/h (40 cfm/ton) would produce a T.D. of 1·5°C (2·7°F) and 135 m^3/tonne/h (80 cfm/ton), a T.D. of 0·5°C (1°F) if the stack depth in each case is 3·5 m (11·5 ft).

(5) The effect of increasing air velocity on fan operating costs. The power requirement watts (horse power HP) rises as the cube of the velocity and the simple case of doubling the air flow rate will result in very much higher charges for capital installation and running costs.

Recommendations

For the producer storing potatoes for the domestic consumption market to a depth of 4 m (13 ft) from time of lifting until marketing, say end of April, a ventilation rate of 135 m^3/tonne/h (80 cfm/ton) is recommended *if* a 0·5°C (1°F) stack T.D. and the optimum storage temperature are desired. However, the practical considerations listed above and the wider range of storage temperatures which are often acceptable in practice would allow a lower rate of 67-86 m^3/tonne/h (40-50 cfm/ton) to be confidently specified in most instances in the U.K.

SECTION 2

RESISTANCE TO AIR FLOW

In order to select a fan, certain basic information must be known, namely:

(1) The quantity of air to be moved in a given time—cubic meters per hour, cubic feet per minute, etc.

(2) The pressure against which the fan is to operate—millimeters (inches) static s.w.g. or total t.w.g. water gauge.

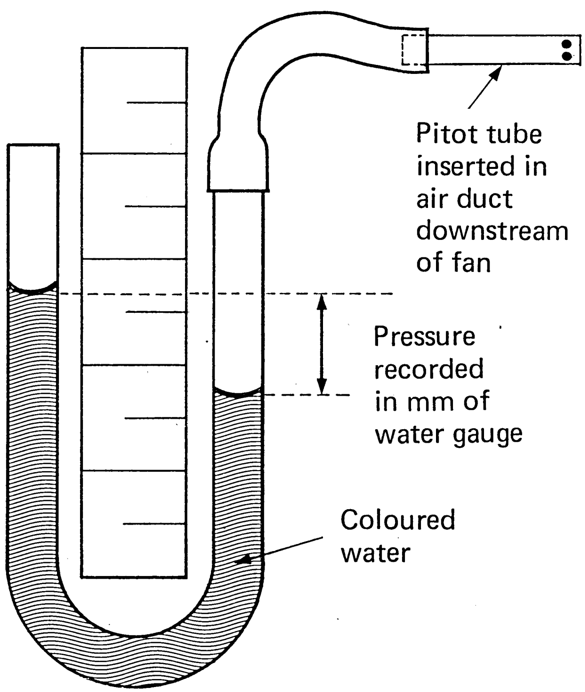
When selecting standard fans for cooling a tolerance of $\pm 5\%$ in performance is realistic. In considering the pressure against which the fan must work, the problem arises as to whether selection should be made on the basis of total pressure or static pressure. Strictly speaking, as total pressure gives a better indication of the capabilities of a fan, system resistance should be calculated and selection made on this basis. But to do this requires knowledge of both fan size and the nature of the connections between fan and duct. Very often these latter dimensions are not known and static pressure must be used as the basis of selection. Because the horsepower absorbed by a fan is a function of air delivery $\times$ pressure or (air delivery)^3, it is worth while considering the nearest fan below the duty required, as the savings in power will reduce capital and running costs. For the same reason, when any doubt exists as to the reliability of calculation of fan duty, margins should be allowed for extra pressure rather than extra volume.

Energy is lost in making air move faster, in negotiating bends, and in overcoming the resistance of heater banks and evaporator coils, and all combine to create a back pressure on the fan. In a potato store there are two components of this resistance, that due to the ductwork, and that due to the potatoes and the exhaust vents. Such resistances require pressure to overcome them. There are three aspects of this pressure.

(a) velocity pressure (Pv.) produced by speed of air movement;
(b) static pressure (s.w.g.) which maintains flow against resistance;
(c) total pressure (t.w.g.) the sum of the velocity and static pressures.

Fig. 4

U-Tube Static Pressure Water Gauge



In a potato store, static water gauge only is usually measured and for fairly clean, unsprouted potatoes, it will not exceed 0·75 mm s.w.g. per meter of height of the stack (0·01 in per foot). If a lot of soil has been included, it may be about 1·85 mm per metre (0·025 in per foot) s.w.g. and in extreme cases where a lot of sprouting has occurred it could rise to 7·5 mm per metre (0·1 in per foot) s.w.g. The resistance component due to duct work is fixed by the size and layout of ducts and can only vary as (volume flow of air)^2. In the case of single straight main air ducts with laterals at 90°, it is unlikely to exceed 12·5 mm (0·5 in) s.w.g. The total static water gauge could vary for a 3·5 m (12 ft) stack from 15 mm (0·62 in) s.w.g. to 43 mm (1·7 in) s.w.g.

With a design air flow of 67 m^3/tonne/h (40 cfm/ton), the current recommendation would be to provide the rated flow into a "back pressure" of 50 mm (2 in) s.w.g. This recommendation makes allowance for the worst conditions envisaged of bad duct design and a very dirty crop. If the design back pressure can be reduced there are very substantial benefits to the installer in terms of reduced capital and operating costs and higher cooling efficiencies.

Practical measurements with a well designed system and a clean crop of potatoes stored at a depth of 4 m (13 ft) have shown that a back pressure of less than 12·5 mm (0·5 in) s.w.g. can be expected. If this condition could be guaranteed in all seasons, then it would be possible to consider very cheap paddle or propeller fans probably reducing installation costs by up to 50%. Whilst obviously the installer has some influence over the design and execution of the duct work and can thereby ensure that this component of the system resistance is minimal no such control can be exercised over the weather and soil conditions and a more realistic pressure capability must be sought.

Recommendations

(1) For a well designed bulk store installation comprising main air duct and laterals, and containing potatoes loaded 4 m (13 ft) deep with soil extraction equipment on the input elevator plus swinging head extension to prevent soil cones. Rated air flow of fan 67m^3/tonne/h (40 cfm/ton) 25 mm s.w.g. (1·0 in s.w.g.).

(2) For an average installation otherwise as above except for the absence of soil extraction equipment. 38 mm s.w.g. (1·5 in s.w.g.).

(3) For an installation with design limitations (i.e. large number of bends in main air duct etc.) and with the possible presence of soil cones. 50 mm s.w.g. (2·0 in s.w.g.).

SECTION 3

FAN SELECTION

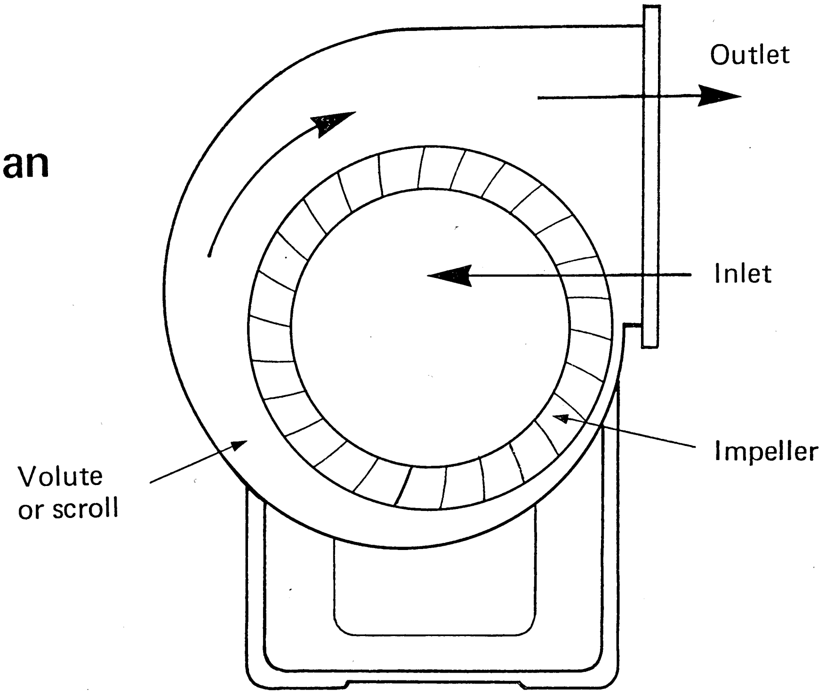
Choice of Fan Type

There are four types of fan—Propeller, Paddle Blade, Centrifugal and Axial. The two latter are of practical interest to the designer of a F.D. system, although propeller fans can be and are used where no duct system is involved, such as roof extraction in bulk stores or for ventilating box stores where there is negligible system resistance (see Section 8).

Centrifugal fans comprise an impeller which rotates in a casing shaped like a scroll as illustrated in Figure 5.

Fig. 5

Centrifugal Fan



There are a number of classifications within the broad grouping of centrifugal fans, of these the backward curve fan is the most common in agriculture. Whilst efficiency in relation to the axial fan is poor, the backward curve bladed centrifugal fan is capable of working against high levels of static pressure. This characteristic is useful in some crop conservation work such as grain drying where a 125-200 mm t.w.g. (5-8 in t.w.g.) is commonly encountered. As previously stated t.w.g. in a potato store seldom exceeds 50 mm (2 in) and this is well within the duty level of the average axial flow fan.

Axial fans comprise impellers of aerofoil cross-section rotating in a cylindrical casing, see Figure 6.

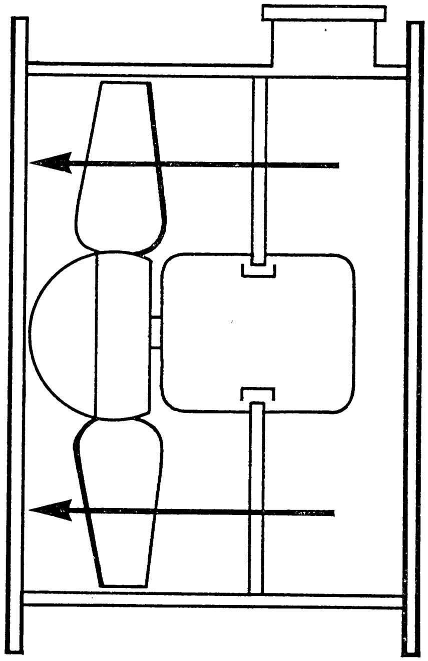
They are efficient, compact and simple to install. The straight through air flow enables the fan to be inserted directly into straight ducting. The duct system is consequently simpler than with centrifugal fans which require connections at 90°. Moreover, no floor space need be occupied. The overall size of an axial flow fan is substantially less than that of a centrifugal unit for an equal duty.

Because the simplest axial fan will deliver air against a s.w.g. of 65 mm (2·5 in) this type of fan would generally be recommended for all F.D. applications in a potato store. If the building is to be multi-purpose and there is a likelihood of drying both grain and potatoes with the same fan installation, then either a centrifugal fan should be specified or measures taken to improve the pressure development of the axial fan. This can be done by the use of guide vanes or multi staging techniques.

The selection of an axial flow fan rather than a centrifugal fan can, because of its improved efficiency, result in a reduction of between 50% and 75% of absorbed motor horsepower thereby reducing capital, installation and operating costs. The cooling efficiency in relation to the potatoes is also higher due to the smaller heating effect from the motor.

Fig. 6

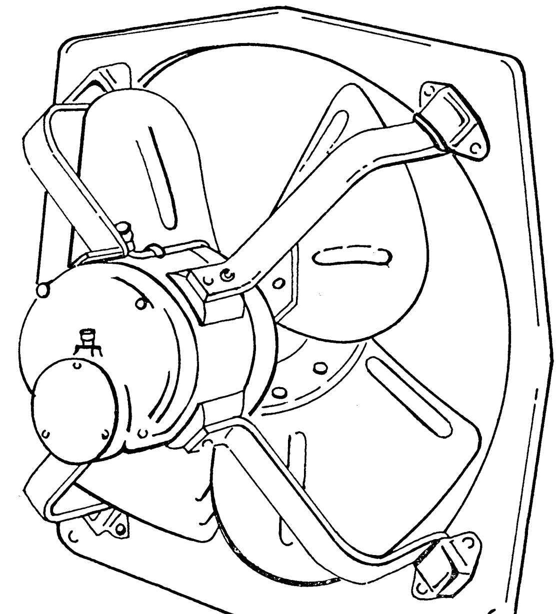
Axial Flow Fan



Propeller fans (see Figure 7), whilst having no application in a conventional ducted F.D. ventilation system in a bulk store, can be useful in box stores and for influencing air movement patterns in the exhausting of ventilated air in other situations. The propeller fan should not be used where the s.w.g. exceeds 12·5 mm (0·5 in). For applications of free intake and discharge they are usually the most practical and economical choice.

Fig. 7

Propeller Fan



Fan Noise

This is an aspect of installation not to be ignored if domestic dwellings are in the proximity. The nature of the cooling duty will cause fan operation frequently to be confined to night time working when noise will cause maximum annoyance. Generally it can be said that smaller high speed fans cost less than larger fans giving the same duty but are almost certain to be more noisy. Axial fans therefore tend to be noisier than centrifugal fans and because of the high tip speed of the fan blades a more objectionable high pitched whine is produced than the deeper pitch of the centrifugal fan.

Fan Motive Power

To an extent it has been assumed in this discussion on fans that they will be electrically driven. Providing electrical power is available on the site, automatic temperature sensing and control is simple. If electrical power is not available it would be imprudent to contemplate long or possibly even medium term storage, if temperature control by F.D. is to be relied upon. The internal combustion engine as a prime mover should be avoided for the following reasons:

(1) Efficiency is only of the order of 20%. The waste energy produced from either a petrol or diesel engine will be expressed as heat causing a temperature rise of the ventilating air of 2·75-5°C (5-9°F).

(2) In most instances the function of the fan is to cool and consequent on (1) ambient air must be that much cooler than the stack temperature to overcome the temperature rise—a problem at the beginning and end of the storage season.

(3) Larger air flows are necessary to compensate for the restriction in the number of hours of ambient air available at the right temperature (from (2)).

(4) Automatic control is difficult if not impossible to provide, further restricting the number of ventilating hours available.

(5) Higher operating costs.

(6) An increase in the V.P.D. accelerating weight loss from the potatoes.

For each useful horse power at the fan shaft at least 10 000 kJ/h (9500 Btu/h) will be evolved as waste heat most of which will find its way into the ventilating air. Unfortunately, as far as potato storage is concerned, the diesel engine driven fan installations commercially available are designed to make optimum use of this waste heat to provide a temperature lift for crop drying.

Recommendations

(1) For a store or ventilating system which is designed for potatoes only, choose a single stage axial flow fan. Improve its efficiency by installing guide vanes and inlet cones.

(2) For a multipurpose installation where the back pressure is very high or the range of operating back pressures is wide. Choose the most efficient of either a centrifugal fan or a multistage contra rotating axial flow fan.

SECTION 4

AIR DISTRIBUTION

Whatever rate of ventilation is decided upon, it is desirable that the volume of air introduced into the building shall be dissipated as evenly as possible through the stored crop. Distribution will be a function of resistance to airflow by the potatoes, duct sizing (cross section and length) and distribution and the situation and sizing of exhaust outlets. Obviously, the most uniform distribution would be achieved by the use of a plenum chamber over the entire floor area, the ventilated brick floor is the only economic example of such a system and is favoured by some producers erecting new bulk storage. The loose laid brick floor is not only economical to lay but provides the advantages of an unobstructed and exceptionally good distribution of ventilating air. The problem of soil build-up between the loose laid bricks obstructing air passages, has yet to be assessed. In all other circumstances in a bulk store some form of ductwork is required. This is likely to take the form of a main air duct supplying a number of lateral ducts with both the main and lateral ducts being either above or below ground level.

Calculation of the size of main ducts and branches is by the velocity method—in which velocity in various sections is selected, the velocity being reduced from a maximum in the main duct to a minimum where entering the potatoes.

Air velocities in ducts (forced draught)

Air intake	6-- 7·5 m/s (1200-1500 ft/min)
Main duct from fan	10—13 m/s (2000-2500 ft/min)
Lateral ducts	>10 m/s (>2000 ft/min)

The optimum cross-sectional area for a main air duct may be derived from:

$$\frac{\text{max fan output (m}^3\text{/h)}}{36\,000}=\text{area (m}^2\text{)}$$

$$\frac{\text{(max fan output cfm}}{2000}=\text{area (ft}^2\text{))}$$

In a system where there is allowance for convective airflows only it is essential to reduce frictional losses to negligible proportions by ensuring duct air speed does not exceed 3 m/s (600 ft/min): it follows that to achieve this aim when convective airflows are at a maximum 13·5 m^3/tonne/h (8 cfm/ton), all ducts must have a minimum free cross-sectional area of 1300 mm^2 (2 sq in) per tonne of potatoes ventilated.

In circumstances of forced draught ventilation 67 m^3/tonne/h (40 cfm/ton), adherence to the 1300 mm^2 (2 sq in) formula for sizing lateral ducts (as is current practice), will result in lateral duct velocities of 14·25 m/s (2800 ft/min)—higher in fact than the main air duct, and contrary to the recommended practice of reducing velocities along the system. It has been argued that a higher figure is acceptable where forced draught is only used occasionally. A change in management recommendations resulting in greater use of the fan would indicate that lateral ducts could, with advantage, be sized to provide 1860 mm^2 (2·88 sq in) of cross-sectional area per tonne of potatoes ventilated at 67 m^3/tonne/h (40 cfm/ton). The advantages are less turbulence, lower system resistance, lower initial fan costs, better air distribution. The disadvantage is higher duct costs.

The Main Air Duct

The main air duct's function is to provide a single attachment point for the fan and to serve as a feeder for all the lateral ducts distributing air under the stored crop. A large number of small fans, one attached to each lateral duct, is never as economical as a single large fan capable of the sum duty. If the main air duct is large enough to provide man access and the laterals have individual flap valves then control of individual parts of a store is possible.

Advice on the positioning of the main air duct is always related to the individual store configuration. Three possible arrangements are outlined in Figures 8, 9 and 10. (Pages 14, 15, 16).

In Figure 8 the main air duct is to one side of the long axis of the store. It may be either inside or outside the perimeter wall of the store or indeed be an actual part of the load bearing wall. The fan house is situated at either end of the main duct, although in some instances there is need for a fan installation at each end. Usually in the latter circumstance, because it is a very large store or where the extended marketing of the crop makes it advantageous to operate only one fan because of a reduced tonnage in store. The half duty air flow rates obtained by this arrangement

Fig. 8

Longitudual Main Air Duct

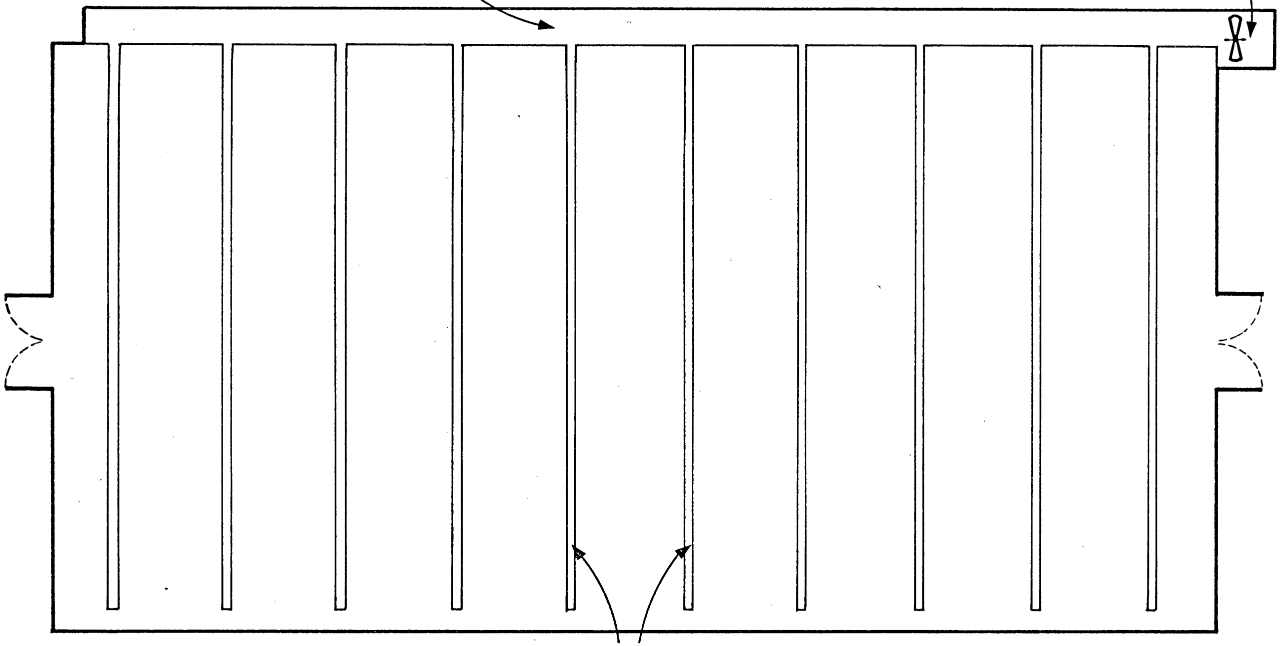


Fig. 9

Centre Main Air Duct

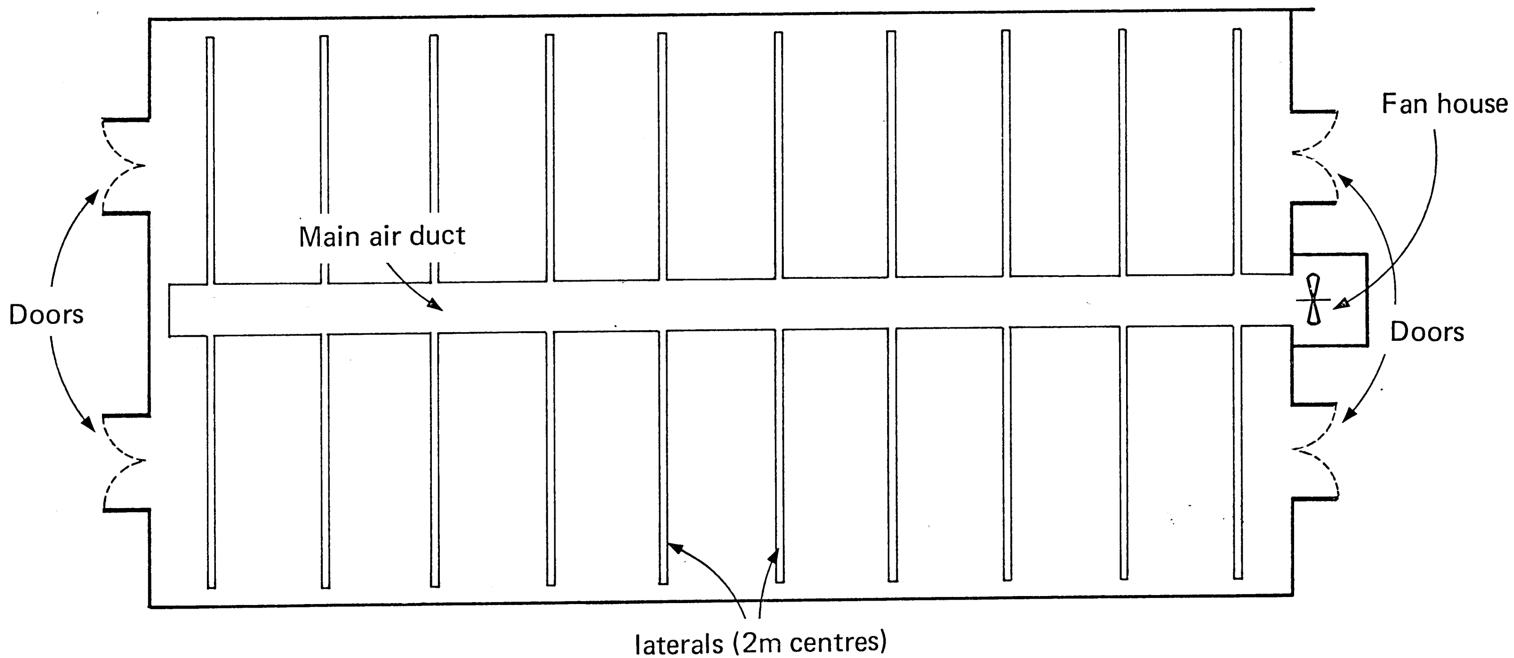
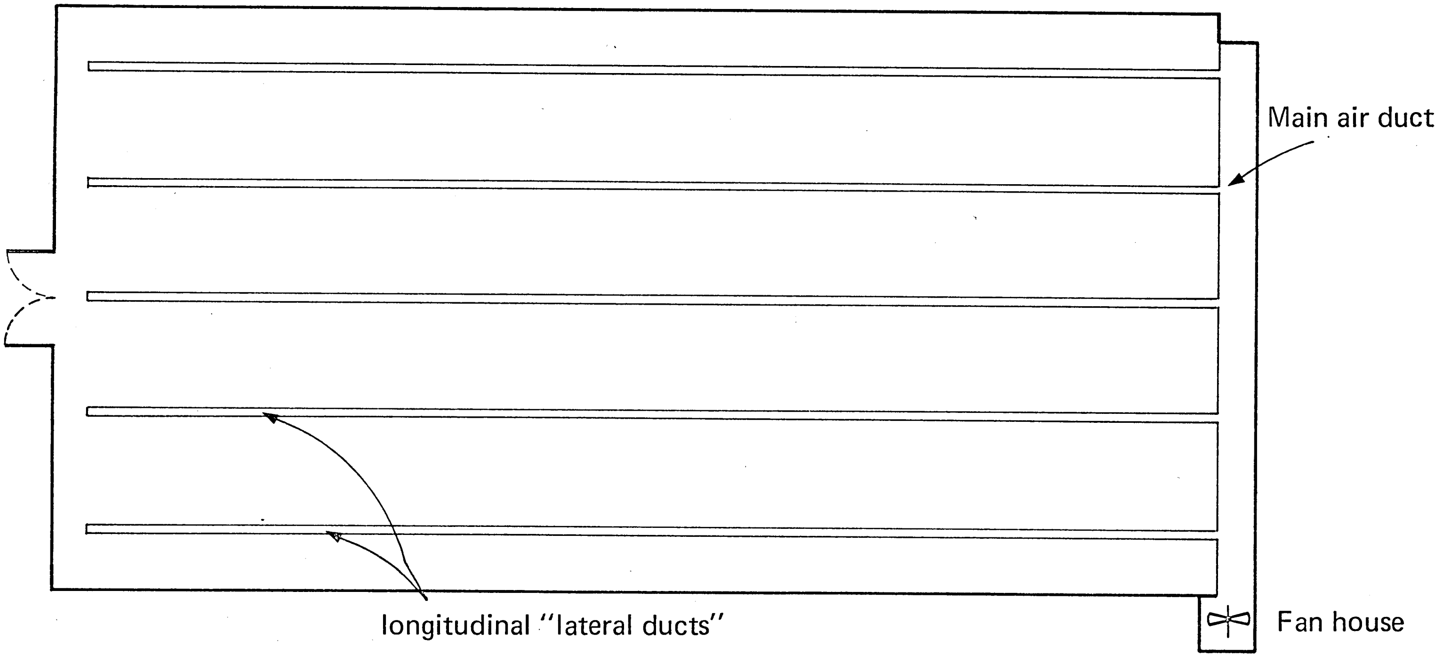


Fig. 10

Lateral Main Air Duct



are useful if recirculation of air within the store is required (see Section 9). Where two opposed fans are working the duct should be divided with a hinged door at the centre point to prevent air turbulence between opposing fans. Another advantage in a twin fan installation is that the size of the main air duct can be reduced.

In new stores there is a trend towards placing the lateral ducts underground and it is often thought necessary to do the same with the main duct. In general one would not advise this because:

(1) The expense of excavation is considerable.

(2) On many sites there are liable to be problems with high water tables.

(3) In the case of Figures 8 and 10 where the duct is inside or part of the wall of the store its cost may be partly defrayed against the cost of the load bearing wall.

In Figure 9 the main duct bisects the store. This layout is often used in wide span building (say greater than 12 m (40 ft)). Here again an above ground duct will frequently be cheaper than a sunken one, and has the added advantage that it can be used as the basis of a physical division, for separating varieties, treatments, etc.

In Figures 8, 9 and 10, the main air duct is straight and unobstructed and it is assumed that the fan is discharging axially into the duct. Any bend, particularly if it is acute, will cause turbulence of the air and uneven velocity pressures in that region affecting volume distribution to the laterals. It will also increase the system's resistance on the fan. If bends are unavoidable, they should be gradual and guide vanes should be fitted if practicable. This latter point is most important if the fan discharge is at right angles to the main run of the duct.

The interior of any main air duct should be smooth offering negligible resistances to airflow. Particularly avoid obstructions in the immediate vicinity of lateral duct take off points. Many proprietory sectional ducts offend this principle.

The construction of ducts often means that they are rectangular or square in section. As an axial flow fan has a circular discharge orifice this must be blended into the duct with a transformation piece. A centrifugal fan has a rectangular orifice. In each case avoid discharging air into a larger or smaller duct without a gradual transformation.

The formula for calculating the size of duct in relation to airflow has already been given. Within reason the greater the cross sectional area of the main duct the more even will be the distribution to the laterals. As the velocity in the duct decreases towards its closed end there is a regain of static pressure and hence discharge velocities increase in this area. This can be counteracted by tapering the duct and/or providing sufficient cross sectional area to keep initial velocities low.

Lateral Ducts

The spacing of these final distribution ducts is a compromise between cost and performance. The closer the spacing, the better the distribution, the further apart the cheaper they are. Much practical experience indicates a maximum spacing of 2 m (6 ft) centres. The flow of air and the problems inherent in wider spacing are illustrated in Figure 11. (Page 19).

The maximum length of lateral ducts is normally 12-14 m (40-45 ft). If lateral ducts up to 18 m (60 ft) long are essential then the recommendation of a cross sectional area of 1300 mm² (2 sq.in) per tonne of potatoes ventilated must be revised to 1860-1940 mm² (2·88-3·00 sq in). For ducts of 27 m (90 ft) length, stepping the duct is the most economical arrangement—see Figure 12. (Page 20). Never exceed 27 m (90 ft) long laterals.

Surface laterals are usually of slatted timber design with gaps between timbers of not more than 25 mm (1 in).

Welded mesh hoops covered with hessian are an alternative.

Sunken lateral ducts offer no advantages in terms of air distribution but they are more convenient and reduce tuber damage when loading and unloading a bulk store. Unlike surface ducts which are triangular the sunken lateral is either square or rectangular and fitted with cover boards, usually rough sawn soft wood up to 50 mm (2 in) thick and 100-150 mm (4-6 in) wide with gaps between each width of up to 20 mm (0·75 in). The calculation of cross sectional area should follow the same rules laid down for surface ducts. The ducts may either be cast into the concrete floor or a useful arrangement using precast concrete blocks is shown in Figure 14. (Page 21).

Increased static pressure at the closed end of a lateral as a result of reduced velocity (as described for main ducts) frequently presents an air distribution problem which is reflected in temperature variations within the store. The pattern of air velocities leaving the lateral is often as shown in Figure 15. (Page 22).

A simple remedial measure to even out the air flow is to vary the spacing gaps along the lateral (particularly easy with sunken ducts). The gap should decrease over the length of the lateral so that at the far end it is one third that at the point of connection with the main duct, i.e. the spacing ratio should be 1:3 over the duct length.

Example 4.1: Potatoes are stored 3 m (10 ft) deep in a building with a system of lateral ducts at 2 m (6 ft) centres each duct being 9 m (30 ft) long. Ventilation is at the rate of 67 m^3/tonne/h (40 cfm/ton). The lateral ducts are below ground level and have cover boards 150 mm (6 in) wide with spaced gaps of 12·5 mm (0·5 in) between each board.

The tonnage of potatoes ventilated by each lateral is:

$$\frac{3\times2\times9}{1{\cdot}585} = 34 \text{ tonne}$$

$$\left(\frac{10\times6\times30}{56} = 32 \text{ tons}\right)$$

where $1{\cdot}585 = \text{m}^3/\text{tonne}$
$56 = \text{ft}^3/\text{ton}$

Therefore total airflow in each lateral duct

$$34\times67 = 2278 \text{ m}^3/\text{h}$$

$$(32\times40 = 1280 \text{ cfm})$$

If the maximum recommended air flow in the duct is 10 m/s (2000 ft/min) the cross sectional area of the duct will need to be

$$\frac{2278}{10\times60\times60} = 0{\cdot}0633 \text{ m}^2$$

$$\left(\frac{1280}{2000} = 0{\cdot}64 \text{ sq ft or 92 sq in}\right)$$

Expressed alternatively

$$\frac{0{\cdot}0633}{34} = 1900 \text{ mm}^2 \text{ per tonne of potatoes ventilated}$$

$$\left(\frac{92}{32} = 2{\cdot}88 \text{ sq in per ton of potatoes ventilated}\right)$$

The total free area, which comprises the spaced gaps between cover boards per 9 m (30 ft) assuming duct measurements are 230 mm (9 in) wide by 255 mm (10 in) deep, is

55 (number of 12·5 mm gaps)$\times230\times12{\cdot}5 = 0{\cdot}158$ m^2 ($55\times9\times0{\cdot}5 = 225$ sq in or 1·56 sq ft)

Therefore Air Velocity at point of entry into the potato stack from the duct is

$$\frac{2278}{0{\cdot}158\times60\times60} = 4 \text{ m/s}$$

$$\left(\frac{1280}{1{\cdot}56} = 820 \text{ ft/min}\right)$$

Providing the velocity at this point does not exceed 5 m/s (1000 ft/min) this would be a satisfactory design calculation. However, in practice, the air is unlikely to leave the duct at a uniform velocity but may range from 2·25 m/s (400 ft/min) nearest the main air duct to 6·8 m/s (1200 ft/min) at the closed end of the duct.

Example 4.2: Using the design conditions as in example 4.1, it is possible to achieve uniform velocities for the air leaving the duct by spacing the gaps between the cover boards as follows:

First	3m (10 ft) (nearest main air duct)	19mm (0·75 in)
Middle	3m (10 ft)	12·5mm (0·5 in)
Last	3m (10 ft)	6·35 mm (0·25 in)

Because the average spacing is 12·5mm (0·5 in) the total free surface area remains 0·158 m^2 (1·56 sq ft). But the volume of air leaving the duct at its extreme ends will now be

$$\text{Far end} \quad \frac{230\times 6{\cdot}35\times 6{\cdot}8\times 3600}{1000\times 1000}=35\ \mathrm{m^3/h}$$

$$\left(\frac{9\times 0{\cdot}25\times 1200}{144}=18{\cdot}8\ \mathrm{cfm}\right)$$

$$\text{Near end} \quad \frac{230\times 19\times 2{\cdot}25\times 3600}{1000\times 1000}=35\ \mathrm{m^3/h}$$

$$\left(\frac{9\times 0{\cdot}75\times 400}{144}=18{\cdot}8\ \mathrm{cfm}\right)$$

This latter example indicates how a reasonably even volume of air leaving the duct can be ensured despite inevitable pressure variations within the duct. In all matters relating to the distribution of ventilation in a store it should be remembered that air will take the path of least resistance. Lateral duct ends should stop 300-450 mm (12-18 in) from the store perimeter wall if excess air loss up the smooth face is to be avoided.

Level filling of a F.D. store is of fundamental importance. If the potatoes are heaped, little or no air will reach the apex, most of it escaping lower down the face.

Fig. 11

Lateral Duct Spacing

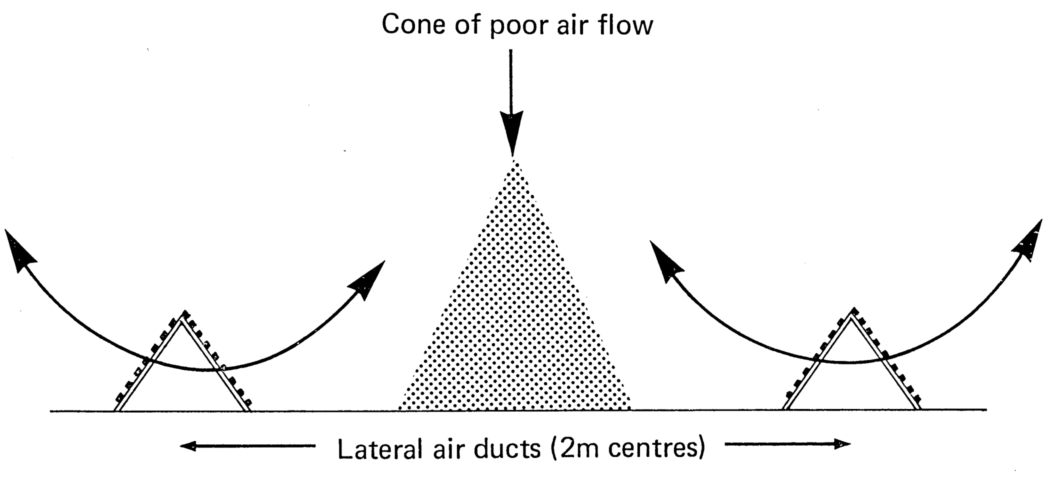


Fig. 12

Lateral Duct Sizing

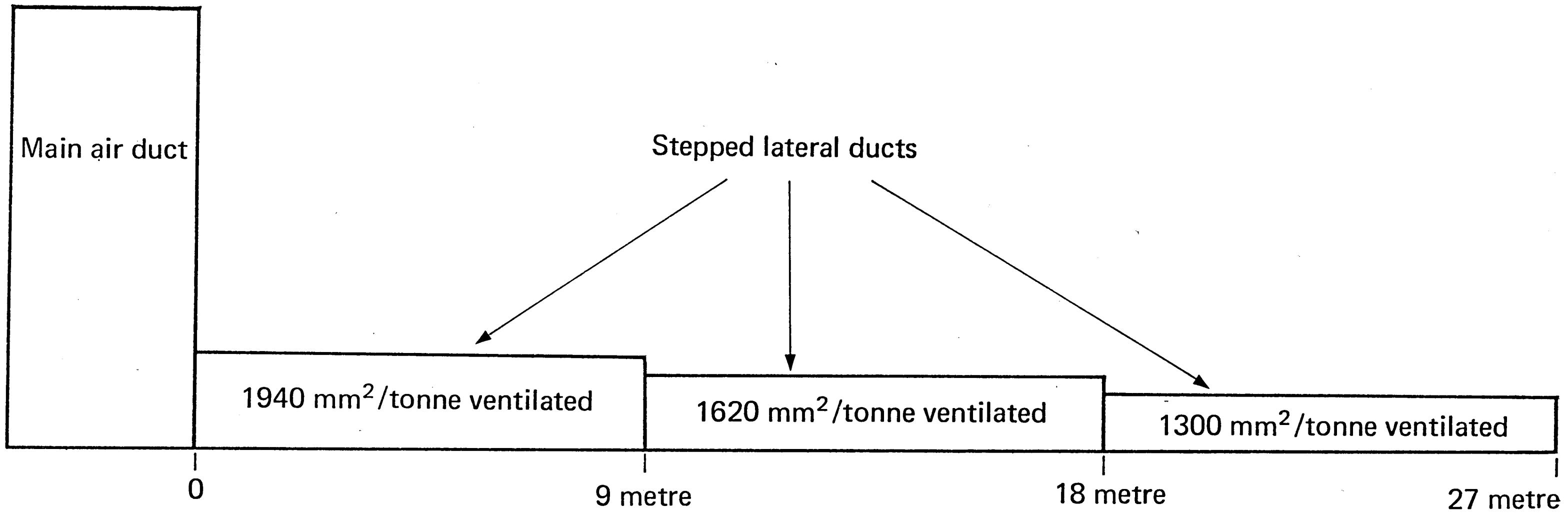
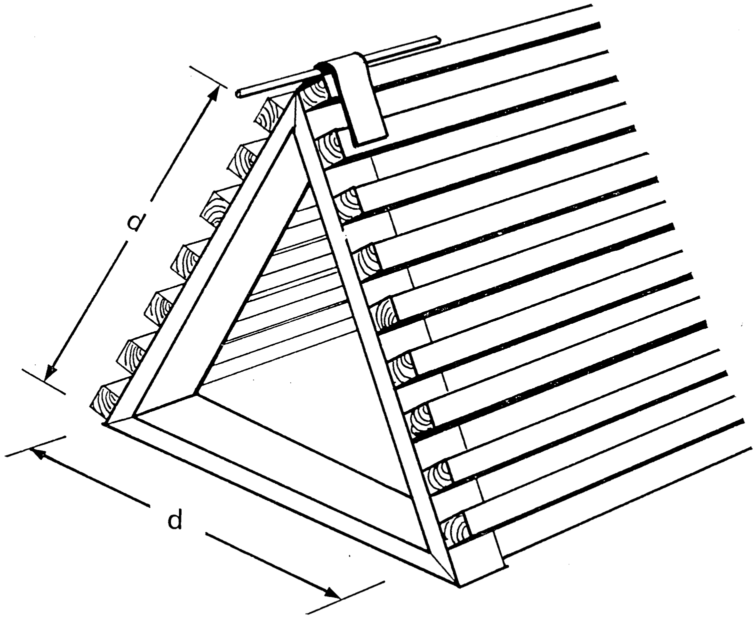


Fig. 13

Dimensions for Slatted Ducts



d (in.)	Tonnage Ventilated*
6	8
7	10
8	13
9	17
10	21
11	26
12	31
13	36
14	42
15	48
16	55
17	62
18	70

*Figure is for single-ended feed: duct is adequate for double this tonnage if centre fed

Fig. 14

Underfloor Lateral Duct

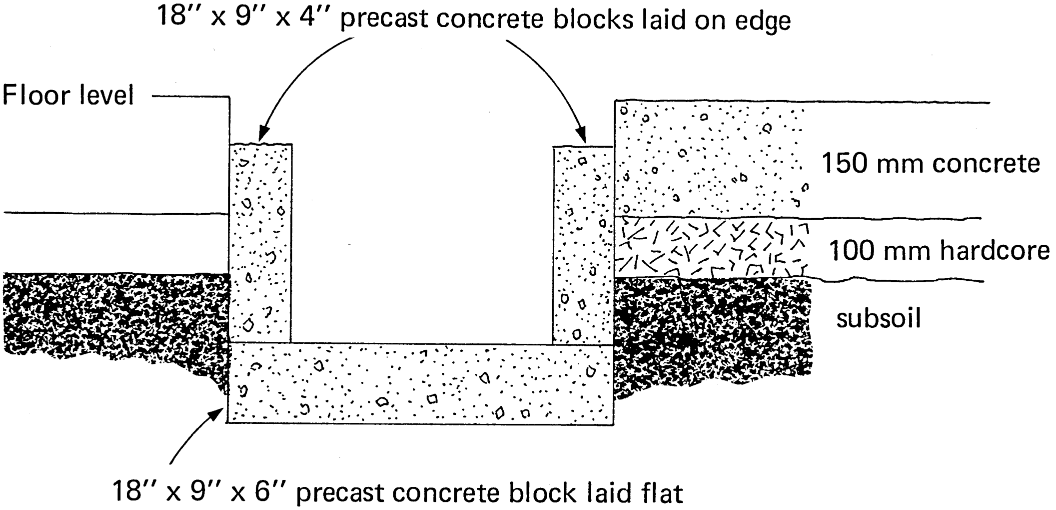


Fig. 15

Possible Change in Air Velocity Leaving Lateral Ducts

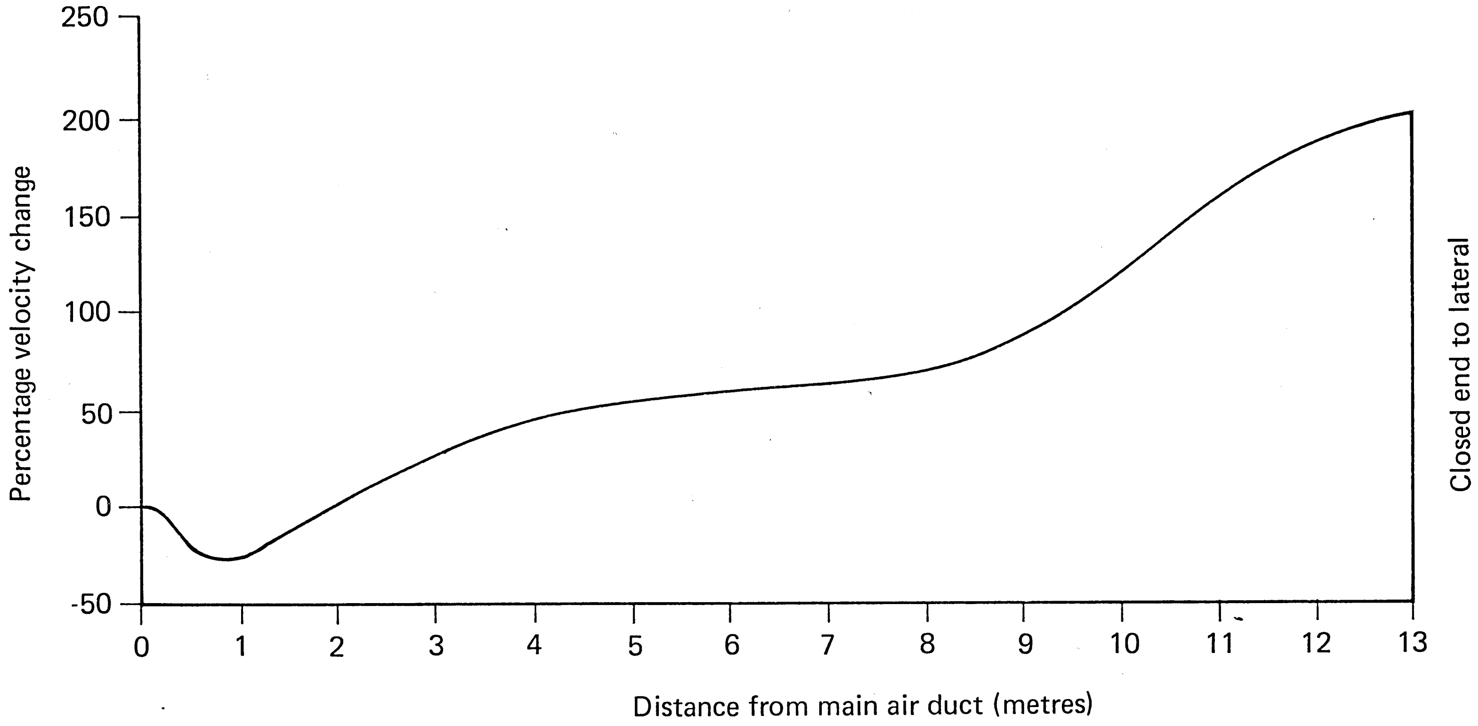


Fig. 16

Pressure Relief Vent

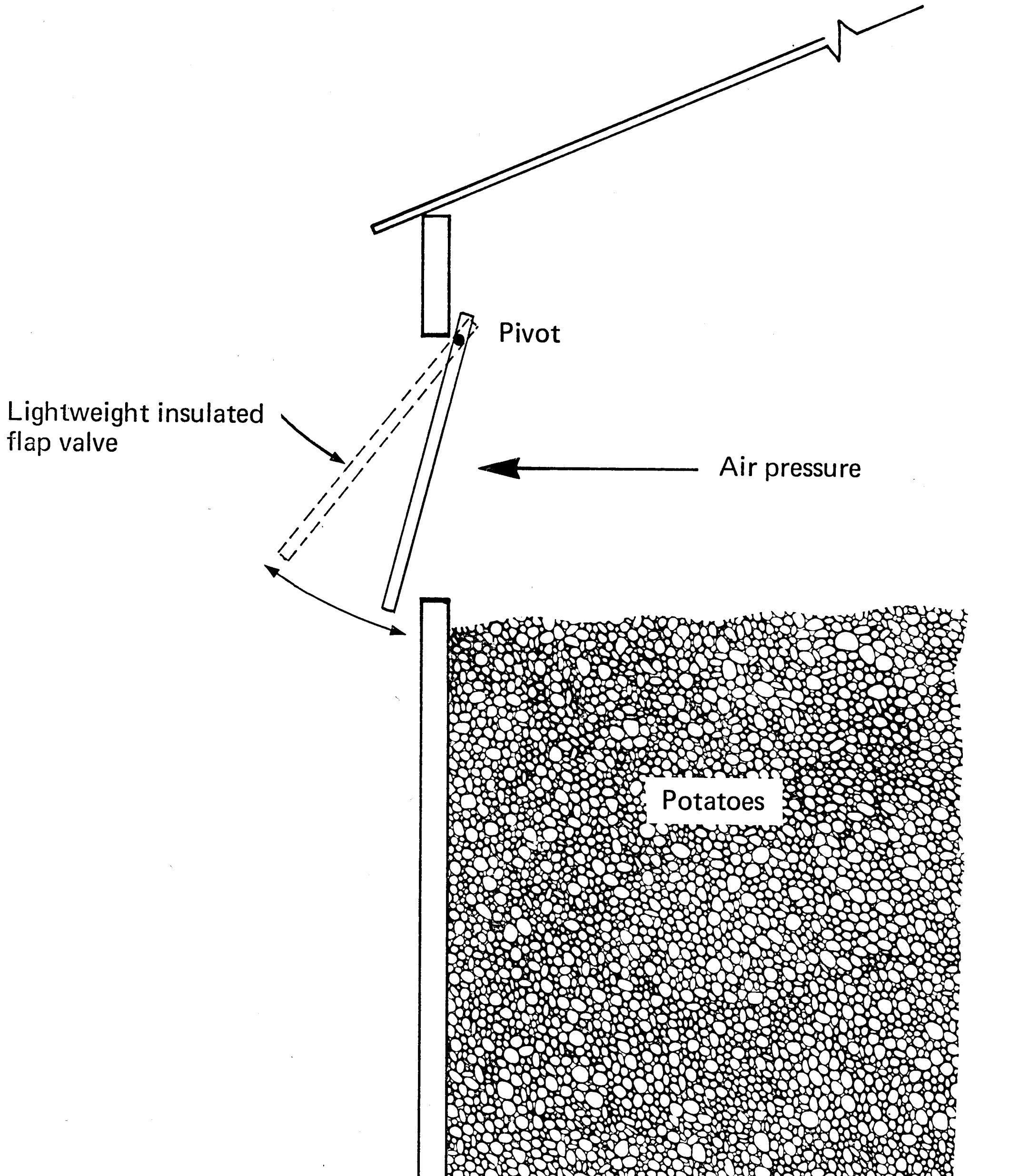
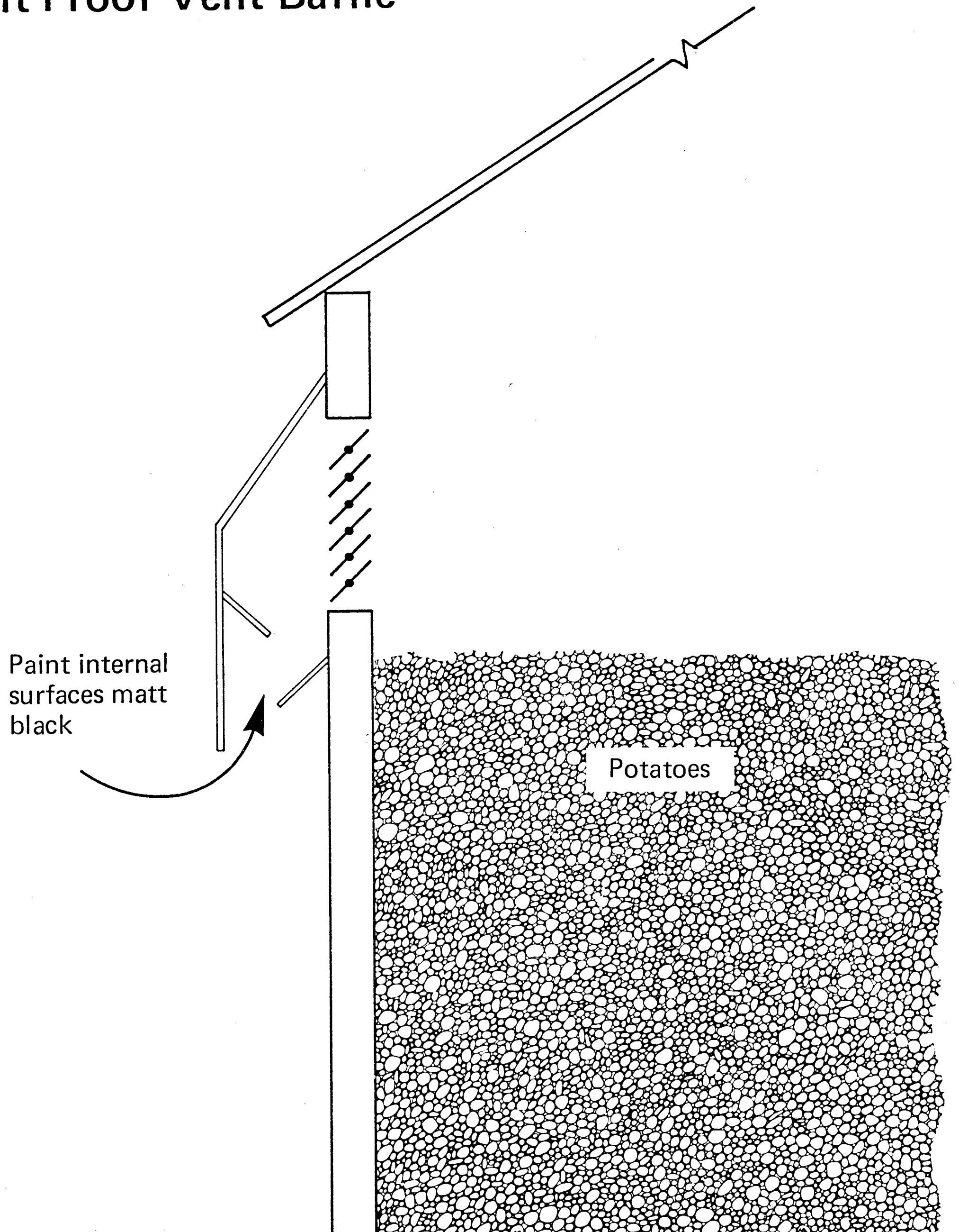


Fig. 17

Light Proof Vent Baffle



Top Ventilation

An aspect of ventilation and distribution which is frequently overlooked or skimped is the arrangement for exhausting ventilating air. In general, air velocities at discharge from the building should not exceed 4 m/s (800 ft/min) if the practice of reducing air velocity progressively through the system is to be maintained. Because the exhaust openings must be controlled, it is not practical to use an open ridge arrangement which would be desirable from a distribution point of view and it is usual to have gable end ventilators or possibly side ventilators. The total free cross sectional area of these ventilators should not be less than 0·068 m^2 per 1000 m^3/h (1·25 sq ft per 1000 cfm) of fan capacity.

Recirculation of air which is discussed in Section 9 of this report requires some modification of the preceding recommendations.

Exhaust vents must be above the level to which potatoes are loaded and as near to the apex of the building as possible. This will increase the chimney effect and aid extraction particularly with convective ventilation. If convection cooling only is to be allowed for, then a total of 0·55 m^2 (6 sq ft) of ventilator opening should be provided for every 100 tonnes of potatoes in store. Ventilators which are normally situated in the gable ends of a building need adequate free air circulation above the level of the stored crop. New buildings which have low pitched roofs $< 22°$, should have 1 m (3 ft) from the top of the level loaded potatoes to eaves height.

If automatic control of F.D. is provided it is advisable to provide automatic opening and closing of the vents. Such control may take the form of louvred shutters linked to a servo motor operated through relays from the fan starter. A simpler but equally effective arrangement is to use simple counterbalanced flap valves as in Figure 16. (Page 23).
The action of the fan coming into operation creates a pressure within the building which will actuate the valve.

If loose straw is not used on top of the crop (see Section 9) whichever ventilator arrangements are provided must be light proof; this can be done with hoods and baffles as in Figure 17. (Page 24).

Recommendations

(1) Aim for a system which connects all lateral distribution ducts to a main air duct.

(2) Avoid bends or any other restriction in the system as far as possible.

(3) Design duct dimensions on a falling velocity basis.

(4) Maximum lateral spacing 2 m (6 ft) centres.

(5) Avoid lateral lengths in excess of 18 m (60 ft).

(6) Try to use the main air duct as a partition or as part of the wall construction.

(7) Put laterals underground to ease mechanisation of loading and unloading and reduce damage.

SECTION 5

HUMIDIFICATION

Humidity and Ventilation in Relation to Weight Loss

Two factors govern the rate of water loss from potatoes—one is the amount of water which can be taken up by air surrounding the tubers before it becomes saturated; the other is the protection against water loss provided by the skin of the potatoes.

The ability of the air to absorb moisture is defined by its water-vapour pressure deficit measured in millibars (mbar). The following table shows the relationship between relative humidity (R.H.) and vapour pressure deficits (V.P.D.) of air at various temperatures.

Table 2

Temperature (°C)	0	5	10	15	20	25
Temperature (°F)	32	41	50	59	68	77
Saturated water vapour pressure (mbar)	6·08	8·65	12·16	16·86	23·09	31·28
Water V.P.D. (mbar)	Relative humidity (%)					
1	83	89	92	94	95	97
2	67	77	83	88	92	94
3	50	65	75	83	87	90
4	34	54	67	77	83	87
5	13	42	59	70	78	84

Relative humidity is the ratio of the vapour pressure in a given space compared with the vapour pressure at saturation, the temperature of the mixture being the same in both cases.

The weight loss from potatoes per mbar V.P.D. is, except in the case of very low air flows, unrelated to ventilation rate. For mature, unsprouted tubers, the percentage weight loss per month after curing has been found to be 0·6% per 1 mbar V.P.D. for all ventilation rates in excess of 12 m^3/tonne/h (7cfm/ton). At 10 m^3/tonne/h (6 cfm/ton) the loss would be 0·5% per 1 mbar V.P.D.

The threshold level below which reduced rates of ventilation cut evaporative losses rises from 12 m^3/tonne/h (7 cfm/ton) for mature, unsprouted tubers, to 40 and 60 m^3/tonne/h (24-35 cfm/ton) for newly harvested and well sprouted potatoes respectively. Rates of ventilation which are regarded as of practical use for cooling or drying mature, unsprouted potatoes exceed the above critical values. It follows that if ventilation is employed, the resultant increase in loss of weight will depend upon the duration of the period of ventilation and the V.P.D. but will be independent of the rate of ventilation during these periods.

Accepting that ventilation is necessary for temperature control, any attempt to reduce evaporative losses must be directed to reducing the V.P.D. of the ventilating air and/or to reducing the periods of ventilation (rapid ventilation for a shorter time).

A reduction in V.P.D. requires the addition of water vapour to the ventilating air which may be carried out with one of two methods.

Steam Injection

This is by far the most effective way of adding water vapour to the air stream, steam being produced from a boiler and injected into the air stream under the control of a humidistat.

Example 5.1: In a 500 tonne store being ventilated at 67 m^3/tonne/h (40 cfm/ton) the total airflow is 33 500 m^3/h (20 000 cfm). Steam is added to maintain a V.P.D. of 0·5 mbar (95% R.H. at an air temperature of 5°C), when ambient air has a V.P.D. of 3 mbar (65% R.H. at an air temperature of 5°C).

Air @ 3 mbar V.P.D. contains 55 grains of moisture/kg
Air @ 0·5 mbar V.P.D. contains 81 grains of moisture/kg.
Therefore water required to be added=81—55=26 grains/kg of air
33 500×1·2=40 200 kg of air per hour, where weight of dry air is 1·2 kg/m³.
The amount of water to be added per hour

$$=40\,200\times26 \text{ grains}$$
$$=\frac{40\,200\times26}{15\,400} \quad (15\,400 \text{ grains}=1 \text{ kg})$$
$$=68 \text{ kg of water per hour}$$

To raise 68 kg of water from a temperature of 5°C to boiling point 100°C would require an electrical immersion heater loading of
$68\times(100-5)\times4187\times2{\cdot}78\times10^{-7}=6{\cdot}68$ kW
where 4·187 calories=1 joule
1 joule $=2{\cdot}78\times10^{-7}$ kWh

Add for latent heat of vaporization to produce steam at 526·68 Calories/kg
$$68\times526{\cdot}68\times4187\times2{\cdot}78\times10^{-7}=41{\cdot}63 \text{ kW}$$

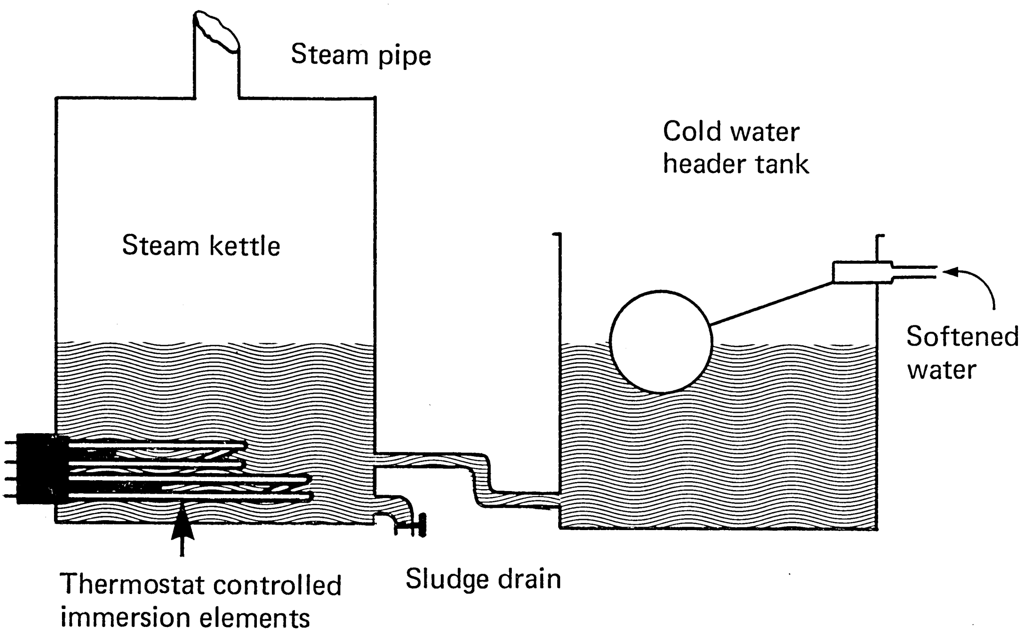
A total energy input of 48 kWh.

It is obvious from these calculations that except under conditions of recirculation of air as when refrigeration is applied, that the addition of steam is grossly uneconomic. During recirculation both the airflow and the quantity of the water for make-up would be greatly reduced allowing for a much smaller and possibly economic loading.

Should steam be required for humidification, a very simple and effective apparatus can be constructed as in Figure 18. Water supply to the steam generator is maintained via a separate cold

Fig. 18

Simple Steam Kettle



water tank and ball valve. A separate tank having a free outlet to the air stream should contain only a small quantity of water and is heated by two immersion elements. The primary and smaller element is thermostatically controlled to maintain the water at a temperature of approximately 90°C (195°F), the second larger element is controlled by the humidistat and supplies the energy required for the latent heat of vaporization. Using twin elements in this fashion enables a quick response to humidistat signals. The humidistat should be positioned in the main air duct, well down stream from the steam injector.

Water Injection

If the magnitude of the heat energy input precludes the use of steam, the alternative is to use cold water in an atomised form injected into the air stream. To be effective, any water applied must be vaporized and the latent heat of vaporization is taken from the ventilating air. Where water is available at wet-bulb temperature, air will absorb water vapour and in converting water to vapour, the air dry-bulb temperature tends to cool to the wet-bulb temperature and the degree of cooling and vaporization is dependent on the efficiency of the air-water mixer.

The cooling effect obtained from this form of humidification is most useful when the primary object of ventilation is cooling.

In the previous example, ambient air at 5°C (41°F) and 69% R.H. would have a wet-bulb temperature of 2·8°C (37°F) and if fully humidified with water atomised into the airstream, the cooling effect would be 2·2°C (5°—2·8°C). In practice, complete mixing is not achieved and dry-bulb temperature is never reduced fully to wet-bulb temperature although up to 90% could be obtained. Similarly, if the water temperature is higher than the wet-bulb temperature, it will have the effect of partially raising the wet-bulb temperature, reducing the difference and, hence, decreasing the evaporative cooling effect. Although this latter state of affairs can be expected with an average mains water temperature of 7°C (45°F), even a 0·5°C or 1°C cooling effect can be most useful.

To revert to the humidifying effect of adding water, in example 5.1, 68 kg of water per hour had to be added to the air flow. In practice, this is possibly best done by atomising the water through standard horticultural misting nozzles.

Example 5.2: One misting nozzle will supply approximately 20 kg of water per hour.

Therefore, number of nozzles required:

$$\frac{68}{20}=4 \text{ (rounded up)}$$

As the process of evaporation is very inefficient, up to 6 nozzles can be provided. Water must be applied under pressure with a pump 6·2-7·5 bar (90-110 lb/in^2) for best atomization (atomization provides a large surface area for evaporation) and the system is again controlled by a humidistat on a simple on/off basis. Because there will be free water, drainage facilities in the main air duct will be needed and care must be exercised to ensure that free water particles entrained into the air stream are not carried into the stored crop. In practice, this means a main air duct of suitable dimensions to keep air velocities low and enable water particles to drop out and give time for vaporization to occur before the air stream enters the lateral ducts.

The preceding examples in this section on humidification obviously apply only to the stated cases and the sizing of equipment will vary as all the factors can vary. As a guideline, with a relative humidity of 98% at 5°C (41°F), no water will be lost from tubers as the equilibrium condition is reached. This figure is, however, dangerously close to condensation conditions and a more safer make-up target would be 90% R.H. Over the length of the potato storage period, the relative humidity of the ambient air will fluctuate markedly but, as a guide, one could use a base figure of 70% R.H., e.g. a deficit of 2·5 mbar vapour pressure at 5°C (41°F). The problem of free water deposition on the potatoes arising through condensation or droplet carry-over are such that considerable care must be exercised if serious losses are to be avoided. Accuracy in monitoring and control equipment is essential and the intending user should take particular note of Section 10.

The irrelevance of relative humidity unless linked to a known temperature as a means of determining weight loss during ventilation has been shown. Weight loss can be expressed in terms of prevailing V.P.D. however. For example, the rate of water loss is 0·17% per week per millibar V.P.D. for mature potatoes, 0·68% per week per millibar V.P.D. for immature damaged potatoes. Every 1% by weight of sprouts increases potential loss by about 0·08% per week/mbar V.P.D.

Table 3

Pressure in millibars of the water vapour contained in saturated air

(1mbar=100 N/m²)

Temp °C	0	1	2	3	4	5	6	7	8	9
0	6·08	6·53	7·01	7·53	8·07	8·65	9·27	9·93	10·63	11·37
10	12·16	12·99	13·87	14·81	15·81	16·86	17·97	19·15	20·39	21·70
20	23·09	24·56	26·11	27·74	29·46	31·28	33·19	35·21	37·33	39·56

Maximum acceptable weight loss in potatoes is 10%. Quality and appearance are increasingly affected after 5% loss.

Recommendations

Use water injection system providing

(1) Management is above average

(2) Crop is to be stored long term

(3) The equipment has a maximum rating of 0·25 kg (0·5 lb) of water/tonne/h

(4) An accurate control system is installed (see Section 10)

(5) Drainage facilities exist in the main air duct

(6) The anticipated return from quality and weight premium justifies the risk.

SECTION 6

HEATING

The preceding sections of this report have all stressed the need to cool potatoes to provide the optimum storage condition. To discuss the need to apply heat would appear to be an anachronism. There are, however, certain conditions under which it may be appropriate.

(1) When cooling is required in store but ambient air temperatures are below zero.

(2) To maintain store temperature during "curing".

(3) To prevent frost damage in store.

(4) To eliminate condensation in roof void of store and in the top layers of potatoes.

(5) To raise tuber temperature prior to handling.

This report does not intend to discuss the requirements of chitting or seed storage buildings except to say that the small volumes of seed involved on the average holding will require supplementary heating to maintain temperature control.

If proportional blending of recirculated and ambient air is not possible then the use of heaters to raise sub-zero temperature ventilating air to a safe figure, is an acceptable alternative (albeit with higher operating costs). Most installations would be based on propane gas or electrical heating elements sited either immediately upstream or downstream of the fan. Because of the considerable volumes of air being used during F.D. it is not usually considered practical or economic to raise the air temperature by more than 2·5°C (5°F).

Example 6.1: In a building where the actual storage temperature is 7°C (45°F) and the desired holding temperature is 4°C (39°F) ventilation is required. If the temperature of the ambient air was —3°C (27°F) it would be necessary to heat the ventilating air to 0°C (32°F) (the minimum considered safe to prevent frost damage). Assume 500 tonne store with ventilation rate of 67 m^3/tonne/h,

$$500 \times 67 \times 1{\cdot}2 \times 2326 \times 2{\cdot}78 \times 10^{-7} = 25{\cdot}99 \text{ kW},$$

where:
weight of dry air is 1·2 kg/m^3 (0·075 lb/ft^3)
specific heat of air is 2326 J/kg (0·241 Btu/lb)

In the example approximately 25 kW of supplementary heat would be needed although in practice the figure could be lower, due to the rise in temperature of the air going through the fan from the motor heat and imparted kinetic energy. It would be advisable to arrange for the heating to be applied in steps, e.g. 3×10 kW heater banks in an electric installation to

i) reduce running costs

ii) to improve cooling efficiency.

The technique is only really applicable during prolonged periods of sub-zero temperatures (say $>$7 days) and would not generally be necessary in England and Wales.

Immediately after loading, the higher the temperature the more rapid periderm formation is and curing temperatures in the band 10°-15°C (50°-60°F) for 18-10 days are recommended. Closing the store and restricting ventilation is the accepted practice to achieve these temperatures during curing. However, prolonged loading of the store and/or adverse field and climatic conditions at the time of store loading could make heat applied selectively to the potatoes through the ventilation system attractive. If such an arrangement were to be at all economic, bearing in mind the high temperature required, then recirculation of the air is necessary.

Heating to prevent frost damage in stores, through heat loss from the fabric of the buildings should not be necessary in the U.K. providing the structure has a U value of 1·36 W/m^2/deg C (0·2 Btu/h/ft^2/deg F) (see Sutton Bridge Report No. 6, Part I).

By applying a small temperature increase to the air space above potatoes in a store, condensation can be prevented and in the case of a light-proof structure the need to straw the top of the stack avoided. One of the functions of this straw is to absorb the condensation formed as a result of warm moist air rising up through the stack and meeting a layer of cooler air in the apex of the building (see Sutton Bridge Report No. 6, Part I, Section 10). Raising the temperature of the air in the roof void by up to 2·5°C (5°F) is one solution, recirculation (see Section 9) is another. It is, of course, an anathma to be raising temperatures in even a localised part of the store when overall it is a general requirement to reduce temperature. Nevertheless, this technique has been successfully used to store processing potatoes without straw but it does call for the application of heat in a rather different manner from the previous cases.

Obviously the main ventilation system cannot be used to apply heat which is required in a localised place. Small fan heaters suitably sited under the eaves and at the gable ends of the building are recommended. Their operation is under thermostatic control and the calculation of total heat input is based on the heat loss × temperature lift required. Work at Sutton Bridge has shown that recirculation of store air is equally effective in eliminating the need for strawing and would usually be more economic to install and operate.

It has been shown that there is a direct relationship between tuber temperature and susceptibility to damage while handling. It is recommended that tuber temperature should be 7·5-10°C (45°-50°F) before handling for grading is attempted. Normal practice would be to shut up the store and cease ventilation some 14-21 days before anticipated movement is to take place (temperature rise in an unventilated stack is of the order of 0·025°C (0·5°F) per 24 hours. Prolonged marketing makes such management control difficult although individual lateral ducts may be closed to give some localised relief when ventilation of the main bulk of the store is taking place. The ability to recirculate heated air could be a useful management aid.

Heat Source

Of the three natural processes of heat transfer, radiation, conduction and convection, only convection is of practical significance. The manner in which heat may be supplied is either

(1) Direct—fired air heater
(2) Direct—fired heat exchanger
(3) Indirect heating system
(4) Electric heater battery

(1) and (4) are the most common heating systems found on farms and the choice of fuel is usually limited to

(a) electricity
(b) Bottled Gas (Propane/Butane)
(c) Fuel Oil (Heavy/Distillate)

Table 4

Fuel	Calorific Value	Heating Efficiency
Electricity	3600 kJ/kWh (3412 Btu/kWh)	98%
Propane	49 000 kJ/kg (21 000 Btu/lb)	85%
Oil	41 000-46 000 kJ/kg (18 000-20 000 Btu/lb	75-80%

All three fuels offer graduated control of heat input and starting can be made automatic, the choice will largely depend upon availability, capital and operating costs. Bearing in mind the very limited need for the application of heat in a potato store, electricity is likely to be the first choice.

1 kilocalorie (kcal) = Heat required to raise the temperature of approximately 3·5m^3 of air by 1 deg C

1 British Thermal Unit (Btu) = Heat required to raise the temperature of approximately 56 ft^3 of air by 1 deg F.

(1 kilocalorie = 4·187 kJ)

An air heater of 1 kW capacity will raise the temperature of an air flow of 3000 m^3/h by approximately 1 deg C or an air flow of 3000 cfm by approximately 1 deg F.

Recommendations

(1) Adopt recirculation techniques in preference to air heating

(2) Apply the absolute minimum of heat at all times to maintain vapour pressure and reduce weight loss.

SECTION 7

LIGHTING

Potato storage buildings require light only for operational purposes, within the building. Natural daylight usually via roof-lights will cause greening of the crop if it is not liberally covered with straw (450-600 mm). Where unstrawed potatoes are going to be stored a light-proof building with artificial lighting is essential.

Illumination Levels

The code of the Illuminating Engineering Society gives a value of 50 lux as being appropriate for potato stores. This value, which should be regarded as a minimum will be sufficient to enable movement in the building without hazard and enable operations such as cursory inspection, loading, unloading, etc. to be performed satisfactorily. Where grading takes place in the building such a value is *not* adequate. The inspection table of a grader should have an illumination level of 150-200 lux and the general area in the immediate vicinity of the riddle 100 lux.

Many graders now have provision for tubular fluorescent fittings directly above the roller table or picking belt and this is obviously an economic way of providing a high level of light intensity where it is required. However, should the contrast between the table and its immediate surrounds be too great, operators will have difficulty in adjusting or accommodating their eyes whenever they raise them from their work. This is one of the reasons for a comparatively high level of illumination in the working area around the grader.

Where at all possible grading should take place at a fixed site either within or without the store and potatoes brought to it rather than visa versa. In this way an adequate level of illumination can be provided without the expense of installing it over the entire store.

Light Source

Most farm lighting is by either tungsten filament, colour-corrected mercury or tubular fluorescent. A symmetrical overall distribution of dispersive fittings is usually appropriate, tubular fluorescent lamps usually being the first choice.

In a potato store any type of fitting is best secured direct to the rafters or purlins of the building. Pendant or catenary suspensions are always subject to inadvertent damage by elevators etc. Arrange for sectional switching so that $\frac{1}{3}$ or perhaps $\frac{1}{4}$ of the store only may be illuminated at any one time (advantageous at the time of store unloading).

Because a potato store is likely to occupy a considerable surface area the wiring costs for a symmetrical lighting system can be high. There is a tendency to use, quite successfully, a limited number of floodlighting projectors usually placed high up (5 m (16 ft) plus) in the gable ends of the building. Fittings are usually high-wattage tungsten halogen and a gable end mounting (both ends) is adequate for most buildings not more than 30 m $\times$ 15 m (100 ft $\times$ 50 ft).

It should be remembered that one of the aims of management is the maintenance of a high relative humidity. The attendant danger of condensation means that drip proof corrosion resistant fittings must always be used and that the whole installation must be earthed. Visible radiation (light) is composed of a number of colours ranging from violet (4000 Angstrom wavelength) through blue, green, yellow, orange to red (7000 Angstrom). Investigations have been carried out to determine if any part of this spectrum will provide adequate overall illumination in store without the risk of tuber greening. In fact, anything within the range 4000-7000 Angstrom will cause greening although the incidence is lowest at about 5000-5500 Angstrom (green). The fluorescent tube which is invariably used to provide light directly over the grading table is available in varying shades of white. Of the six shades recognised in B.S. 1270, White, Daylight, Natural and Colour Matching are the most suitable.

Recommendations

(1) Install 50 lux over entire store area

(2) Provide localised higher levels where needed, e.g. grading area.

SECTION 8

BOX STORAGE

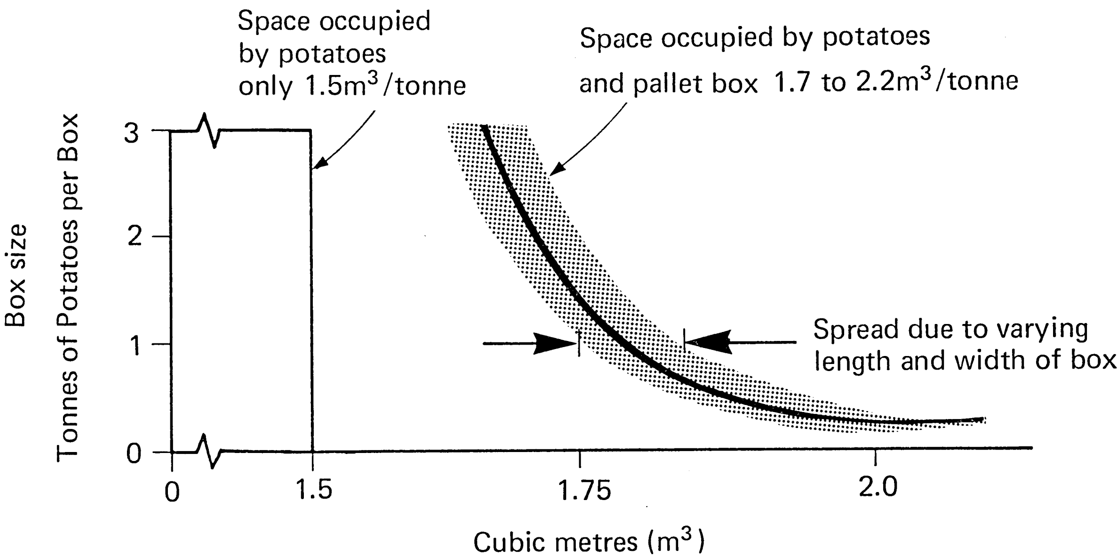
Throughout the earlier sections of this Report reference has always been to the requirements of bulk stored potatoes. For potatoes stored in pallet boxes some amendments have to be made especially under the heading of air distribution.

Forced Draught Ventilation

A pallet box behaves as a miniature unventilated bulk store and therefore heat will be taken away from the potatoes by convective air currents within the boxes until such time as the potatoes and the store air reach a state of thermodynamic equilibrium. Because the dimensions of the boxes are relatively small the temperature of the potatoes will rapidly approach that of the store air and it has been calculated that they will have a maximum temperature of 1-2°C (2-4°F) above that of the store air.

Potatoes in boxes require 1·4 m³/tonne (50 ft³) and the boxes of potatoes in a store occupy about 2·4 m³/tonne (85 ft³), the difference between volumes being mainly air space between boxes, with approximately 60% more free air space than in a comparable bulk stack.

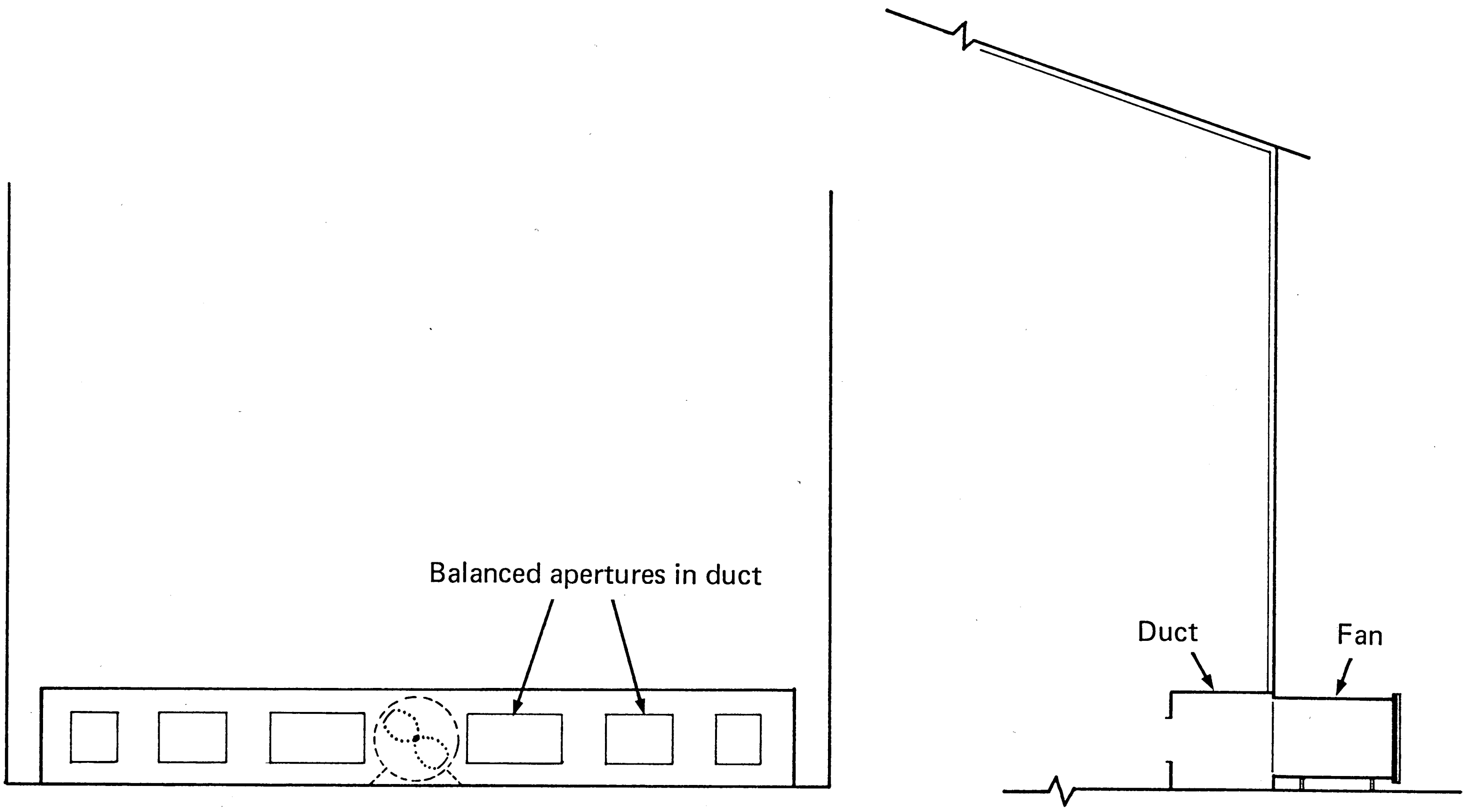
Fig. 19 Minimum Cubic Metres of Storage Needed Per Tonne of Potatoes (Does not include aisle space and ceiling clearance)



It would seem reasonable to assume therefore that providing this warm air can be removed from around the boxes and replaced with fresh air at the right temperature, nothing further is necessary for temperature control. This hypothesis is in fact practised by the majority of growers storing in boxes, reliance being placed upon the establishment of convective air currents and the natural ventilation of the building. Forced draught ventilation to provide controlled *air changes* in the store will give more precise control over temperature and will shorten the time taken whenever a temperature reduction needs to be effected. Rates of ventilation for bulk stores were assessed bearing in mind limits on acceptable stack temperature gradients as well as cooling rates in relation to climatic conditions. A lower rate of ventilation in a box store is acceptable because of the reduced effect of depth of stacking on temperature gradients. 35 m³/tonne/h (20 cfm/ton) will usually be adequate.

Fig. 20

Simple Balanced Distribution Duct for Box Stores

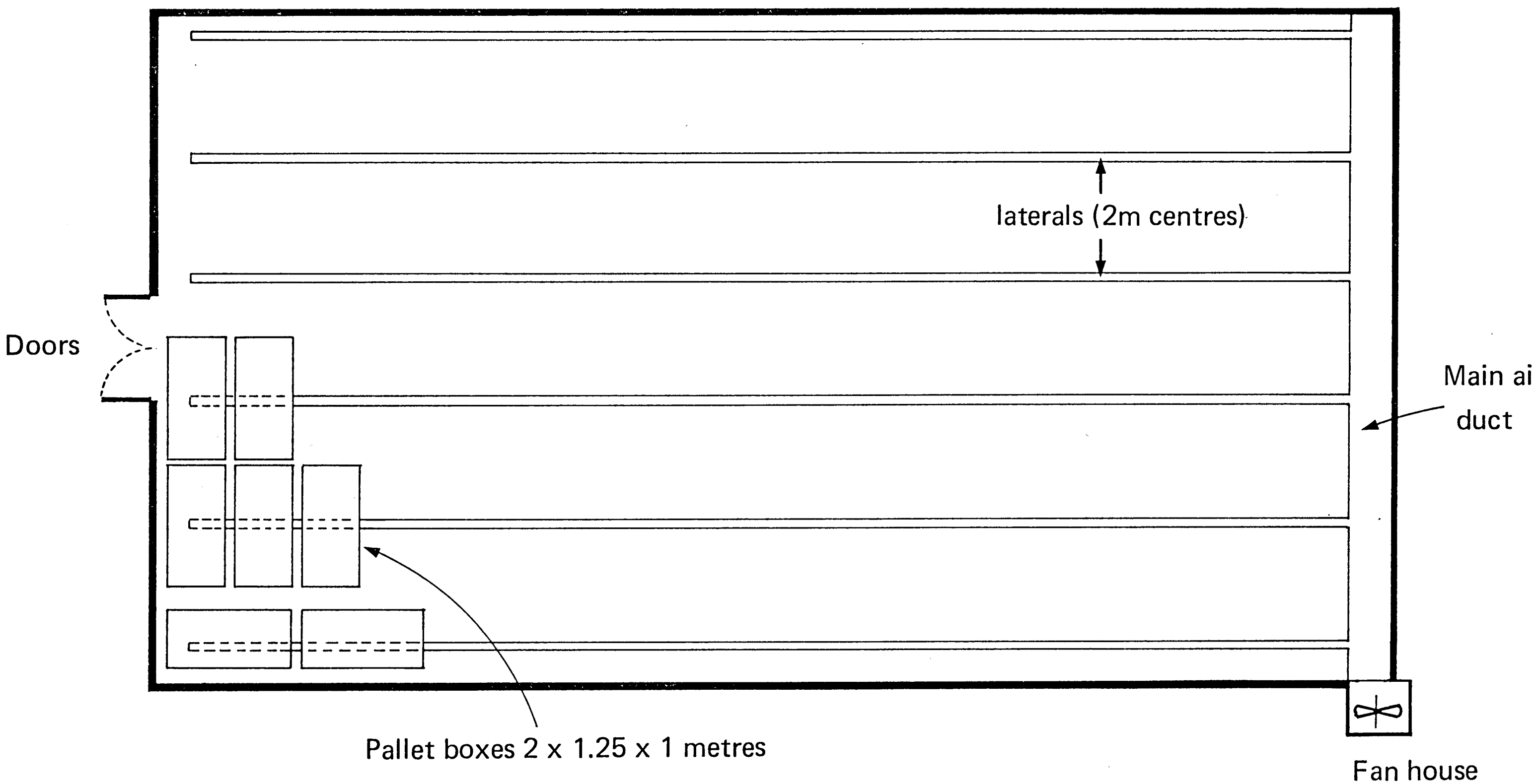


Front elevation

End elevation

Fig. 21

Underfloor Lateral Ducts in a Box Store



Air Distribution

Notably on the Continent and in the U.S.A. the application of F.D. is carried out in such a way that the air is forced through individual boxes or blocks of boxes. Unless this is done air will not pass through the potatoes until the resistance caused by the speed of the air in the spaces between boxes equals the resistance of the potatoes to air flowing through them. Distribution to individual boxes or even columns of boxes is an expensive ducting exercise which may be dispensed within a commercial store where recordings have indicated that the temperature of the produce rapidly approaches to within 1-2°C (2-4°F) of that of the store air. This is because the minimum dimensions of a box are relatively small and because warm air can move out through the six faces of a box.

Some attempt to control the overall pattern of air movement is advisable. In single fan installations one entry point can cause localised problems due to high air velocities and some preliminary distribution across one face of the store is helpful.

In Figure 20 a simple balanced cross main air duct at ground level ensures that air travelling the length of the store does so across its full width before being exhausted at the far end.

In Figure 21 a system of laterals (underground) is connected to the main duct. Each lateral is positioned underneath a column of boxes.

Figure 22, which is a side elevation indicates the principle of bringing in cold air at low level and exhausting at high level.

Some general principles for single fan ventilation systems in box stores follow:

(1) Use gaps between boxes and alleyways etc. to guide air in the desired direction.

(2) Exhaust at high level whether fresh air is brought in at high or low level.

(3) Where a number of exhaust points are used they should be equidistant from the inlet and as in the case of a single exhaust as far from the inlet as possible.

(4) Make the system as simple and inexpensive as possible to offset the cost of boxes.

Fig. 22

Relative Position of Air Inlet to Exhaust Point

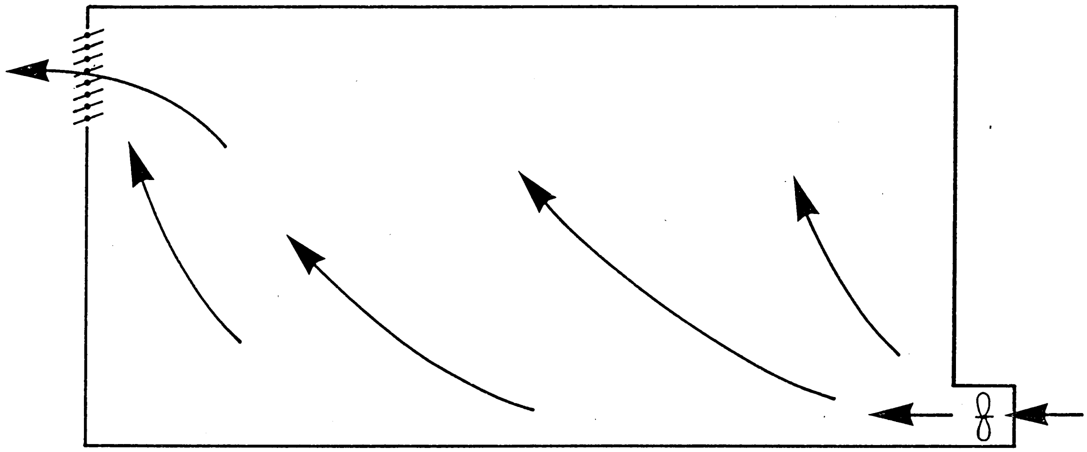
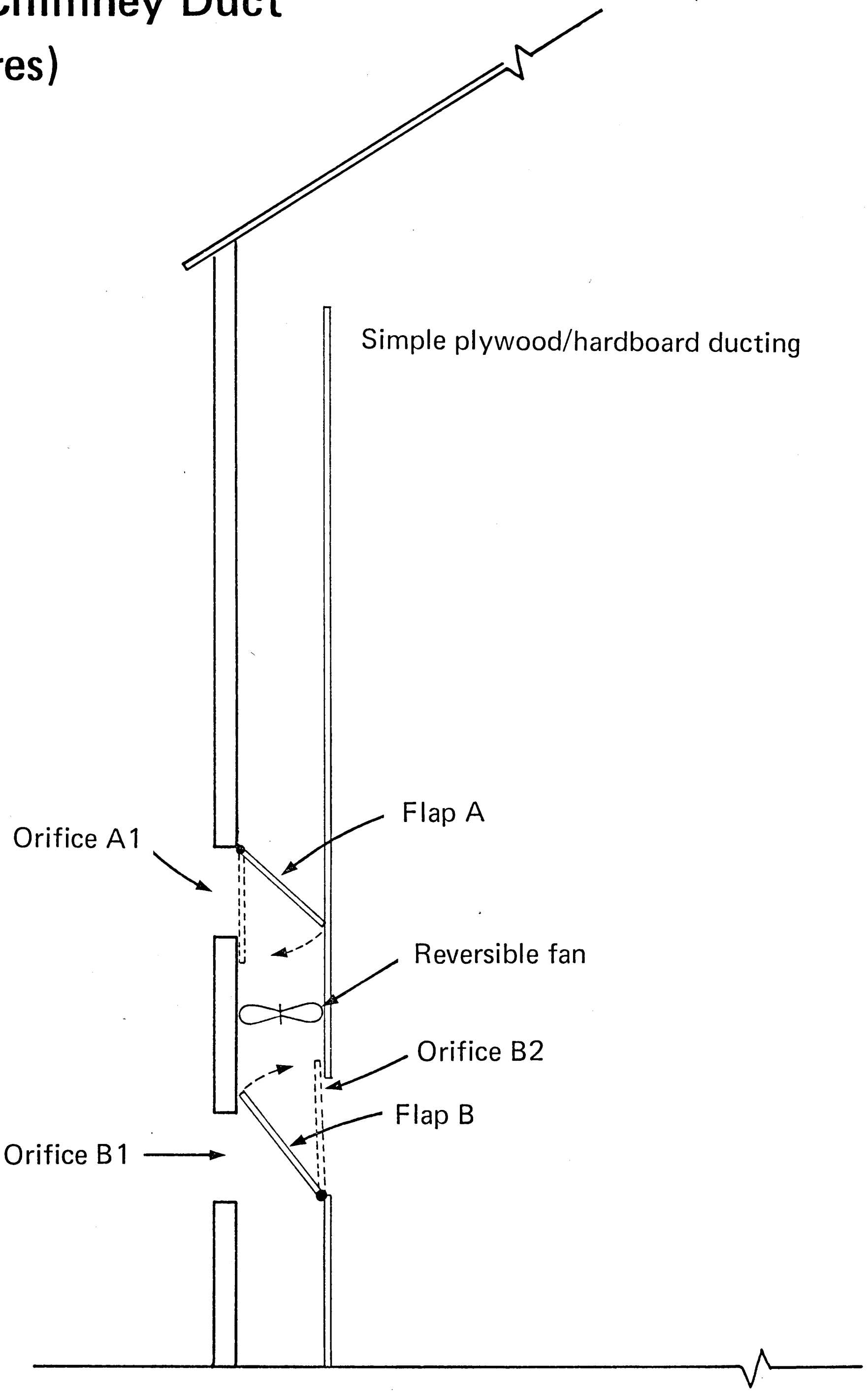


Fig. 23

Vertical Chimney Duct (Box Stores)



An alternative to the single fan installation is the use of a number of fans placed around the perimeter of the store, their sum air flow equalling the total requirements of the building. If a simple vertical chimney duct as in Figure 23 is provided and a reversible fan is installed, great flexibility is achieved.

Example 8.1: With flaps A and B in solid line position and the fan blowing downwards, ambient air is drawn in at orifice A.1 and discharged into the store at low level at orifice B.2.

Example 8.2: Reversing the fan exhausts the store from low level.

Example 8.3: With flaps A and B in the dotted line position and the fan blowing upwards, ambient air is introduced at the new orifice B.1 and discharged into the store at high level from the top of the chimney.

Example 8.4: With flap A in the dotted line position and flap B in the solid line position, recirculation of air in the store is possible in two directions depending upon fan direction.

Example 8.5: With flaps A and B in dotted line position the store can be exhausted from high level.

Where two or more fans are installed in a store the overall ventilation rate may be altered by selective switching of the fans. Some localised control of ventilation in a big store is also possible with this system. An axial flow or propeller fan which is reversed will only provide 60-70% of its rated air flow and it should therefore be installed working in the direction in which it is most normally going to be used.

The selection of a fan or fans should be made on the assumption that total air flow resistance does not exceed 25 mm s.w.g. (1 in s.w.g.). If extractor fans are used in tandem with input fans, ensure that the rated air flow is complementary. To provide extractor fans which have a lesser duty than the input fans will prejudice the ventilation rate rather than assist or improve it. Extractor fans in box stores are only necessary if

(1) special control over air flow direction and volume is required.

(2) the pressure capability of the input fan or fans is insufficient. In general, extractor fans are an unnecessary duplication of cost.

Recommendations

(1) Ventilation should be at the rate of 35 m^3/tonne/h (20 cfm/ton)

(2) A simple dispersal main duct at the air entry point should be used, in single fan installations.

SECTION 9

RECIRCULATION

The ability to recirculate air within a potato store is an important management aid. In its most simple mode it has been shown that the temperature gradient in a stack, which will occur during periods of non-ventilation can be considerably reduced. At the other end of the scale, recirculation and mixing of store and ambient air can greatly improve the seasonal efficiency of a F.D. cooling system. There are other "spin off" gains that will be enumerated.

Temperature Gradient Control

Where store cooling is by ventilation with ambient air, there are obviously periods, particularly at the beginning and end of the storage season, when ventilation will not be possible because of unfavourable conditions. These conditions may extend for several days and the rapid establishment of a temperature gradient (up to 2°C per metre (1°F per foot) depth of potatoes) will give temperatures markedly above the optimum in parts of the store. Recirculation will reduce this gradient.

Table 5 illustrates the order of magnitude of recirculation rates necessary to maintain given temperature gradients.

Recirculation can be continuous whenever F.D. is stopped or it can be automatically actuated whenever a temperature gradient of, say, 2°C (4°F) is exceeded. Alternatively, it may be carried out on an intermittent basis, say, 10 minutes in every 60 minutes, timeclock controlled to be activated whenever F.D. ceases. If recirculation is practised the cardinal rule is that it must be regular. Recirculation applied to a stack which has been allowed to develop a marked temperature gradient can result in condensation in the lower layers of the crop.

Table 5

Temperature Gradient between top and bottom of stack (4 metres)	Recirculation Rate
0·6 °C (1 °F)	34·4m³/tonne/h (20 cfm/ton)
1·2 °C (2 °F)	17·2m³/tonne/h (10 cfm/ton)
1·8 °C (3 °F)	8·6m³/tonne/h (5 cfm/ton)

Air Mixing

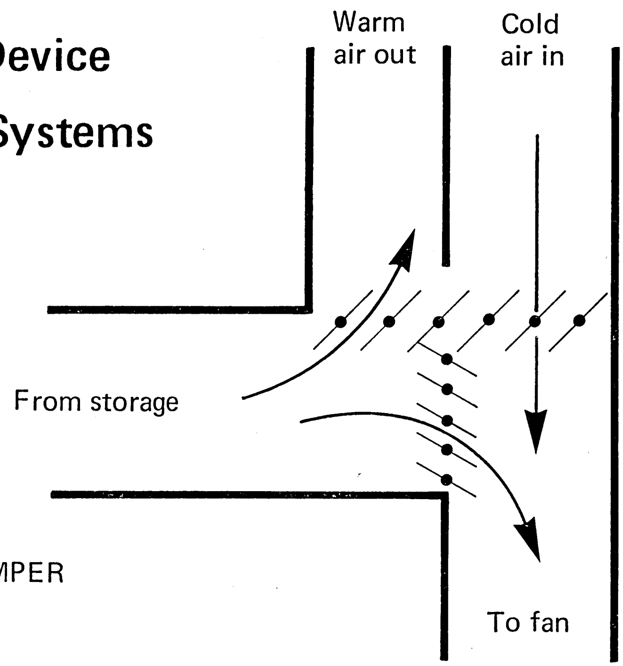
The blending of proportions of recirculated and ambient air as a means of controlling the temperature of ventilating air is widely practiced in the U.S.A. and is gaining acceptance in the U.K.

Example 9.1: The air temperature in a potato store is 7°C (45°F) and cooling is required to reduce the temperature to 5°C (41°F). The outside air temperature is —4°C (25°F); by mixing 1 part of recirculated air at 7°C (45°F) with 1 part of ambient air at —4°C (25°F) the temperature of the ventilating air can be brought to 1·5°C (35°F). It would be safe to ventilate potatoes with this latter figure whereas air at —4°C (25°F) would freeze potatoes.

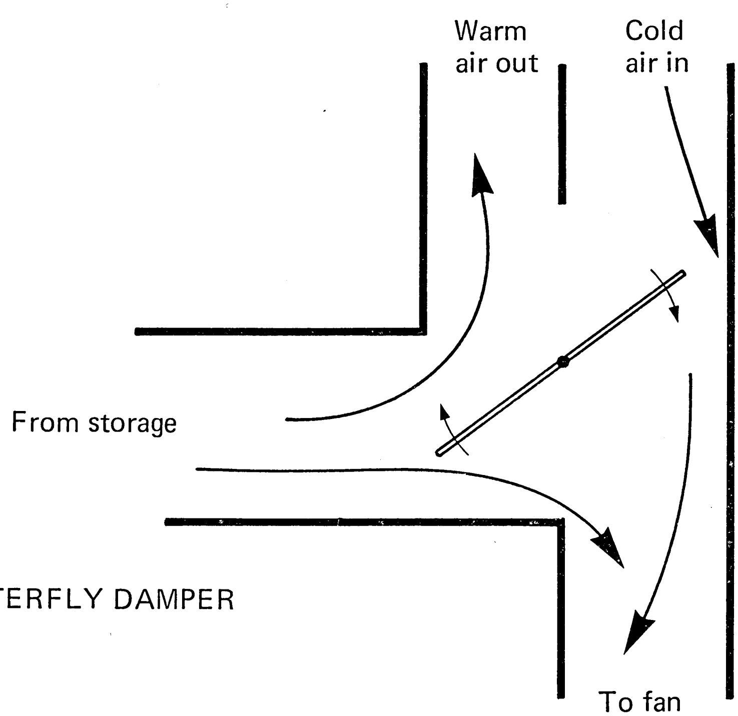
If mixing/blending facilities are not available either ventilation must be stopped altogether or air heaters used to raise the air temperature to a safe value (see Section 6). The latter will be expensive to operate and heated air will have a greater V.P.D. than recirculated air thereby increasing weight loss from the potatoes. Obviously, by varying the proportions of recirculated and ambient air any temperature between the limits of the outside air temperature and the store air temperature can be achieved. Using such a system, ventilation at any time can be realised providing ambient temperatures are below store temperatures—how much lower does not matter. In addition the provision of these facilities automatically allows recirculation to be practiced when ever ambient temperatures are above store temperatures giving the closest control of store conditions possible without recourse to refrigeration.

Fig. 24

Proportioning Device for Air Mixing Systems



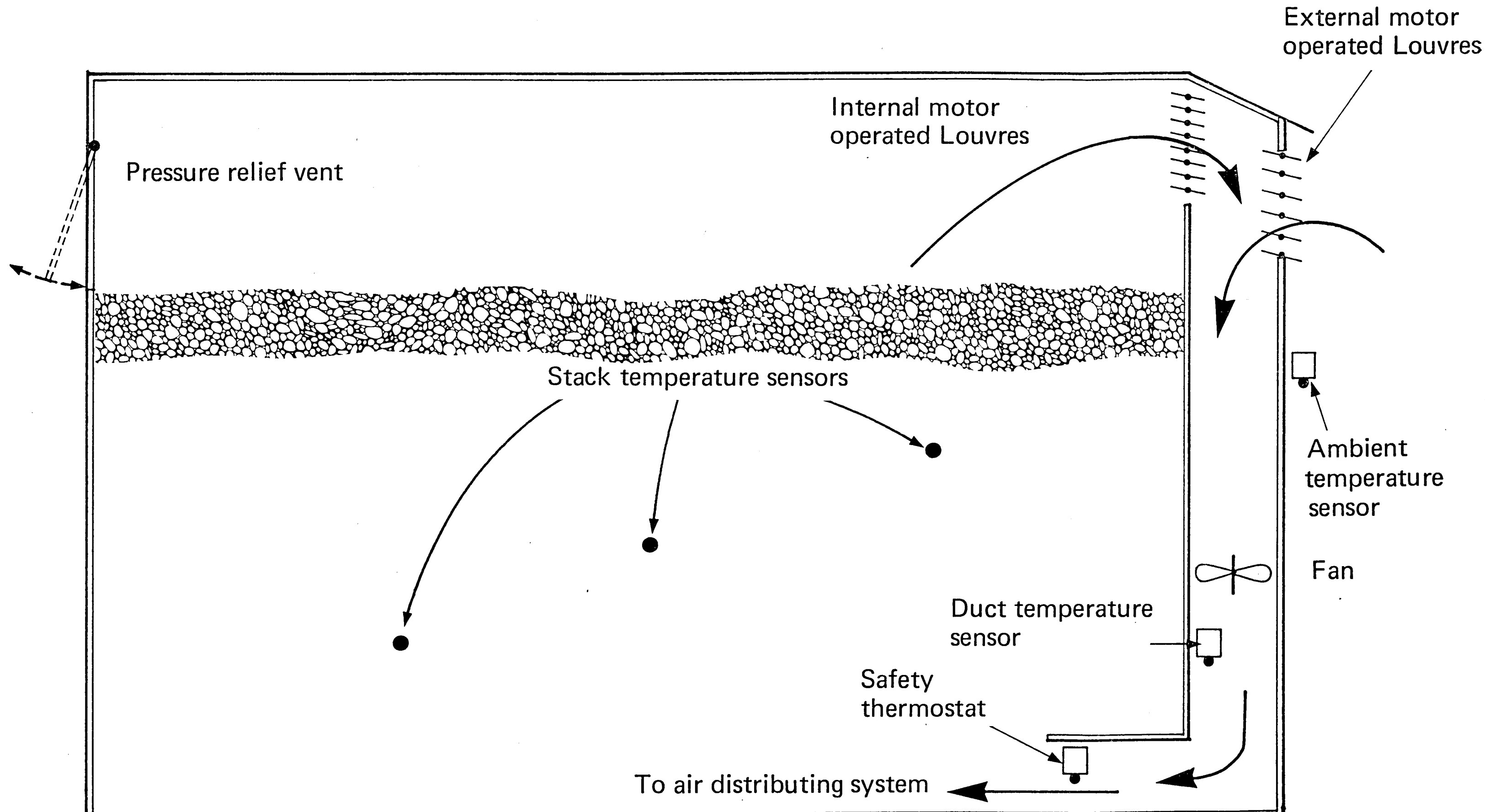
LOUVRE DAMPER



BUTTERFLY DAMPER

Fig. 25

Ambient Air Mixing/Recirculation Unit



A system for air mixing using either a single butterfly or a system of louvres is shown in Figure 24. Louvres will provide better mixing and finer control of proportions than a single butterfly vane and where the system is automatically operated by a servo motor approximately 2·5 times less torque is required. A louvre or butterfly vane system can be manually or automatically controlled. A manual system will require frequent adjustment (each day) and where temperatures hover around the danger point —2-0°C (28-32°F), frequently the case in this country, freezing damage could occur. Manual control would be based on temperature sensing in the main air duct downstream of the fan (the fan turbulence completes the mixing process). A frost guard thermostat would obviate the danger of freezing but with incorrect damper setting could lose many useful hours of ventilation. If louvre or damper operation is made automatic (proportioning control) 100% utilisation of suitable climatic conditions is assured. In principle the operation would take the following steps:

(1) The crop temperature is sensed and compared with a pre-set required holding temperature.

(2) If it exceeds the value of the holding temperature and the current ambient temperature, which is also sensed, the fan will be started.

(3) A duct temperature senser pre-set to the minimum desired ventilating temperature controls the proportioning servo motor which is linked to the store and ambient louvres. Louvres rotate to provide the correct proportions of recirculated and ambient air.

(4) A safety frost guard thermostat is set at a figure below the duct temperature senser and will close down the system should ventilating air fall to this value (this condition would only apply if some component failed).

All the preceding steps are continuous.

Figure 25 is a more detailed illustration of the arrangement in Figure 24 with the important qualification that a pressure relief vent is incorporated into the store structure simplifying considerably the louvre system.

Condensation

Traditionally stores have always had gable end vents which are usually open to the weather. During periods of non-ventilation wind blows through the store from these vents removing the warm air rising from the stack as a result of convective ventilation. Warm humid air rising from the stack will meet this boundary layer of cooler air above the potatoes and condensation is likely to occur on the fabric of the building and in the top 400-500 mm of potatoes. Even if the roof of the store is insulated, condensation will still occur because the inner face of the insulation has been cooled by the cold air entering at the gable ends. In this situation of open vents, condensation conditions can be produced in the first few minutes of F.D., particularly when low rates are used, before the cold air in the apex of the building is expelled by pressurisation. The classic remedy for this situation is to apply 300-600 mm of straw to the exposed surfaces of the potatoes lifting the condensation layer above them and eliminating the damaging effects of drip. Applying and removing straw is time consuming and during riddling it is a considerable nuisance.

Providing a sealed building, with no free ventilation of the space above the potatoes, will prevent warm air being removed by cold ambient air. So long as the roof is insulated to a value of 0.56 W/m^2 deg C (0·1 Btu/ft^2/h/degF) then condensation conditions are unlikely to occur in this country (see Sutton Bridge Report No. 6, Part I). The need for straw on the top of the potatoes is eliminated because the whole of the store environment is a stable condition with only minor variations in temperature and humidity between the potatoes and other parts of the building. Recirculation either constantly or intermittently ensures that this condition remains so.

Free gable end ventilation is not compatible with recirculation in such a system; pressure relief vents are the best means of exhausting cooling air.

Other benefits from recirculation include

i) providing air movement and reducing temperature differentials during curing, whilst maintaining a low vapour pressure deficit.

ii) dispersing chemicals such as sprout suppressants, fungicides and bactericides.

Fig. 26

Recirculation Using Secondary Fan System

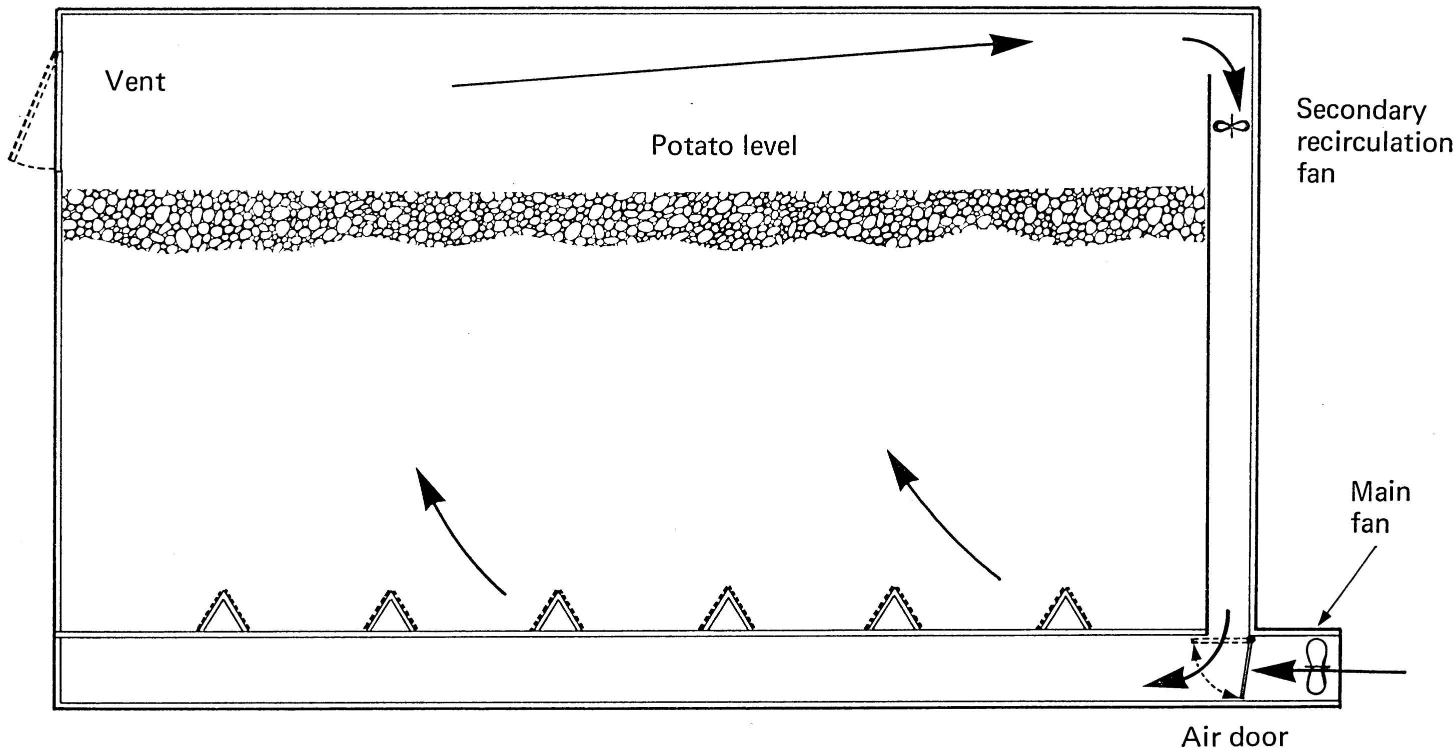
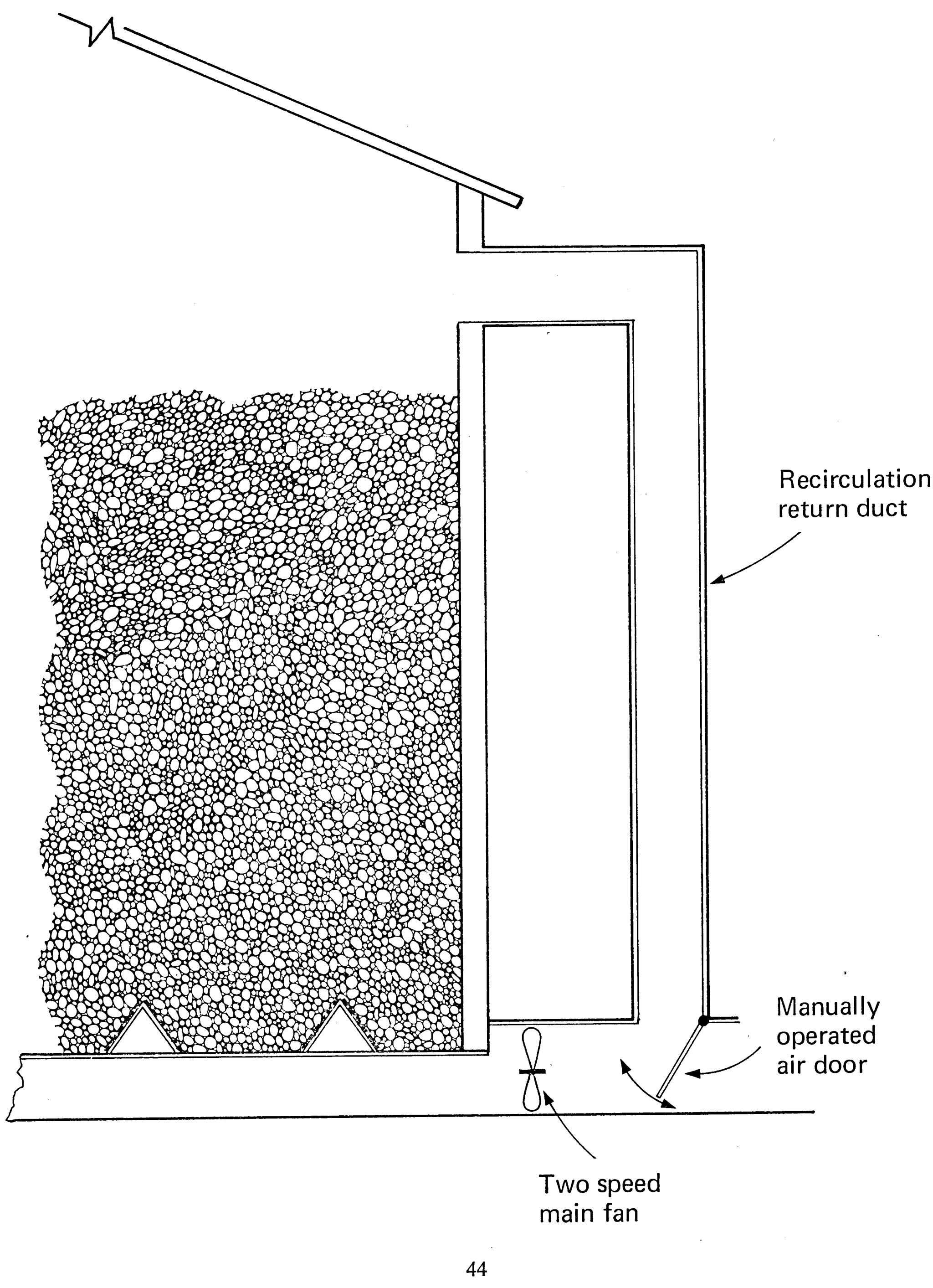


Fig. 27

Recirculation Using Main Fan and Return Duct



Fans for Recirculation

There are three separate mechanical systems commonly used for recirculation although different control systems give many permutations.

(1) Recirculation fan and chimney independent of main ventilating fan.

The layout adopted in Figure 26 is that used at Sutton Bridge and is well suited to the adaption of existing stores. It provides simple recirculation without air mixing. Although two fans have to be purchased (main and recirculation) chimney duct cost is cheap, framed plywood/hardboard surfacing. Running costs are also low as the fan is specifically sized to the duty which should never exceed 35 m^3/tonne/h (20 cfm/ton).

(2) Two speed main fan with recirculation duct.

If the main ventilating fan has a dual winding and can be run at low speed, the entire F.D. system can be used providing arrangements are made to return air to the fan inlet, from the apex of the building. Depending upon the configuration of this duct arrangement air mixing as well as recirculation is possible. The capital cost of a dual wound fan is high, operating costs are low.

(3) Main fan used intermittently.

This is the cheapest installation of the three, it can also provide air mixing and recirculation providing ducting arrangements are suitable.

Bulk stores with a single fan installation will normally have a single return point for re-circulated air. This return point is usually at one end of the building, and such an arrangement will give satisfactory recirculation of the air from up to 30 m (100 ft) away from the return point. For greater distances a simple balanced hardboard duct suspended in the apex of the roof should be used. Recirculation in multi fan installations (typically box stores) has been dealt with in Section 8.

Recommendations

(1) Do not exceed 35 m^3/tonne/h (20 cfm/ton) as a recirculation air flow rate, unless intermittent use of a larger fan is decided upon.

(2) If recirculation facilities are to be provided, consider also air mixing with automatic proportioning control—the benefits are considerable.

(3) The roof must be insulated if recirculation is practiced (see Sutton Bridge Report No. 6, Part I).

(4) If recirculating intermittently do so on a regular basis.

SECTION 10

CONTROL AND MONITORING

Only two aspects of a potato store environment need to be constantly controlled and monitored, they are Temperature and Relative Humidity.

Monitoring

Temperature

There are two broad classifications

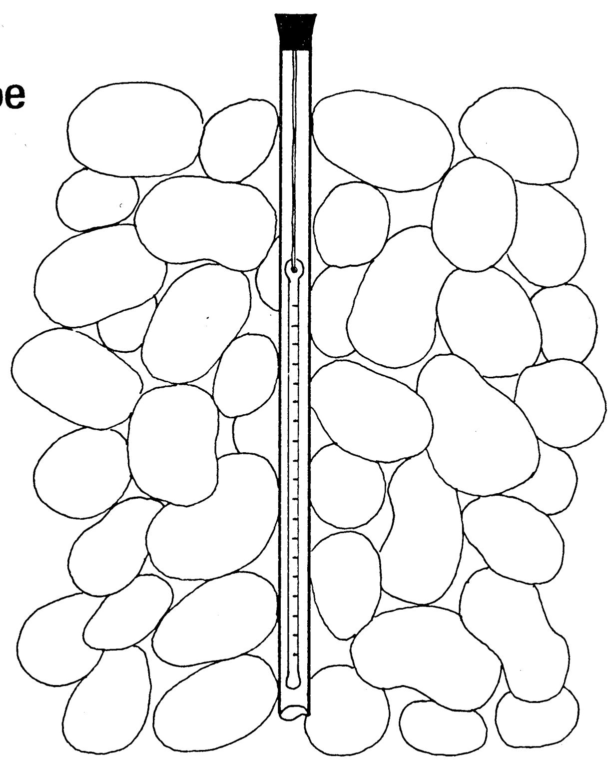
i) Direct reading instruments.

ii) Remote station indicators.

In the first category mercury or alcohol in glass thermometers are widely used. In the context of a potato store, thermometers for measuring potato temperature are normally suspended in the air space between individual tubers approximately 450mm (18 in) below the surface of the stack. This is usually the warmest point, although it is useful to place thermometers at all levels in the crop to determine stack temperature gradients and pinpoint any localised "hot spots". The number of thermometers needed is *not* less than one per 50 tonnes stored and where remote station indicators are used it is advisable to have some mercury thermometers as a check, say one per 100 tonnes stored. If nothing else, this type of thermometer requires entry to the store and encourages visual and odouriferous appraisal, which provides possibly the most reliable indication of potato condition in storage.

Fig. 28

Thermometer in Tube



Alkathene tubing can be used to suspend the thermometer in but it should always be sealed at the top to prevent frost penetration via the tube in uninsulated stores. Unless a standard reference thermometer is used the average instrument is only accurate to within 1°C (2°F).

Remote station indicators, the second category, e.g. thermistors or thermocouples, are operated by a change of resistance, in the sensing head, to the path of an electric current. The signal is used to actuate a scale or digital display and a number of sensing heads are connected to one instrument.

Thermistors produce a very large signal in response to a change in temperature and although the response is non-linear, selection and calibration give an accuracy to within 1°C (2°F) through a range of about 50°C (90°F). The length of the lead cable from the instrument to the sensing head will not affect the reading and therefore this type of indicator is suited to large stores which have distant probes. The large signal produced is suitable for control functions.

Thermocouples produce a very much smaller signal 1:1000 (thermistor), are also non-linear, although to a lesser extent, range 400°C (720°F). Any attempt to increase the length of lead cable must take into account the changed resistance which will result. The need to amplify the signal before it can operate a scale display or a control function makes this part of the equipment expensive.

Humidity

A variety of empirical methods have been devised over the years to meet special conditions, each with its own advantages and disadvantages. The simplest method is the use of a hygroscopic material which alters its dimensions by a large enough amount to make it actuate a mechanism. This mechanism can then be used to indicate or record the relative humidity directly. More accurate instruments rely on the passage of an electrical current between two conductors separated by (typically) a hygroscopic salt.

An indirect method (wet and dry bulb thermometer) uses the cooling effect produced by evaporating water from a fabric sleeve surrounding a thermometer bulb. By the use of tables or slide rules the depression in temperature between wet and dry bulbs can be used to determine accurately the relative humidity, dew point, vapour pressure, etc.

It is important that both types of instrument be sited where direct condensation cannot affect them. In a potato store where the relative humidity is frequently in excess of 70% the cotton or hair sensing element is extremely unreliable even with frequent calibration.

Air Flow Velocity and Resistance

Measurement of these values has been dealt with in Sections 1, 2, 3 and 4. Air velocity and hence volume is often only measured at the time a new store is commissioned if at all, manufacturers fan performance tables usually being considered sufficient guide to the performance of the system. Static water gauge measurement is a useful indication of seasonal variations in the soil content of a stack, degree of sprouting, settling, etc.

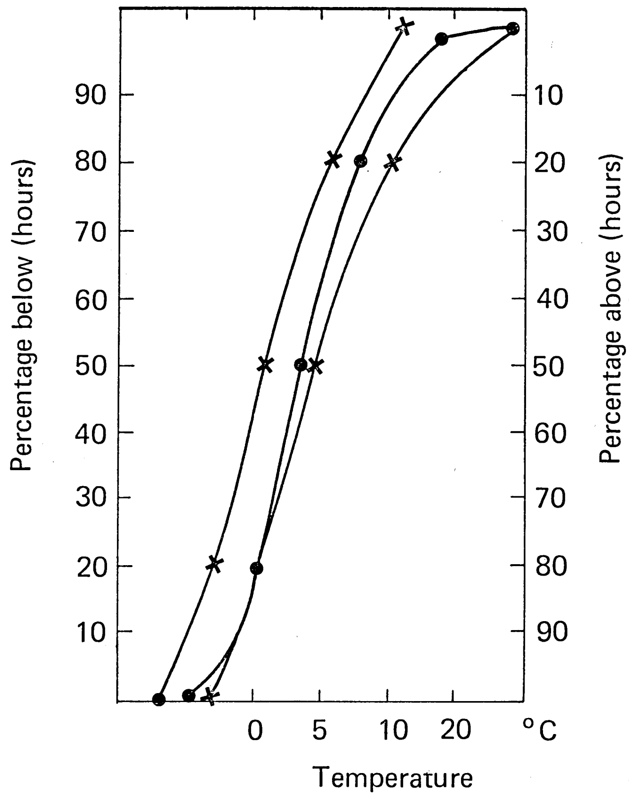
Control

Temperature

The operation of fans may be in response to manual, semi-automatic or fully automatic signals. The cost of automation is small in relation to the capital cost of the store, fans and distribution system or indeed the value of the crop in store. An Ogive curve plotted for the late months of storage indicates a rapidly decreasing percentage of time when air is available for cooling.
Only automatic control can give 100% utilisation of these hours and where storage beyond March is contemplated should be considered mandatory. It is as well to remember that any increase in store temperature above the predetermined optimum is conducive to thermal runaway, i.e. respiration rate rises, sprouting may start, etc. Automation will largely prevent this situation occurring. Manual or semi-automatic control requires a high degree of management and forecasting ability, and above all it is inconvenient as a large proportion of the ventilating hours occur at night:

Fig. 29

Ogive Curve (April)



● Mean daily temperature

× Mean min/max daily temperatures

SEMI-AUTOMATIC SYSTEMS:

(1) Manual starting and stopping of fan in response to a visual appraisal of stack and ambient thermometers. A frost guard thermostat is fitted, to prevent ventilation continuing should the ambient air temperature drop below 0°C (32°F).

(2) Store, ambient and frost guard thermostats fitted. The store thermostat is set at required holding temperature with contacts normally closed (n.c.) when the temperature of store is above that value. The ambient thermostat is adjusted daily to a value approximately 3°C (5·5°F) below the *actual* potato store temperature, contacts normally open (n.o.) above this value. Frost guard thermostat set at, say, 0°C (32°F), contacts normally closed (n.c.) above this value. All three thermostats wired in series. Ventilation is automatic whenever the store temperature exceeds the set value and ambient air temperature is above freezing but below actual store temperature. Continual re-adjustment of ambient thermostat is essential.

AUTOMATIC SYSTEMS:

(1) A differential thermostat with frost guard thermostat is required. The instrument has two sensing elements, one inside the store (potato temperature) and one outside the store (ambient temperature). The differential between the two is sensed and fan operated whenever the outside value is lower than the inside value. A frost guard thermostat overrides the differential thermostat.

(2) As above plus stack temperature thermostat in series overriding differential when potato temperature falls to holding figure preventing further cooling.

Further thermostatic controls are needed where air mixing is practiced (see Section 9).

Humidity

If artificial humidification of the air is undertaken then it must be in response to a fully automatic monitoring and control system. Manual control can too easily allow saturation conditions to develop with the risk of rotting in the stored crop. The target R.H. is 90% and therefore any control system must be above all accurate at this value and quick to respond. A system based on electronic hygrostats or wet and dry bulb sensors is recommended. By using standard thermistors or thermocouples as a pair one sensor can become the "wet bulb" by wrapping it in a wick attached to a small water reservoir. Electronic comparison of the signals will indicate the prevailing relative humidity.

Optimum storage conditions require control of temperature and relative humidity. If there is no artificial humidification, should the humidity of the ambient air determine whether ventilation is carried out? Normally temperature would be considered the dominant management factor and certainly if ambient air had to be at not only the right temperature, but also at, say, 90% R.H. little or no ventilation would be possible in the average season.

The average V.P.D. during the storage season in the U.K. is 1·5 mbar, i.e. 75% R.H. at 0°C, 83% R.H. at 5°C. A compromise solution might be to stop ventilation should the ambient R.H. fall to below, say, 70% R.H. A humidistat in series with the fan starter and temperature controls would be required. Some adjustment of the value quoted may be required in the light of experience.

Siting of Controls

Electro mechanical thermostats i.e. Bimetal, Stem, Vapour Pressure and Liquid Expansion thermostats have only one sensing element. In some instances the sensing head can be remote from the switching instrument and connected by a capillary tube. Obviously severe limitations arise in the siting of such thermostats particularly within the store. Most problematical is the siting of the senser in large potato stores, one senser in a 1000 tonne store is unlikely to present a representative picture of the temperature in that building!

Electronic thermostats using thermistor or thermocouple sensors are extremely flexible with regard to siting and they can be easily suspended out of the way during loading and unloading. An electronic thermostat can be obtained which uses multiple thermistor probes and averages the values for control purposes. For stores in excess of 300 tonnes, multiprobe averaging techniques ought to be considered essential.

Ambient temperature measurement must be away from any external influences, particularly direct sunlight. The frost guard thermostat is best situated in the main air duct downstream of the fan. This is because a fan installation can raise the air temperature by between 1-3°C (2-6°F), being a function of velocity pressure and motor heat. This latter point should be borne in mind when making the settings for any thermostats. In setting semi-automatic and differential thermostatic control systems, after allowance for fan heating effects have been made, there should still be a difference of at least 2°C (4°F) between the temperature of the ventilating air and the crop if any useful cooling is to be achieved.

Recommendations

(1) Have adequate thermometers in the store. Do not rely entirely upon remote station temperature readout.

(2) Read thermometers regularly and record values—this is useful management data.

(3) Automatic control of fan by differential thermostat will give the closest and most satisfactory control.

(4) For medium to large stores use multiprobe temperature averaging for control.

(5) Consider carefully the siting and setting values of all controls.

SECTION 11

REFRIGERATION

The relevant sections of this report have clearly indicated that the climatic conditions in this country permit ambient cooling to maintain optimum potato storage temperatures until mid March (see Figure 1). Thereafter the probability of holding at 4-5°C (low 40's °F) is reduced, until by mid April irrespective of the ventilation rate storage temperatures inevitably rise. Prolonged storage until the end of June at a constant temperature is of course possible by the use of refrigeration. Finding a marketing outlet prepared to pay the premium necessary to cover the not insubstantial costs of refrigeration, in return for the generally enhanced quality of such supplies must be the first consideration. Part of the increased market return necessary must come from the general upward move in prices as the season progresses and therefore refrigerated storage should only be of interest to the long term storer.

Calculation of Refrigeration Duty

Failure to provide sufficient thermal extraction carries heavy penalties. Usually as a result of heat gain through the structure, thermal runaway conditions can develop which may produce substantial deterioration in the last months of storage. This is particularly serious where the capital and operating costs for the equipment have been borne to that date. On the other hand, oversizing the plant can make the whole exercise uneconomic due to the very high capital cost of refrigeration equipment in relation to the value of the crop. The maximum duty is made up of the following components:

(1) Heat produced by crop respiration.
(2) Heat gain through structure.
(3) Heat gain by air leakage.
(4) Heat from fan and any other equipment.
(5) Cooling load from field heat to storage temperature.

HEAT GAIN THROUGH RESPIRATION:

Some values for heat production have already been given in Table 1. Table 6 gives immediate response values for the temperature conditions enumerated. During long term storage, some increase in heat production will occur due to sweetening—this is more marked the lower the temperature and the longer the storage period.

Table 6

Storage Temperature °C (°F)

Respiration Rate	0 (32)	5 (41)	10 (50)	15 (59)	20 (68)
Kilojoules/tonne	21	31	42	63	84
Kilocalories/tonne	5	7·5	10	15	20
Btu/ton	20	30	40	60	80

HEAT GAIN THROUGH STRUCTURE:

In long term bulk storage the heat gain is the largest component of the refrigeration duty. It is made up of two parts (i) gain due to conduction from outside air to store air; (ii) gain due to solar radiation (see Sutton Bridge Report No. 6, Part I). Solar heat gain is important from April onwards and mostly occurs through the roof of the building. Steady state conditions can be assumed to apply to the walls.

HEAT GAIN BY AIR LEAKAGE:

Indicative air leakage rates for the average long term potato store are given in Table 7.

Table 7

Volume of Building		Air changes/hour (leakage)
m^3	ft^3	
28	1000	0·43
142	5000	0·18
283	10 000	0·12
566	20 000	0·09
1133	40 000	0·06
2832	100 000	0·03

HEAT GAIN FROM EQUIPMENT:

Table 8 gives typical values for heat gain due to electric motors.

Table 8

Heat Equivalent of Electric Motors

Motor Horse Power (HP)	Megajoule/HP/h (Btu/HP/h)	
	Connected load in refrigerated space *(i)*	Motor losses outside refrigerated space *(ii)*
$\frac{1}{8}$-$\frac{1}{2}$	4·48 (4250)	2·68 (2545)
$\frac{1}{2}$-3	3·90 (3700)	2·68 (2545)
3-20	3·11 (2950)	2·68 (2545)

Notes: (i) For use when both useful output and motor losses are dissipated within refrigerated space; e.g. motors driving circulation fans, etc.

(ii) For use when motor losses are dissipated outside refrigerated space and useful work of motor is expended within refrigerated space; e.g. pump on a circulating chilled water system, fan motor outside refrigerated space driving fan circulating air with refrigerated space.

COOLING LOAD:

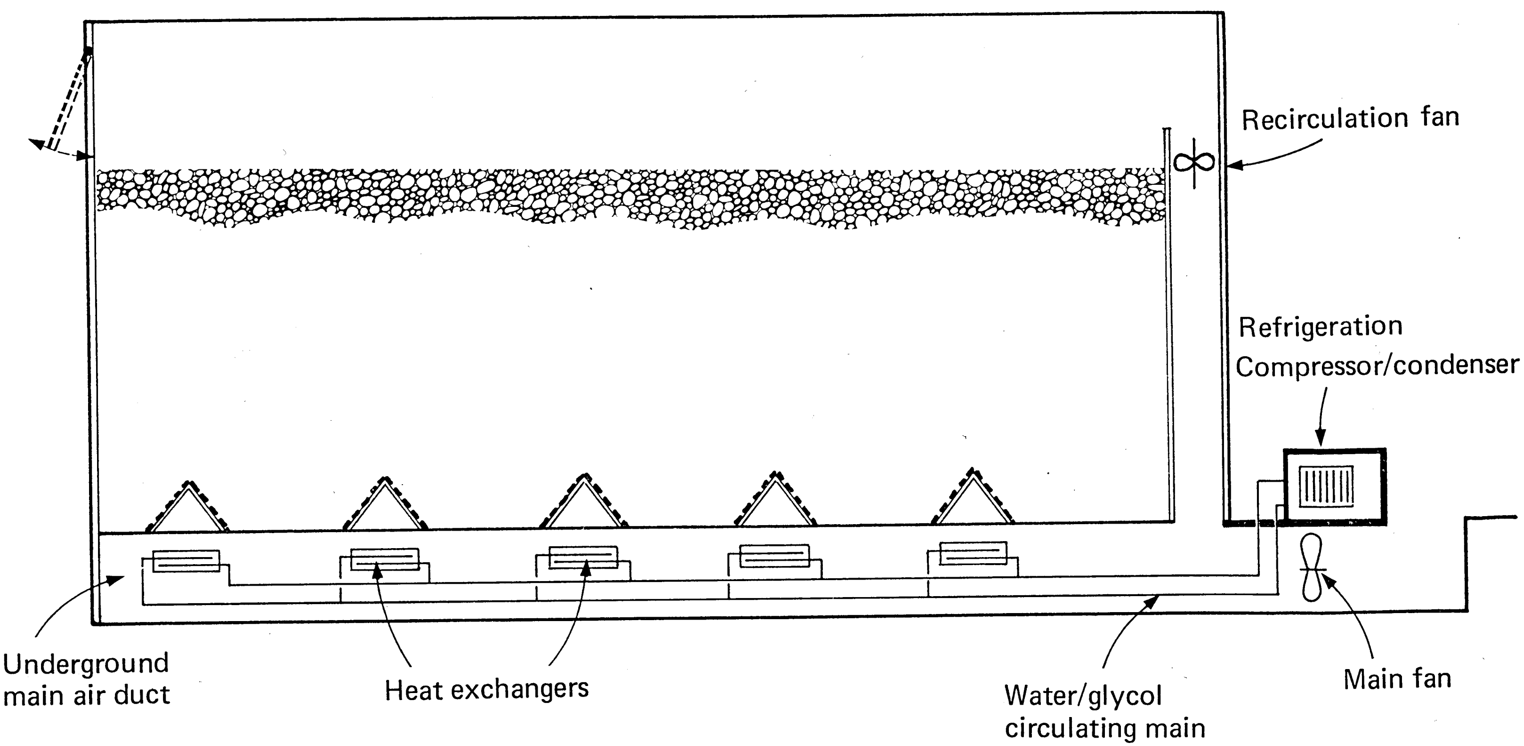
If the field heat of the crop plus the associated high respiration rates applying at that point in time have to be catered for, this can represent a major component of the total refrigeration duty (50-70%). A slower cooling rate is more acceptable for potatoes than for some other vegetable crops. It is recommended that F.D. be used to reduce the store temperature in the first instance, e.g. from, say, 15°C (60°F) at the end of the curing period to 7°C (45°F) before allowing the refrigeration plant to make any final reduction and cope with the steady state condition. F.D. will be cheaper to install and operate than additional refrigeration capacity for this short lived condition.

Air Flow Rates

Recirculation of air in a refrigerated store is essential. To introduce ambient air at possibly 21°C (70°F), cool it to 4°C (40°F) before passing it through the potatoes only to exhaust it several degrees higher is extremely wasteful. Recirculation rates over the evaporator coil of a direct refrigeration system or the cooling coil of an indirect system are normally on a per tonne stored basis in the range 17-35 m^3/tonne/h (10-20 cfm/ton). Any heat exchanger, extracting a given quantity of heat will affect the temperature of that air in direct relation to the volume of air being passed over it.

Fig. 30

Indirect Refrigeration System (Sulton Bridge)



Example 11.1: A 500 tonne potato store has a recirculation airflow rate over the refrigeration cooling coils of 17·5 m^3/tonne/h and the potatoes are respiring and producing 50 kilojoules per tonne per hour. The total heat to be removed is $500 \times 50 =$ 25 000kJ/h. therefore 8750 m^3 of air will have its temperature reduced at the cooling coils by 2.28°C when removing 25 000 kJ, (1kJ will raise 0·8 m^3 of air by 1°C). If the recirculation rate is raised to 35 m^3/tonne/h the air will have its temperature reduced by only 1·14°C at the cooling coils if the precise theoretical extraction is achieved. This latter figure reduces the likelihood of ice formation on the coils and provides a more acceptable temperature gradient in the store.

One of the principal considerations is the sizing of the actual heat exchanger. The broad requirement is for the minimum temperature drop on the air flow which will remove the heat without the air temperature falling below dew point and producing condensate problems. Similarly the design aim for the heat exchanger or water chilled cooler is for a temperature differential (T.D.) across the coils small enough to prevent ice formation. Ice build up on the coils reduces the efficiency of the plant, restricts the flow of recirculating air (magnifying the problem) and represents a weight loss from the potatoes. A low T.D., 2·5-5°C (5-10°F) requires a large surface area to extract a given quantity of heat and this is one of the major contributing factors to the cost of refrigeration equipment applied to potatoes.

Refrigeration Systems

(1) Direct. This form of installation has the recirculated air drawn directly over the evaporator coil of the unit before being blown through the potatoes. One of the problems with this type of installation is that of physically accommodating the large evaporator coil necessary to give a low T.D. It can however be portable equipment; has a quicker response and lower maintenance costs than an indirect system and is more efficient.

(2) Indirect. Here the evaporator coil is used to cool a reservoir of water/glycol which is pumped to remote heat exchangers. The heat exchanger, there may be one or more, are placed downstream of the circulating fan frequently in the main air duct of a bulk store or a recirculatory chimney in a box store.

A number of heat exchangers are used in one of the bulk stores at Sutton Bridge, Figure 30. This allows a large element of heat exchange surface area with a small T.D. $<$3°C (6°F).

Indirect systems are not usually portable; the water/glycol reservoir does, however, provide some capacity to overcome peak loads.

Control

If the crop temperature is to be controlled with refrigeration throughout the storage period, then with the exception of the initial temperature pull down, this will be done by simple thermostatic control, either by

(1) Continuous operation of recirculation fans with thermostatic control of compressor on direct system, or chilled water reservoir and pump on indirect system.

(2) Thermostatic control of compressor or pump together with the fan. By suitably linking the control of refrigeration equipment to the F.D. facilities it is possible to operate during the storage season using automatic control with first preference on the main F.D. fans for cooling, the refrigeration only becoming operative when cooling is required during unfavourable ambient conditions. The attraction of this arrangement is the substantially reduced running costs. The penalty is likely to be higher weight loss during F.D. periods when the R.H. is often considerably below the optimum 95% obtainable off the coil of a good refrigeration installation.

Recommendations

(1) Use Forced Draught Ventilation for initial temperature pull down.

(2) Restrict heat gain to the building by attention to sealing and the provision of an adequate U value, minimum value 0·56 $W/m^2/deg\ C$ (0·1 $Btu/ft^2/h/degF$).

(3) Unless accurately calculated assume total heat gains for storage until mid June to be 100-150% of respiration heat load.

(4) Allow respiration rate of 52 kJ/tonne (50 Btu/ton).

(5) Take expert advice on the detailed installation.

CONCLUSIONS

This is the second part of a two part publication in which the first part dealt with storage buildings. In a potato store both the performance of the storage building and its environmental control equipment are firmly interlinked and the very common practice of designing environmental control systems without reference to the performance characteristics of the storage building is to be discouraged.

The correct sequence of operations is:

(1) To determine the needs of the crop and specify the environmental conditions which must be provided.

(2) To design an environmental control system which will provide and maintain the conditions laid down in (1).

(3) To wrap the crop and its associated environmental control equipment in a suitable package—the storage building.

So frequently the order of consideration on the farm is the exact reverse of this sequence.

It is hoped that the discussion of a wealth of alternatives interlaced with some firm recommendations (arising from the Potato Marketing Board's research at Sutton Bridge and other work) will promote better storage conditions for the potato crop.

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Appendix I

SI CONVERSION FACTORS

Quantity	British or Common Unit	× Factor	=	SI Unit
Length	inch	$2{\cdot}54 \times 10^{1}$	mm	millimeter
	foot	$3{\cdot}048 \times 10^{-1}$	m	metre
Area	square inch	$6{\cdot}452 \times 10^{2}$	mm^2	square millimeter
	square foot	$9{\cdot}290 \times 10^{-2}$	m^2	square metre
Volume	cubic foot	$2{\cdot}832 \times 10^{-2}$	m^3	cubic metre
Velocity	foot/minute	$5{\cdot}080 \times 10^{-3}$	m/s	metre/second
Mass	pound	$4{\cdot}536 \times 10^{-1}$	kg	kilogramme
	ton	$1{\cdot}016 \times 10^{3}$ 1·016	kg tonne	kilogramme
Density	pound/foot^3	$1{\cdot}602 \times 10^{1}$	kg/m^3	kilogramme/cubic metre
Volume Flow Rate	cubic foot/minute	$4{\cdot}719 \times 10^{-4}$ 1·699	m^3/s m^3/h	cubic metre/second cubic metre/hour
Pressure	pound force/inch^2	$6{\cdot}895 \times 10^{3}$ $6{\cdot}895 \times 10^{-2}$	N/m^2 bar	Newton/square metre bar
	inch H_2O (4°C)	2·491	mbar	millibar
	millimeter H_2O (4°C)	$9{\cdot}807 \times 10^{-2}$	mbar	millibar
	millimetre Hg	1·333	mbar	millibar
	atmosphere (standard)	1·013	bar	bar
Work	horsepower hour	$2{\cdot}685 \times 10^{6}$	J	joule
	kilowatt hour	$3{\cdot}6 \times 10^{6}$	J	joule
Power	horsepower	$7{\cdot}457 \times 10^{2}$	W	watt
Temperature (rise)	degree fahrenheit	$5{\cdot}500 \times 10^{-1}$	deg C	degree celsius
Heat	British Thermal Unit (Btu)	$1{\cdot}055 \times 10^{3}$	J	joule
	horsepower hour	$2{\cdot}685 \times 10^{6}$	J	joule
	kilowatt hour	$3{\cdot}6 \times 10^{6}$	J	joule
	kilocalorie	$4{\cdot}187 \times 10^{3}$	J	joule
Heat flow	Btu/hour	$2{\cdot}931 \times 10^{-1}$	W J/s	watt joule/second
	horsepower	$7{\cdot}457 \times 10^{2}$	W	watt
	(12 000 Btu/h) ton refrigeration	$3{\cdot}517 \times 10^{3}$	W	watt
	kilocalorie/hour	1·163	W	watt
Heat Energy Content	Btu/pound	$2{\cdot}236 \times 10^{3}$	J/kg	joule/kilogramme
Moisture Content	grain/pound	$1{\cdot}429 \times 10^{-1}$	g/kg	gramme/kilogramme
Light	foot candle	$1{\cdot}076 \times 10^{-1}$	lx	lux
	lumen/foot^2	$1{\cdot}076 \times 10^{1}$	lx	lux

Appendix II

TEMPERATURE CONVERSION TABLE

°F	°C	°F	°C	°F	°C	°F	°C
0	—17·8	54	12·2	108	42·2	162	72·2
1	—17·2	55	12·8	109	42·8	163	72·8
2	—16·7	56	13·3	110	43·3	164	73·3
3	—16·1	57	13·9	111	43·9	165	73·9
4	—15·6	58	14·4	112	44·4	166	74·4
5	—15·0	59	15·0	113	45·0	167	75·0
6	—14·4	60	15·6	114	45·6	168	75·6
7	—13·9	61	16·1	115	46·1	169	76·1
8	—13·3	62	16·7	116	46·7	170	76·7
9	—12·8	63	17·2	117	47·2	171	77·2
10	—12·2	64	17·8	118	47·8	172	77·8
11	—11·7	65	18·3	119	48·3	173	78·3
12	—11·1	66	18·9	120	48·9	174	78·9
13	—10·6	67	19·4	121	49·4	175	79·4
14	—10·0	68	20·0	122	50·0	176	80·0
15	— 9·4	69	20·6	123	50·6	177	80·6
16	— 8·9	70	21·1	124	51·1	178	81·1
17	— 8·3	71	21·7	125	51·7	179	81·7
18	— 7·8	72	22·2	126	52·2	180	82·2
19	— 7·2	73	22·8	127	52·8	181	82·8
20	— 6·7	74	23·3	128	53·3	182	83·3
21	— 6·1	75	23·9	129	53·9	183	83·9
22	— 5·6	76	24·4	130	54·4	184	84·4
23	— 5·0	77	25·0	131	55·0	185	85·0
24	— 4·4	78	25·6	132	55·6	186	85·6
25	— 3·9	79	26·1	133	56·1	187	86·1
26	— 3·3	80	26·7	134	56·7	188	86·7
27	— 2·8	81	27·2	135	57·2	189	87·2
28	— 2·2	82	27·8	136	57·8	190	87·8
29	— 1·7	83	28·3	137	58·3	191	88·3
30	— 1·1	84	28·9	138	58·9	192	88·9
31	— 0·6	85	29·4	139	59·4	193	89·4
32	0	86	30·0	140	60·0	194	90·0
33	0·6	87	30·6	141	60·6	195	90·6
34	1·1	88	31·1	142	61·1	196	91·1
35	1·7	89	31·7	143	61·7	197	91·7
36	2·2	90	32·2	144	62·2	198	92·2
37	2·8	91	32·8	145	62·8	199	92·8
38	3·3	92	33·3	146	63·3	200	93·3
39	3·9	93	33·9	147	63·9	201	93·9
40	4·4	94	34·4	148	64·4	202	94·4
41	5·0	95	35·0	149	65·0	203	95·0
42	5·6	96	35·6	150	65·6	204	95·6
43	6·1	97	36·1	151	66·1	205	96·1
44	6·7	98	36·7	152	66·7	206	96·7
45	7·2	99	37·2	153	67·2	207	97·2
46	7·8	100	37·8	154	67·8	208	97·8
47	8·3	101	38·3	155	68·3	209	98·3
48	8·9	102	38·9	156	68·9	210	98·9
49	9·4	103	39·4	157	69·4	211	99·4
50	10·0	104	40·0	158	70·0	212	100·0
51	10·6	105	40·6	159	70·6		
52	11·1	106	41·1	160	71·1		
53	11·7	107	41·7	161	71·7		

Appendix III

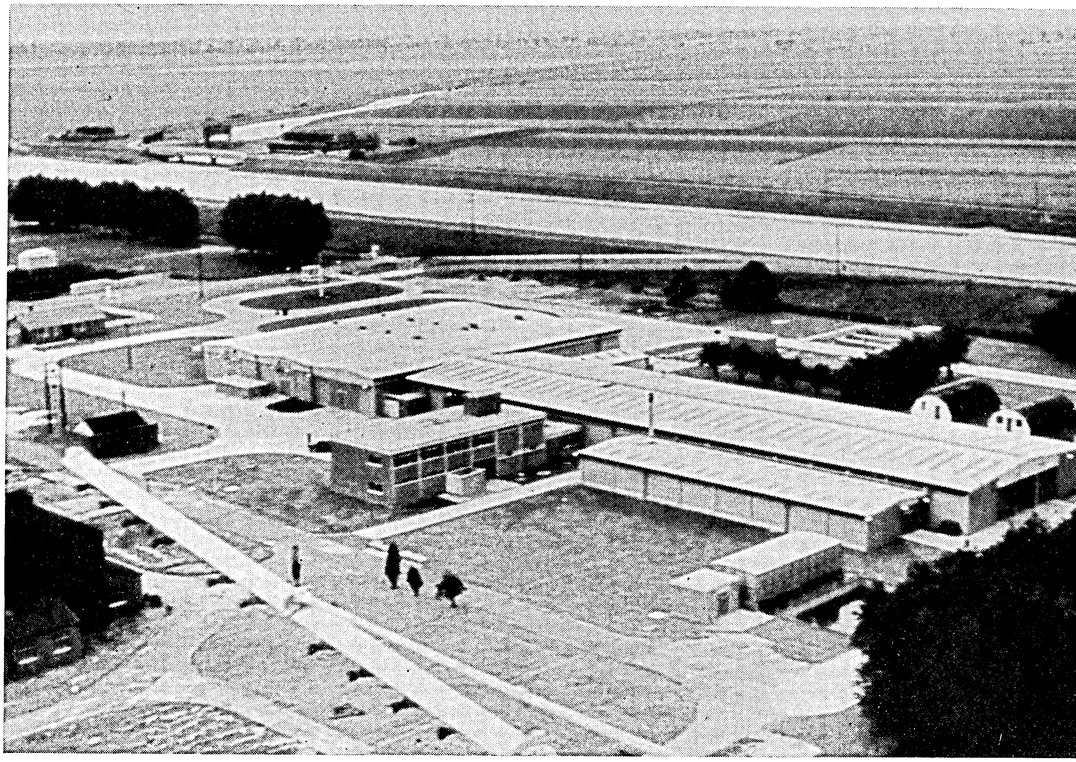
PREFIXES FOR SI UNITS

Prefix	Symbol	Factor		
tera	T	10^{12}	=	1 000 000 000 000
giga	G	10^{9}	=	1 000 000 000
mega	M	10^{6}	=	1 000 000
kilo	k	10^{3}	=	1 000
milli	m	10^{-3}	=	0·001
micro	u	10^{-6}	=	0·000 001
nano	n	10^{-9}	=	0·000 000 001
pico	p	10^{-12}	=	0·000 000 000 001

Appendix IV

ABBREVIATIONS

Btu/h/ft²/deg F	—	British Thermal Units per hour per square foot per degree Fahrenheit.
Btu/ton/h	—	British Thermal Units per ton per hour.
Cal/tonne/h	—	Calories per tonne per hour.
cfm	—	cubic feet per minute.
F.D.	—	forced draught ventilation.
H.P.	—	horse power
kJ/tonne/h	—	kilojoules per tonne per hour.
kWh	—	kilowatt hour.
m^3/tonne/h	—	cubic metres per tonne per hour.
m/s	—	metres per second.
R.H.	—	relative humidity.
s.w.g	—	static water gauge
t.w.g.	—	total water gauge.
T.D.	—	temperature difference.
W/m^2/deg C	—	watts per square metre per degree Centigrade.



The Board's Experimental Station at Sutton Bridge in Lincolnshire commenced full operation in October, 1964.

The principal operations with which it is concerned are:—

1. Handling and storing potatoes in varying conditions.
2. Finding the most efficient methods of dressing out samples to a quality standard.
3. The test marketing of quality grades.
4. The disposal to the best advantage of outgrades.

Its purpose, therefore, is to find ways of obtaining the best possible return from the potato crop and to pass on the information to producers.

Full details of all reports currently available are given at the end of this booklet.

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Report No. 7 The Harvesting, Handling and Storage of Potatoes in Bulk and Boxes.

MISCELLANEOUS

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The Production of Potatoes for Canning.

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