

# DESIGN OF AIR DISTRIBUTION SYSTEMS FOR CLOSED GREENHOUSES

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## Abstract

In order to maintain uniform environmental conditions within a greenhouse, and hence uniform crop production, it is important that the air distribution system be correctly designed. This is more critical in a closed greenhouse, specifically designed for carbon dioxide enrichment, because low natural air infiltration can result in significant vertical and horizontal stratification of temperature, humidity, and carbon dioxide concentration.

The use of perforated polyethylene ducting to distribute air within greenhouses has become increasingly common. Ducting can be used for general air circulation, to eliminate stratification, and to deliver warm, cool, and CO<sub>2</sub> enriched air to precise points within the greenhouse. With careful design the greenhouse engineer should be able to ensure that all plants within a greenhouse receive optimal environmental conditions.

The designer of perforated ducting is faced with a number of choices regarding the size and layout of the system: the number of ducts to be used and their position within the greenhouse relative to the crop rows, the length and diameter of the duct, the number, size, shape, and spacing of the discharge holes, and the static pressure within the duct. Other factors that will affect the selection are the required air circulation rate and air velocity within the plant canopy to enhance heat and mass transfer without causing damage, and the temperature of the air being delivered. Once a duct has been selected the designer would like to know its system characteristics so that the most efficient inflation fans can be selected.

## 1. Introduction

Several researchers have studied the discharge versus pressure characteristics of perforated ducts, and modelled them mathematically. Most efforts have been focused on the problem of non-uniform discharge along the length of a duct with uniformly spaced holes. This non-uniformity arises due to a combination of changes in the static pressure and the discharge coefficient at each hole along the duct. The first phenomenon occurs as a result of the combined effects of frictional losses, and the static pressure regain effect, induced by a loss in forward momentum along the duct. The second phenomenon arises when the ratio of the static pressure to the velocity pressure in the duct is low.

Under these conditions the discharge coefficient can be significantly less than the normal value for a submerged orifice.

Carpenter (1972) and Zamir (1973) performed experimental evaluations of perforated ducts from which they developed simple procedures for designing ducts with non-uniform hole spacing for near uniform discharge. Bailey (1975) conducted a detailed study of the way in which the static regain and discharge coefficients varied along a perforated duct. Based on this work, Bailey (1982) developed a computer program to design ducts with non-uniform hole spacing to give near uniform discharge. Saunders & Albright (1984) developed a computer model of perforated duct performance, but assumed that the static regain and discharge coefficients were constant along the length of the duct.

In 1988 a computer program based on the work of Bailey (1985) was developed and was validated by Amos (1989) and Studman et al (1991). As with the computer programs of Bailey (1982) and Saunders & Albright (1984) this implementation is based on an implicit model and involves iteration to find the static pressure at the beginning of the duct for a chosen total discharge rate, and hence the overall discharge coefficient of the duct. Amos (1989) developed an explicit mathematical model for the overall discharge coefficient of a duct based on the aperture ratio (ratio of total area of perforations to the cross-sectional area of the duct), and the ratio of the duct length to diameter. This explicit model has recently been revised based on the data of Studman et al (1991), and further tests using the implicit simulation model. There is also good agreement with the experimental data of Saunders & Albright (1984) and Zamir (1973).

## 2. Theory

The flow rate versus pressure relationship for a perforated duct is a complex function of the following factors: the diameter and length of the duct; the size, shape and number of outlet perforations; and the spacing of the perforations. In the following discussion only ducts with circular shaped perforations are considered.

The total flow in a duct,  $Q_o$ , is equal to: the product of the average inlet velocity and the duct area (Equation 1); and the sum of the outflows at each perforation (Equation 2).

$$Q_o = v_o A = v_o \frac{\pi D^2}{4} \quad (1)$$

$$Q_o = \sum_{i=1}^n Q_i = \sum_{i=1}^n \left( C_{di} a_i \sqrt{\frac{2p_i}{\rho}} \right) = \sum_{i=1}^n \left( C_{di} \frac{\pi d_i^2}{4} \sqrt{\frac{2p_i}{\rho}} \right) \quad (2)$$

From Bernoulli's equation, the total flow in the duct can also be expressed in terms of the inlet static pressure,  $p_o$ , according to Equation 3.

$$Q_o = \check{C}A\sqrt{\frac{2p_o}{\rho}} = C\frac{\pi D_2}{4}\sqrt{\frac{2p_o}{\rho}} \quad (3)$$

The overall discharge coefficient,  $C$ , has been found by non-linear regression to be a function of the aperture ratio,  $na/A$ , and the ratio of duct length,  $L$ , to duct diameter,  $D$ . (Equation 4 has a correlation coefficient of  $R^2 = 0.997$ ).

$$C = 0.1077 + 0.6424\left(\frac{na}{A}\right)^{1.634} - 0.001402\left(\frac{na}{A}\right)^{2.49} \left(\frac{L}{D}\right)^{0.793} \quad (4)$$

The data used for this fit were derived from 672 simulation runs of the computer model and are shown in Figure 1. The range of parameters tested included duct lengths between 10 and 100m, duct diameters between 0.15 and 1m, hole diameters between 13 and 75mm, 50 to 100 holes per row, one, two or three rows of holes and air flow rates between 0.1 and 2 m<sup>3</sup>.s<sup>-1</sup>. This resulted in aperture ratios between 0.203 and 2, and duct length to diameter ratios between 10 and 667. The computer model was found to agree well with experimental data (Amos, 1989; Studman et al 1991).

For aperture ratios of less than 1.5 a simpler relationship was also found (Equation 5). Equation 5 has a correlation coefficient of  $R^2 = 0.971$ .

$$C = 0.69\left(\frac{na}{A}\right) \quad (5)$$

Figure 2 shows that ducts with aperture ratios of less than 1.5 give the most uniform discharge. The variability parameter used in Figure 2 is a Chi-squared test statistic calculated from the difference between simulated and average discharge rate at each hole.

Using this information, relationships for air circulation requirements, and jet momentum theory it is now possible to set out a design procedure for perforated polyethylene ducts.

### 3. Design Procedure

It is suggested that the design of perforated polyethylene ventilation ducts in a closed greenhouse should be based on the following procedure:

1. Calculate the total air circulation rate.
2. Determine the number of ducts, their position and length.
3. Determine the number of rows of holes around duct circumference.
4. Determine the discharge jet throw distance.
5. Select the design static pressure for the duct.
6. Calculate hole size.
7. Calculate number of holes.
8. Calculate duct diameter.
9. Check aperture ratio and inlet pressure. Return to step 7 if required.
10. Check hole size and jet throw. Return to step 6 if required.

11. Draw system curve, including losses in supply ducts, elbows and takeoffs. Select suitable inflation fan. Return to step 5 if required.

The procedure assumes that all holes will be of uniform size and spacing. While it is conceded that this is sub-optimal, it will lead to a duct which is easy to manufacture, without significant loss of uniformity.

### 3.1 Air Circulation Rate

The correct air circulation rate in a closed greenhouse is important to avoid stratification, stagnant air pockets, and to ensure that the crop is not subjected to excessively hot or cold draughts of air from the air conditioning system. For many years the American Society of Agricultural Engineers (ASAE, 1992) has recommended a minimum air circulation rate of 15 to 20 air charges per hour to provide good mixing. Our recommendation is 30 air changes per hour. This extra requirement is justified to avoid hot and cold draughts from the air conditioning equipment.

By equating the heat loss from a greenhouse:

$$q_{loss} = UA_f \Delta T_{io} \quad (6)$$

with the heat input from the heating system:

$$q_{heat} = Q \rho C_p \Delta T_h = \frac{NV}{3600} \rho C_p \Delta T_h \approx \frac{NV}{3} \Delta T_h \quad (7)$$

it can be shown that:

$$\Delta T_h \approx \frac{3}{NV} UA_f \Delta T_{io} \approx \frac{3}{Nz} U \Delta T_{io} \quad (8)$$

Similarly if the net heat gain of a greenhouse from solar radiation and through the cover:

$$q_{gain} = \tau F I_o A_f - UA_f \Delta T_{io} \quad (9)$$

is equated with the heat removal of the cooling system:

$$q_{cool} = Q \rho C_p \Delta T_c = \frac{NV}{3600} \rho C_p \Delta T_c \approx \frac{NV}{3} \Delta T_c \quad (10)$$

it can be shown that:

$$\Delta T_c \approx \frac{3}{NV}(\tau FI_o A_f - UA_f \Delta T_{io}) \approx \frac{3}{Nz}(\tau FI_o - U \Delta T_{io}) \quad (11)$$

For a typical closed single skin greenhouse the overall heat loss coefficient,  $U$ , is  $10\text{W.m}^2\text{floor.K}^{-1}$ , and the average height,  $z$ , is  $3\text{m}$ . When heating is used to maintain the inside-outside temperature difference,  $\Delta T_{io}$ , at  $20\text{K}$  then an air circulation rate,  $N$ , of  $30$  air-changes per hour will give a temperature rise through the heating system,  $\Delta T_h$ , of approximately  $7\text{K}$ . For the same greenhouse when the outside insolation,  $I_o$ , is  $1000\text{W.m}^2$  and the cooling system is maintaining the inside outside temperature difference,  $\Delta T_{io}$ , at  $-5\text{K}$ , then the temperature drop through the cooling system,  $\Delta T_c$ , will be approximately  $13\text{K}$ , (assuming the greenhouse light transmission factor,  $\tau$ , is  $0.7$  and the sensible heat load fraction,  $F$ , is  $0.5$ ).

### 3.2 Position of Ducts, Number and Length

The position of the ducts employed, their number, and length will depend on the layout of the greenhouse, the type of crop grown, and the way it is trained or benched. The three most common positions are: overhead, in the apex of the roof; under-bench; or between crop rows. The first two choices usually lead to the selection of a few large ducts, whereas the third will normally require a large number of small ducts. For the overhead configuration, at least one duct is required per span. For large spans, more than one duct may be required. Overhead ducts will cause some light loss and may not give good air circulation in tall crops. While under-bench systems do not cause light loss they may not give adequate air flow rate across the top of the benches. Ducts placed within rows give good air circulation in tall crops but may interfere with cultivation procedures. Care must also be taken to avoid excessive air velocities at the leaves.

The length of the duct will usually be determined by the length of the greenhouse, crop rows, or benches. In a very large greenhouse, more than  $30\text{m}$  long, it can be advantageous to limit the duct length to half the greenhouse or row length and feed air from either the centre or ends. This will avoid excessively high air velocities at the beginning of the duct, which cause greater non-uniformity of discharge. These decisions will also be influenced by practical considerations such as the design of the greenhouse and the proposed location of the air conditioning equipment.

### 3.3 Number of Rows of Holes

For overhead ducts it is usual to have two rows of holes angled downward at about  $120^\circ$  from the vertical so that the jets discharge down the angle of the roof. For under-bench ducts one or two rows of holes can be used depending on the placement of the bench relative to the walls. For small ducts within the crop rows, one row of holes pointing vertically upwards, or two rows of horizontally opposed holes is most common. When a large number of holes are required in a relatively short duct, it may be desirable

to have three rows of holes. This allows for optimum aperture ratio without holes being too close together, and weakening the duct.

### 3.4 Discharge Jet Throw Distance

The throw of a jet of air from a ventilation duct has been studied extensively (Wilson et al, 1983). A jet of air contains four identifiable zones. In the first zone (~4 diameters) the core velocity is essentially the same as the outlet velocity. In the second zone (~8 diameters) the velocity decreases with the inverse square root of the distance from the outlet. The most important part of the jet is the third zone (~25 to 100 diameters) where the velocity decreases inversely to the distance from the outlet. The fourth zone is the terminal zone in which the core velocity decreases rapidly to still air conditions. In the third zone the distance from the outlet for a given core velocity of the air is given by:

$$X = \frac{v_i}{v_x} K \sqrt{C_{di}} a \quad (12)$$

The value of  $K$  is typically 5.7.

The discharge velocity,  $v_i$ , is a function of the static pressure inside the duct relative to that outside,  $p_i$ , and is given by:

$$v_i = \sqrt{\frac{2p_i}{\rho}} \quad (13)$$

Ducts should be designed in such a way that the jet velocity at the furthest point between two ducts, or at the plants, is not more than  $1.0\text{m.s}^{-1}$ , and not less than  $0.5\text{m.s}^{-1}$ . This is slightly higher than still air ( $\approx 0.3\text{m.s}^{-1}$ ), and helps avoid pockets of stagnant air.

### 3.5 Duct Static Pressure

For a typical duct the inside static pressure relative to ambient,  $p_i$ , should be between 25 and 100Pa. If the static pressure is less than 25Pa at any point the duct will not inflate properly, will flap, and total flow will be reduced. Pressures above 100Pa are not recommended since the fan power required increases as the pressure increases. Initially a value of 50Pa should be used to ensure that the inlet static pressure does not fall below 25Pa. This may be reduced later.

### 3.6 Hole Size

If Equation 13 is substituted into Equation 12 together with the hole area,  $a$ , in terms of the hole diameter,  $d$ , then rearrangement gives:

If  $v_x$  is assumed to be  $0.5\text{m.s}^{-1}$ , and  $C_{di}$  is assumed to be 0.64 (submerged sharp edged orifice), then:

$$d = \frac{v_x X}{K} \sqrt{\frac{2\rho}{\pi p_i C_{di}}} \quad (14)$$

$$d \approx \frac{X}{10 \sqrt{p_i}} \quad (15)$$

For a static pressure of 50Pa the best hole size will be:

$$d \approx \frac{X}{70} \quad (16)$$

In practise the actual hole diameter should be selected from those conveniently made with punches produced from available pipe sizes.

### 3.7 Number of Holes

A first estimate of the number of holes required can be found from equation 2 by assuming that the discharge coefficient,  $C_{di}$ , and static pressure,  $p_i$ , are constant along the duct. Thus:

$$n \approx \frac{4Q_o}{C_{di} \pi d^2} \sqrt{\frac{\rho}{2p_i}} \quad (17)$$

Note that the discharge coefficient in equation 17 is not strictly the same as that in equations 12 to 14.

By comparing equations 17 and 3 and setting  $p_i$  to  $p_o$ , it can be seen that:

$$C = C_{di} \left( \frac{na}{A} \right) \quad (18)$$

Using equation 5 it can be shown that a suitable value for the average discharge coefficient,  $C_{di}$ , is 0.69. Using this, a simplification can be made to find the number of holes required. Thus:

$$n \approx \frac{1.4 Q_o}{d^2 \sqrt{p_o}} \quad (19)$$

with the proviso that  $n$  should be an exact multiple of the number of rows of holes, and that the space between holes should not be less than 1.5 hole diameters. The actual number of holes must be decided iteratively with the duct diameter and the desired inlet pressure.

### 3.8 Duct Diameter

A first estimate of the duct diameter can be made by considering the maximum desirable velocity at the entrance of the duct. The work of Bailey (1975) showed that the actual discharge coefficient,  $C_{d1}$ , for any perforation would decrease from the theoretical value for a sharp edged orifice of 0.64, as the ratio of the static pressure to the velocity pressure in the duct decreased. The actual discharge coefficient is found to be less than 90% of the theoretical when:

$$\frac{2p_o}{\rho v_o^2} < 1.5 \quad (20)$$

For an entrance static pressure,  $p_o$ , of 25Pa then the inlet velocity,  $v_o$ , should be kept below 5.3m.s<sup>-1</sup>. The respective maximum velocities for inlet pressures of 50 and 100Pa are 7.5 and 10.5m.s<sup>-1</sup>.

The minimum duct diameter can thus be found by rearranging equation 1 to give:

$$D \approx \frac{2}{\sqrt{\pi}} \sqrt{\frac{Q_o}{v_o}} \quad (21)$$

For an average value of  $v_o$  then the minimum duct diameter can be found from:

$$D > 0.4 \sqrt{Q_o} \quad (22)$$

In a similar manner to the hole size the actual duct diameter chosen will depend on commercially available sizes of lay-flat.

### 3.9 Check Aperture Ratio and Inlet Static Pressure

Having obtained initial estimates of the number of holes and the duct diameter the aperture ratio should now be calculated. If the aperture ratio is greater than 1.5 then the duct discharge will be non-uniform. To counteract this the duct diameter should be increased, or the number of holes decreased. In practise an aperture ratio of around 1 seems to give the best compromise between uniform discharge and avoiding high inlet pressures.

The suitability of the duct design can be checked by calculating the overall discharge coefficient,  $C$ , from Equation 4, and using this with Equation 3 to determine the inlet static pressure,  $p_o$ . If  $p_o$  is less than 25Pa then the duct will not inflate properly at the entrance, and the aperture ratio should be decreased. The most desirable inlet pressure will depend on the type of inflation fan used.



### 3.10 Check Jet Throw and Hole Size

The actual jet throw at the beginning of the duct can now be calculated using Equations 12 and 13. If the jet throw is considered to be too different from the design specification then the hole size may be altered or the number of holes changed. Each will alter the aperture ratio and hence the inlet static pressure. Equation 15 can be used to calculate a new hole size. This process can be repeated as required.

### 3.11 Fan Selection

When a suitable duct has been designed the system curve can be drawn using the value of the overall discharge coefficient,  $C$ , calculated from Equation 4, and friction losses calculated for supply ducts, elbows, and takeoffs. A suitable fan can then be selected by matching the fan curve with the system curve to find the operating point. If a fan can not be found where the actual operating point gives the design air flow rate it may be necessary to alter the design static pressure in the duct, and reselect the various duct dimensions.

## 4. Conclusion

A method has been presented for designing low pressure perforated polyethylene ducts for closed greenhouses. The method is based on a well validated implicit mathematical model of duct performance, jet discharge theory, and an explicit equation for the overall discharge coefficient of a duct, based on duct parameters. Ventilation ducts designed according to this procedure should provide adequate and uniform air-circulation to ensure that all plants within a greenhouse receive similar conditions of air temperature, humidity, and carbon dioxide concentration. Care is also taken to avoid plants being affected by excessively hot or cold air, and to ensure that there is adequate air movement within the canopy of large plants to enhance heat and mass exchanges at the leaf surfaces.

## 5. Definition of Symbols

$A$	cross-sectional area of duct ( $m^2$ )
$A_r$	floor area of a greenhouse ( $m^2$ )
$a$	area of a hole ( $m^2$ )
$C$	overall discharge coefficient of the duct
$C_{di}$	discharge coefficient of a hole
$C_p$	specific heat capacity of air ( $1.0kJ.kg^{-1}.K^{-1}$ )
$D$	diameter of duct (m)
$d$	diameter of holes (m)
$F$	sensible heat load fraction of absorbed solar radiation
$I_o$	insolation above greenhouse ( $W.m^{-2}$ )
$L$	length of the duct (m)
$N$	air change rate in greenhouse ( $hr^{-1}$ )

$n$	number of holes in the duct
$p_i$	static pressure inside the duct (Pa)
$p_o$	static pressure at the duct entrance (Pa)
$Q$	total air circulation rate in greenhouse ( $\text{m}^3.\text{s}^{-1}$ )
$Q_i$	air flow from a hole ( $\text{m}^3.\text{s}^{-1}$ )
$Q_o$	total air flow in duct ( $\text{m}^3.\text{s}^{-1}$ )
$q_{\text{cool}}$	cooling load of a greenhouse (W)
$q_{\text{gain}}$	net heat gain of a greenhouse (W)
$q_{\text{heat}}$	heating load of a greenhouse (W)
$q_{\text{loss}}$	net heat loss of a greenhouse (W)
$V$	volume of greenhouse ( $\text{m}^3$ )
$v_i$	air velocity of a discharge jet at the hole ( $\text{m}.\text{s}^{-1}$ )
$v_o$	air velocity at duct entrance ( $\text{m}.\text{s}^{-1}$ )
$v_x$	air velocity of a discharge jet at distance $X$ from hole ( $\text{m}.\text{s}^{-1}$ )
$U$	overall heat loss coefficient of a greenhouse ( $\text{W}.\text{m}^{-2}.\text{K}^{-1}$ )
$X$	throw distance of a jet of air (m)
$z$	average height of greenhouse (m)
$\Delta T_c$	temperature drop through cooling system (K)
$\Delta T_h$	temperature rise through heating system (K)
$\Delta T_{io}$	temperature difference between inside and outside of greenhouse (K)
$\rho$	density of air ( $1.2\text{kg}.\text{m}^{-3}$ )
$\tau$	average light transmission factor of greenhouse

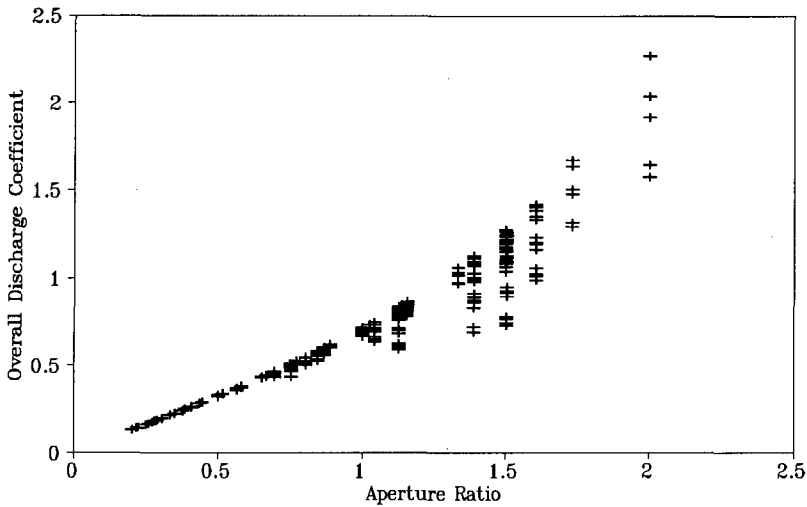
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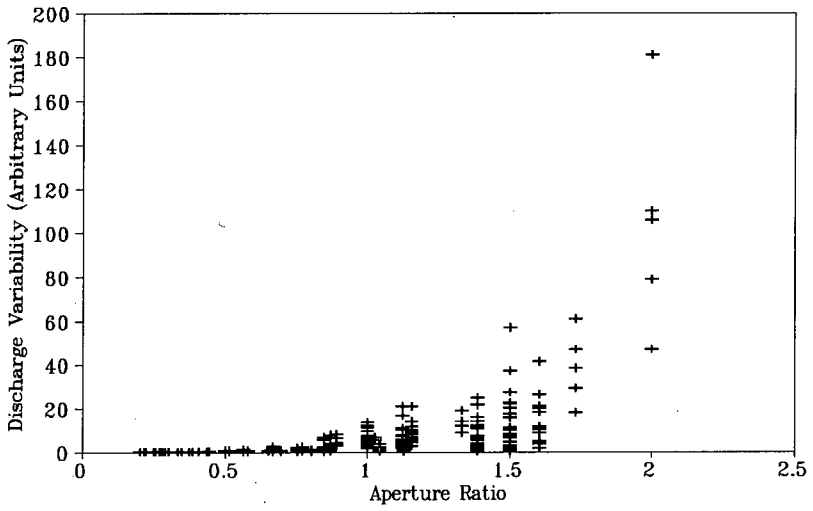
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**Figure 1** Overall Discharge Coefficient  $C$  versus Aperture Ratio for 672 simulated ducts. (For discussion see section 2).



**Figure 2** Variability of air discharge versus Aperture Ratio for 672 simulated ducts. (For discussion see section 2).