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REVIEW PAPER

Convective and Ventilation Transfers in Greenhouses, Part 1: the Greenhouse considered as a Perfectly Stirred Tank

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In this paper, the characterization and modelling of the most relevant convective transfers contributing to the elaboration of the greenhouse climate are reviewed. Convective transfers include heat and mass transfers between air and solid surfaces (walls, roof, leaves) along with air, heat, water vapour and tracer gas transfers to or from the inside air. Adopting the assumption that the greenhouse is a perfectly stirred tank, the specific characterization methods associated with this approach are reviewed. The perfectly stirred tank approach requires the assumption of uniform temperature, humidity and CO_2 content inside the greenhouse and uses a 'big leaf' model to treat the plant canopy and describe the exchanges of latent and sensible heat with inside air. The simulation of the ventilation processes associated with this simplified approach is based on the Bernoulli equation and on the experimental determination of semi-empirical parameters by means of air exchange rate measurements. The techniques used to measure temperature and air exchange rates measurements pertaining to the whole greenhouse volume are presented. A complete panorama of the studies in relation to the transfer coefficients between the different surfaces together with the ventilation performances of various greenhouse types are also presented.

This paper is the first part of a review of the convective transfers in greenhouses and in the second paper, a similar study based on the approach of the distributed climate is presented.

1. Introduction

Radiative and convective transfers are the main exchange processes directly influencing crop production through photosynthesis and transpiration. Their combined results also act on the greenhouse microclimate which in turn affects crop growth and development. The quantity and quality of the radiation that the covering material allows to enter the greenhouse have been studied in detail in several review papers since the beginning of the greenhouse industry in Northern Europe (Nisen, 1969; Nisen & Dogniaux, 1975; Nijskens *et al.*, 1985), in Japan (Kozai *et al.*, 1978) and in Mediterranean climatic conditions (Critten, 1983; Papadakis *et al.*, 2000; Wang & Boulard, 2000a). However, because of the diversity of the elementary processes involved in convection in greenhouses, only sketch studies have been devoted to convective transfers. As the greenhouse industry migrated gradually from the Northern regions of Europe towards lower latitudes where greenhouses need to be cooled throughout the year, ventilation flux becomes increasingly crucial and a variety of greenhouses have been constructed to meet the requirements of commercial agricultural production in warm regions. The best way to meet the needs of commercial agricultural production is an optimal management of the greenhouse climate. It should include the effective use of solar energy, air and soil heating, ventilation and cooling, humidity control, CO_2 enrichment, nutrient supply and so on. An essential process is the air exchange between the inside and outside of the greenhouse. It directly affects the

Notation

A	area of the opening, m^2
A'	area of the ventilator, m^2
b	slope of temperature profiles, $K\,m^{-1}$
B, B'	constants
c	specific humidity or gas concentration in air, $kg\,kg^{-1}$
$\bar{C}$	average wind coefficient
C'	fluctuating wind coefficient
C_d	discharge coefficient
C_p	specific heat of air, $J\,kg^{-1}\,K^{-1}$
C_t	overall wind coefficient
C_w	wind coefficient
D_a	saturation deficit of air, Pa
E	crop transpiration density, $kg\,m^{-2}\,s^{-1}$
E'	crop transpiration rate, $kg\,s^{-1}$
F	supply or removal rate of tracer gas, $m^3\,s^{-1}$
g	gravitational acceleration, $m\,s^{-2}$
$\mathbf{g}$	gravitational acceleration vector
G	ventilation function
Gr	Grashof number
h	height of the centre of inflow area, m
h'	vertical distance between the centres of two openings, m
h_c	heat transfer coefficient, $W\,m^{-2}\,K^{-1}$
h_0	height of the neutral plane for pressure, m
H	opening height, m
k	thermal conductivity of air, $W\,m^{-1}\,K^{-1}$
l_l	leaf area index
L	characteristic length, m
Le	Lewis number
m	constant
n	constant
N	air exchange rate, s^{-1}
Nu	Nusselt number
p	constant
$P, \Delta P$	pressure, pressure drop, Pa
$\overline{\Delta P}$	mean pressure drop, Pa
$\Delta P'$	fluctuating pressure drop, Pa
Pr	Prandtl number
P_v	shading factor
q	heat flux density, $W\,m^{-2}$
Q	airflow rate, $m^3\,s^{-1}$
r_a	aerodynamic resistance, $s\,m^{-1}$
r_l	stomatal resistance, $s\,m^{-1}$
Ra	Rayleigh number
Re	Reynolds number
Ri	Richardson number
R_g	global solar radiation density inside the greenhouse, $W\,m^{-2}$
S	greenhouse section area perpendicular to the mean ventilation flux, m^2
t	time, s
$T, \Delta T$	temperature, temperature difference, K
u	air speed, $m\,s^{-1}$
$\boldsymbol{u}$	velocity vector
U	characteristic speed, $m\,s^{-1}$
V	greenhouse volume, m^3
w	width of the ventilator, m
ΔW	head loss, $J\,kg^{-1}$
z	vertical co-ordinates or vertical position in vents, m
α	vent opening angle, deg
β	thermal expansion coefficient, K^{-1}
δ	slope of saturated water vapour pressure versus temperature, $Pa\,K^{-1}$
γ	psychrometric constant, $Pa\,K^{-1}$
λ	latent heat of water evaporation, $J\,kg^{-1}$
ρ	fluid density, $kg\,m^{-3}$
ζ	pressure drop coefficient

Subscripts

a	air
bot	bottom
in	inside
l	leaf
out	outside
r	roof opening
s	side opening
th	thermal
top	top
v	vegetation
w	wind
0	reference

Superscripts

*	saturation

transport of sensible heat, water vapour and CO_2 to or from the interior air. Therefore, an exact understanding of the mechanisms of air exchange can be used to control air temperature and CO_2 concentration and to lower excessive humidity caused by plant transpiration.

Up to now, simplified schemes such as the perfectly stirred approach (Udink ten Cate, 1980) have been employed to model heat and mass transfers. It is assumed that temperature, humidity and CO_2 content are uniform inside the greenhouse. A single-species 'big leaf' model for the plant canopy is used to characterize the sensible and latent heat exchanges with inside air. Heat and mass transfer coefficients between air and solid surfaces (leaf, walls, roof, soil and heat exchangers) are calculated from appropriate correlation formulae derived from well-documented data on flat plates subjected to tangential air flows in various convective regimes. The description of the ventilation process also agreed well with this simplified approach because airflow through open ventilators is linked to pressure difference across the opening due to buoyancy and wind forces. The Bernoulli equation was used to express these mechanisms and values of the semi-empirical parameters of the equation were either derived from direct determination of the discharge coefficients or by *in situ* determination (by fitting an overall coefficient of wind efficiency on ventilation to measured air exchange rate).

These theoretical and experimental approaches are compatible and associated with global heat and mass transfer balances, and help determine the mean greenhouse climate and in dimensioning climate control systems such as heating, cooling, humidification, dehumidification (Van Meurs & Stanghellini, 1989; Issanchou, 1991; Jolliet & Bailey, 1994) and CO_2 enrichment (Bailey *et al.*, 1997). This review takes into account all convective transfers involved in greenhouses and a presentation of the experimental and modelling techniques of convective transfers will be provided based on the homogeneity of inside air.

2. Fluid flow equations based on the homogeneity hypothesis

2.1. *Macro-model*

The fluid flow equations constitute a micro-model that help describe with precision the dynamic behaviour of each point in the flow. In greenhouse ventilation situations, some assumptions can be taken into account in order to determine a model that matches the spatial domain of interest. These assumptions are:

(a) steady-state flow—variations in flow conditions (*e.g.* external wind) are considered to be negligible for an extended period of time;
(b) inviscid flow—the diffusive effects of viscosity are limited to restricted areas (boundary layers, wakes, *etc.*) and viscous effects are therefore neglected in this model;
(c) homogeneous flow—the physical properties and the flow velocity field are considered uniform in each constitutive part of the domain of interest (irrotational flow); and
(d) gravitational flow—the resultant of the external forces is limited to the vertical gravitational force.

When applied to fluid flow equations, these assumptions lead to the determination of a macro-model of flow with significant simplification of the equations.

2.2. *Bernoulli equation*

Taking into account assumptions (a)–(d), the momentum equation for the fluid for each location in the fluid flow (Bird *et al.*, 1965) becomes

$$\tfrac{1}{2}\rho \,\mathrm{grad}\, u^2 = -\mathrm{grad}\, P + \rho \mathbf{g} \tag{1}$$

where ρ is the fluid density, u is the speed of fluid, *i.e.* the modulus of the velocity vector $\mathbf{u}$, P the pressure and $\mathbf{g}$ the gravitational acceleration vector.

When integrated to a streamline joining two points A and B, with vertical co-ordinates z_A and z_B, Eqn (1) becomes

$$P_A + \rho g z_A + \tfrac{1}{2}\rho u_A^2 = P_B + \rho g z_B + \tfrac{1}{2}\rho u_B^2 = constant \tag{2}$$

This equation is the best-known form of the *Bernoulli equation* for inviscid flows. It expresses the conservation of mechanical energy as three constitutive parts: pressure energy, potential energy and kinetic energy (all these terms have units of work per volume unit). Note that according to the hypothesis of irrotational flow, this relation is also valid for expressing the conservation of energy between two sections perpendicular to the flow.

The assumption of an inviscid fluid leads to the absence of diffusive effects in the flow (irrotational behaviour) and any energy dissipation due to shear-stress work in the flow. Energy dissipation is an irreversible process that prevents absolute conservation of energy. It is very important to take this process into account in most practical situations where fluid viscosity cannot be ignored. The easiest way of taking this term into account in Eqn (2) is to express the energy lost between points A and B as a head loss ΔW_{AB}:

$$P_A + \rho g z_A + \tfrac{1}{2}\rho u_A^2 = P_B + \rho g z_B + \tfrac{1}{2}\rho u_B^2 + \rho \Delta W_{AB} = constant \tag{3}$$

The experimental determination of head losses is generally done by pressure measurements.

A pressure drop ΔP_{AB} is used which gives the following equation:

$$P_A + \rho g z_A + \tfrac{1}{2}\rho u_A^2 = P_B + \rho g z_B + \tfrac{1}{2}\rho u_B^2 + \Delta P_{AB} = constant \tag{4}$$

This pressure drop ΔP_{AB} can be related to the disposable kinetic energy with

$$\Delta P_{AB} = \tfrac{1}{2}\zeta \rho u_A^2 \tag{5}$$

where ζ is the pressure drop coefficient (dimensionless) the value of which depends on flow conditions (flow regime, geometry, *etc.*).

2.3. *Heat and mass transfer between surfaces and airflow*

In greenhouse conditions, two kinds of solid surfaces occur: the greenhouse walls, generally made of glass or plastic, and the leaves of the crop. In the vicinity of these surfaces, fluid flow cannot be considered to be inviscid. Viscous effects are responsible for the creation of a dynamic boundary layer where momentum is dissipated by friction. If there is a temperature difference between the walls or the leaves and the flow, a thermal boundary layer that is closely linked to the dynamic boundary layer is formed. In addition, if there is a difference in humidity between the leaves and the airflow, a boundary layer formed by vapour concentration superimposes on the two previous boundary layers in the immediate vicinity of the leaf. The determination of the temperature and concentration gradients helps to determine accurately the heat or mass transfer rates between the air and the walls or leaves. Different correlation formulae are deduced from fluid flow equations, adjusted to take into consideration the boundary layer conditions, with the help of different characteristic groups (Schlichting, 1955). These groups of numbers, also called dimensionless or adimensional numbers, are defined with the help of the characteristic flow conditions. Hence, the macro-model is still valid for the determination of energy and mass balance of walls and leaves in the domain of interest.

Standard characteristic numbers for the greenhouse situation are listed in Table 1. The flow regime depends on the value of the Reynolds or Rayleigh number, Re and Ra: for values of Re lower that 5×10^4 or values of Ra less than 10^8, fluid flow is laminar but becomes turbulent for greater values. For greenhouse crops, the characteristic order of magnitude for leaf length L_l is generally 10^{-1} m, and 10^{-1} m s^{-1} for characteristic speed U. The flow regime thus remains laminar over the entire surface of the leaves. In this situation, the mean value of the Nusselt number Nu is directly deduced from the boundary layer theory (see Section 3.1.1). Furthermore, Schuepp (1993) has shown that the relations for laminar flow on a leaf surface are valid for situations where the boundary layer becomes turbulent on the leeward part of the leaf surface. These relations are not valid for the entire crop because of the interaction between the boundary layers generated by the different leaves.

For free convection situations at the walls or leaves, the spatial position of the solid plane must be taken into account in order to determine the relations for heat and mass transfers. Correlation formula deduced from theory or from experience are available for most situations

Table 1
Definition of the characteristic dimensionless numbers adapted for greenhouse configurations

Characteristic group	*Wall*	*Leaf*
Reynolds number Re *(forced convection)*	$\mathrm{Re} = \dfrac{UL_p}{\nu}$	$\mathrm{Re} = \dfrac{U_{in}L_l}{\nu}$
Grashof number Gr *(free convection)*	$\mathrm{Gr} = \dfrac{g\beta\Delta T_p L_p^3}{\nu^2}$	$\mathrm{Gr} = \dfrac{g\beta\Delta T_l L_l^3}{\nu^2}$
Rayleigh number Ra *(free convection)*	$\mathrm{Ra} = \dfrac{g\beta\Delta T_p L_p^3}{\nu\alpha} = \mathrm{Gr\,Pr}$	$\mathrm{Ra} = \dfrac{g\beta\Delta T_l L_l^3}{\nu\alpha} = \mathrm{Gr}\,Pr$
Richardson number Ri *(mixed convection)*	$\mathrm{Ri} = \dfrac{g\beta\Delta T_p L_p}{U^2} = \dfrac{\mathrm{Gr}}{\mathrm{Re}^2}$	$\mathrm{Ri} = \dfrac{g\beta\Delta T_l L_l}{U^2} = \dfrac{\mathrm{Gr}}{\mathrm{Re}^2}$
Prandtl number Pr *(heat transfer)*	$Pr = \dfrac{\nu}{\alpha}$	$Pr = \dfrac{\nu}{\alpha}$
Nusselt number Nu *(heat transfer)*	$\mathrm{Nu} = \dfrac{h_{cp}L_p}{k}$	$\mathrm{Nu} = \dfrac{h_{cl}L_l}{k}$
Schmidt number Sc *(mass transfer)*	—	$\mathrm{Sc} = \dfrac{\nu}{d_{vap}}$
Sherwood number Sh *(mass transfer)*	—	$\mathrm{Sh} = \dfrac{h_{vap}L_l}{d_{vap}}$
Lewis number Le *(heat and mass transfer)*	—	$\mathrm{Le} = \dfrac{\alpha}{d_{vap}} = \dfrac{\mathrm{Sc}}{Pr}$

Note: d_{vap}, water vapour diffusivity; g, gravitational acceleration; h_{cl}, mean convective heat transfer coefficient on the leaf; h_{cp}, mean convective heat transfer coefficient on the wall; h_{vap}, mean convective vapour transfer rate on the leaf; k, thermal conductivity of air; L_p, characteristic length of the wall; ΔT_l, characteristic temperature difference between air and leaf; ΔT_p, characteristic temperature difference between air and wall; U, characteristic velocity; U_{in}, characteristic velocity in the greenhouse; L_l, characteristic length of the leaf (Schuepp, 1993); α, thermal diffusivity of air; β, thermal expansion coefficient; ν, kinematic viscosity of fluid.

(Bejan, 1984; Holman, 1986). The relations for situations relevant to heat and mass transfers from greenhouse walls and from the crop are given in detail in Section 3.1.1.

3. Experimental and modelling techniques

Experimental methods are essential to understand the air exchange between the inside and outside of greenhouses and to validate the results of theoretical calculations. A few measuring techniques have been developed such as the tracer gas method, pressure field and wind speed measurement methods while the use of computational methods has now become popular due to the rapid development of numerical techniques. Computational methods include fundamental calculations, multiple regression models from indirect measurements, and pressure distribution and energy balance methods. Compared with the experiments, computational methods may be performed more rapidly since they are based on very simple measurements. Also they are not subject to the problem of scaling effect and may be less expensive as the cost of full-scale experiments is steadily rising. A detailed discussion of these methods will be carried out here for the case of the perfectly stirred tank approach.

3.1. *Determining convective exchanges*

Convection heat transfer is one of the most important mechanisms of heat loss in greenhouses. Convective exchange occurs between the cover, the soil, the vegetation and the interior air and between the cover and the exterior air. It is clear that the process of heat transfer is governed by a combination of forced convection (due to the wind pressure) and free convection, due to buoyancy forces caused by temperature differences between the solid surfaces of the walls, the soil, the plants and the air. These two convection modes are dependent on greenhouse type, outside climate and ventilation conditions. In well-ventilated greenhouses, forced convection is dominant, due to strong air movement. In tightly closed greenhouses, due to very low interior air velocities, free convection is the most common process.

The convective heat flux density is proportional to the temperature difference ΔT between surfaces (*e.g.* the cover) and air (Holman, 1986). The proportionality is generally given by the convective heat transfer coefficient h_c. The expression for the convective heat flux density q thus becomes

$$q = h_c \Delta T \tag{6}$$

An experimental estimation of a convective heat transfer flux may be extremely complicated and not possible to perform for many surface shapes. An empirical approach is thus used in the experiments so that the convective heat transfer coefficients can be linked to well-known non-dimensional groups.

3.1.1. *Fundamental calculation of convection heat transfer*

The heat transfer coefficient h_c depends on convection modes and flow types (laminar or turbulent) and is deduced from the appropriate Nusselt number Nu according to the laminar boundary layer theory (Holman, 1986):

$$h_c = \frac{k\,\mathrm{Nu}}{L} \tag{7}$$

where L is the characteristic length of the solid surface and k is the thermal conductivity of the air. The characteristic length is related to the shape of the object and measures the length of the surface covered by the laminar flow. Many researchers (De Halleux *et al.*, 1991; Pieters *et al.*, 1994; Wang & Boulard, 2000b) proposed characteristic lengths of a few centimetre for bare soil, a few metres for soil covered with plastic film, and 10 cm for a tomato leaf surface. For the greenhouse cover, the roof slope length (length of the glass panes) is generally used.

In free convection, heat transfer takes place through the fluid motion induced by temperature gradients. In such cases, Nu may be expressed as a function of the Grashof and Prandtl numbers Gr and Pr (Monteith, 1973):

$$\mathrm{Nu} = B(\mathrm{Gr}\,\mathrm{Pr})^n \tag{8}$$

where B and n are constants depending on geometry and flow type and are determined with the help of experiments. In forced convection, the Nusselt number as a function of Reynolds and Prandtl numbers is usually expressed as (Monteith, 1973; Holman, 1986)

$$\mathrm{Nu} = B'\,\mathrm{Re}^p\,\mathrm{Pr}^m \tag{9}$$

where B', p and m are constants that depend on the geometry and flow type. In many situations, the heat transfer is by mixed convection due to the equal importance of the external pressure field and buoyancy effects (Hieber, 1973; Papadakis *et al.*, 1992; Lamrani *et al.*, 2001).

The Nusselt number expression for laminar and turbulent flows in free and forced convection modes along a flat plate can be found in Table 2 according to Monteith (1973) and Campbell (1977). Other constants in Eqns (8) and (9) from Table 2 are also found in the literature (Fujii & Imura, 1972; Raithby & Hollands, 1975; Holman, 1986) but they are very close to each other. In greenhouses, heat convection occurring between the air and vegetation, cover and soil surface can be calculated directly when non-dimensional groups are

Table 2
Nusselt number along a flat plate according to Monteith (1973) and Campbell (1977)

Convective mode	*Laminar flow*	*Turbulent flow*
Free convection	$\mathrm{Nu} = 0{\cdot}54(\mathrm{Gr\,Pr})^{1/4}$	$\mathrm{Nu} = 0{\cdot}14(\mathrm{Gr\,Pr})^{1/3}$
Forced convection	$\mathrm{Nu} = 0{\cdot}67\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}$	$\mathrm{Nu} = 0{\cdot}036\mathrm{Re}^{4/5}\,\mathrm{Pr}^{1/3}$

Note: Gr, Grashof number; Nu, Nusselt number; Pr, Prandtl number; Re, Reynolds number.

replaced by the above relations with the parameter values for air at 20°C. In the case of free convection and laminar flow between vegetation and air, a horizontal leaf with two faces exposed to the air contributes 1·5 times the normal convective flux whether it is warmer or colder than the air as the heat transfer is only about half as efficient for cooled flat surface facing up or warm surface facing down (Campbell, 1977; De Halleux *et al.*, 1991; Wang, 1998). In the turbulent flow mode, the flow remains turbulent on the side of the leaf where convection is reinforced but becomes laminar on the side where flow is impeded. However, under forced convection, the exchange is twice as effective as the normal convective flux, because simultaneous convective heat losses occur on both sides of the leaves.

In order to determine the convective heat transfer coefficient, a criterion has to be defined for identifying the convective mode (forced or free) and the type of flow (laminar or turbulent). The Richardson number Ri (Table 1) suggests a criterion to distinguish free from forced convection. When Re^2 is much larger than Gr, buoyancy forces are negligible and forced convection is dominant while the inverse condition results in free convection. The critical values of Ri are given in Table 3 for horizontal flat plates similar to the cover, soil and vegetation layers. Distinction between laminar and turbulent flows is based on Gr for free convection and Re for forced convection. The criterion corresponding to the air at 20°C is also provided in Table 3 for convenience.

Table 3 is useful for setting the criteria for the various combinations of the modes of convective heat exchange and types of airflow between air and cover, vegetation as well as the soil surface. After the convection mode and flow type have been determined, the heat transfer coefficients can be calculated for the cover, the vegetation and the soil surface in greenhouses (Table 2). In practical situations, researchers may obtain heat transfer coefficients for specific greenhouse geometry.

3.1.2. *Cover-air and soil-air exchanges*

Convective heat exchange between cover surface and outside air is generally considered to be forced and turbulent heat transfer, mainly influenced by external wind speed. Several experimental studies to determine the convective coefficient on the outside cover were conducted in different greenhouses under specific *in situ* conditions. The empirical formulae for convective coefficients are shown in Table 4. In order to compare the difference in these coefficients, the value of convective coefficients is plotted as a function of external wind speed by fixing the characteristic length at 2 m and a temperature difference of 2°C (*Fig. 1*). The values of convective heat coefficients are close to each other when external speed is below 3 m s^{-1} except in the Garzoli and Blackwell (1987) formula. Over a given range of the wind speed, the estimation values from Kanthak (1970), Tantau (1975), Watmuff (1977) and De Halleux (1989) are similar. Some of authors have added a constant term to take into account the temperature effect when external wind speed is low. The significant difference observed was probably due to protocol differences used to measure wind speed. But unfortunately no mention of it was made in their research.

Table 5 shows the convective coefficient for the inner cover surface at a normal air temperature in green-

Table 3
Criteria for the determination of convection modes and flow types according to Monteith (1973) and Campbell (1977)

Choice criterion of convection mode		*Convective mode*	*Laminar flow*	*Turbulent flow*
General criterion	$\mathrm{Ri} = \dfrac{\mathrm{Gr}}{\mathrm{Re}^2} < 0{\cdot}1$	Forced convection	$\mathrm{Re} < 5 \times 10^4$	$\mathrm{Re} > 5 \times 10^4$
Criterion for 20°C air	$\dfrac{L\Delta T}{U^2} < 3$		$UL < 0{\cdot}75$	$UL < 0{\cdot}75$
General criterion	$\mathrm{Ri} = \dfrac{\mathrm{Gr}}{\mathrm{Re}^2} > 16$	Free convection	$\mathrm{Gr} < 10^8$	$\mathrm{Gr} > 10^8$
Criterion for 20°C air	$\dfrac{L\Delta T}{U^2} > 484$		$L^3\Delta T < 0{\cdot}63$	$L^3\Delta T > 0{\cdot}63$

Table 4
List of empirical formulae for convective heat transfer coefficients between the outer cover surface and the air according to different authors

Heat transfer coefficient h_c *($W m^{-2} K^{-1}$)*	*Conditions*	*Source*
$3{\cdot}49U$	20 m by 10 m greenhouse	Kanthak (1970)
$5{\cdot}6\dfrac{U^{0{\cdot}8}}{L^{0{\cdot}2}}$	Turbulence and Ra > 10^5	Tantau (1975)
$2{\cdot}8 + 3{\cdot}0U$	Solar collectors	Watmuff (1977)
$2{\cdot}8 + 1{\cdot}2U$	Venlo-type greenhouse ($U \leqslant 4\,m\,s^{-1}$)	Bot (1983)
$1{\cdot}32\Delta T^{0{\cdot}25}U^{0{\cdot}8}$	Tunnel-type greenhouse	Kittas (1986)
$7{\cdot}2 + 3{\cdot}84U$	Plastic greenhouse	Garzoli and Blackwell (1987)
$5{\cdot}96\dfrac{U^{0{\cdot}8}}{L^{0{\cdot}2}}$	Large-scale greenhouse	De Halleux (1989)
$0{\cdot}95 + 6{\cdot}76U^{0{\cdot}49}$	Polyethylene-covered greenhouse ($U \leqslant 6{\cdot}3\,m\,s^{-1}$)	Papadakis *et al.* (1992)

Note: L, characteristic length; Ra, Rayleigh number; Re, Reynolds number; U, characteristic speed; ΔT, characteristic temperature difference.

houses. Compared with Table 2, heat transfer in greenhouses is considered both in the free and turbulent convection modes. However, the derived convective heat transfer coefficient varies greatly, especially in the empirical Lamrani *et al.* (2001) model because it was established for a half-scale greenhouse (*Fig. 2*). All the models in Table 5 are suitable for describing heat convection in closed or in open greenhouses with a low ventilation rate, but the lack of adequate information on external wind speed and greenhouse vent openings may pose a problem. Convective heat transfer coefficients depend on temperature differences and air speed around the greenhouse components studied. Interior air speed is a function of the ventilation rate which itself depends primarily on the vent opening angle and external wind speed (Boulard & Baille, 1995; Wang *et al.*, 1999). If the greenhouse is well ventilated, interior air speed increases with external wind speed and heat transfer becomes forced convection.

The calculation of natural ventilation in greenhouses (see below) has now been well defined. The interior air speed may be determined according to the airflow

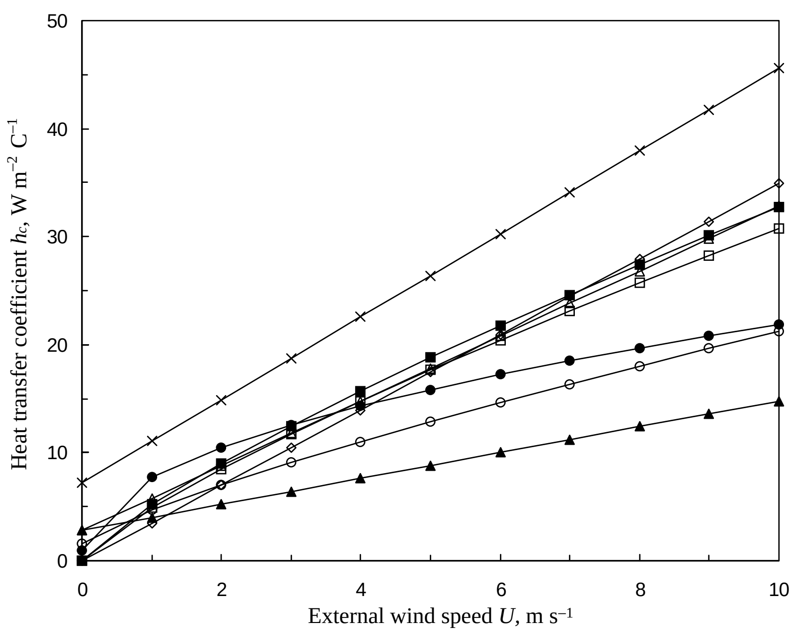

Fig. 1. Variation of the convective heat transfer coefficient h_c for the outside cover surface as a function of the external wind speed according to different authors: -◊-, Kanthak (1970); -□- Tantau (1975); -△-, Watmuff et al. (1977); -▲-, Bot (1983); -▲-, Kittas (1986); -×-, Garzoli and Blackwell (1987); -■-, De Halleux (1989); -●-, Papadakis et al. (1992)

Table 5
List of empirical formulae for convective heat transfer coefficients between inner cover surface and air according to different authors

Heat transfer coefficient h_c ($W m^{-2} K^{-1}$)	*Conditions*	*Source*
$1{\cdot}93\Delta T^{0{\cdot}33}$	20 m by 10 m greenhouse	Kanthak (1970)
$1{\cdot}247\Delta T^{0{\cdot}33}$	Vertical wall	Tantau (1975)
$3{\cdot}3\Delta T^{0{\cdot}33}$	Screened greenhouse	Stoffers (1985)
$4{\cdot}3\Delta T^{0{\cdot}25}$	Tunnel-type greenhouse	Kittas (1986)
7·2	Plastic greenhouse	Garzoli and Blackwell (1987)
$1{\cdot}86\Delta T^{0{\cdot}33}$	Large-scale greenhouse	De Halleux (1989)
$2{\cdot}21\Delta T^{0{\cdot}33}$	Small polyethylene-covered greenhouse	Papadakis *et al.* (1992)
$2{\cdot}97\Delta T^{0{\cdot}33}$	Screened greenhouse	Miguel *et al.* (1998)
$8{\cdot}0\Delta T^{0{\cdot}33}$	Confined greenhouse	Lamrani *et al.* (2001)

Note: ΔT, characteristic temperature difference.

pattern in the greenhouse studied. Wang *et al.* (1999) used the following equation to estimate the interior air speed U in a double-span plastic greenhouse:

$$U = \frac{Q}{S} \tag{10}$$

where S is the greenhouse section area perpendicular to the direction of the mean ventilation flux Q. The average air speed calculated agreed to a large extent with the experimental value measured by a networked sonic anemometer system. This equation may be successfully applied to models of crop transpiration (Boulard & Wang, 2000). The average air speed in greenhouses is useful in calculating the convective coefficient for soil and cover surfaces according to Tables 2 and 3.

There have been fewer studies on heat exchange between the soil surface and air. In actual greenhouses, a large part of the soil is covered by plants and heat losses from the soil surface are small. The daily mean value of the convection heat flux density between the soil and air is of the order of 10 W m^{-2}, owing to night inversion of the soil temperature gradient. This explains why heat losses from the soil are often neglected in static models. In a dynamic model, however, heat convection from the

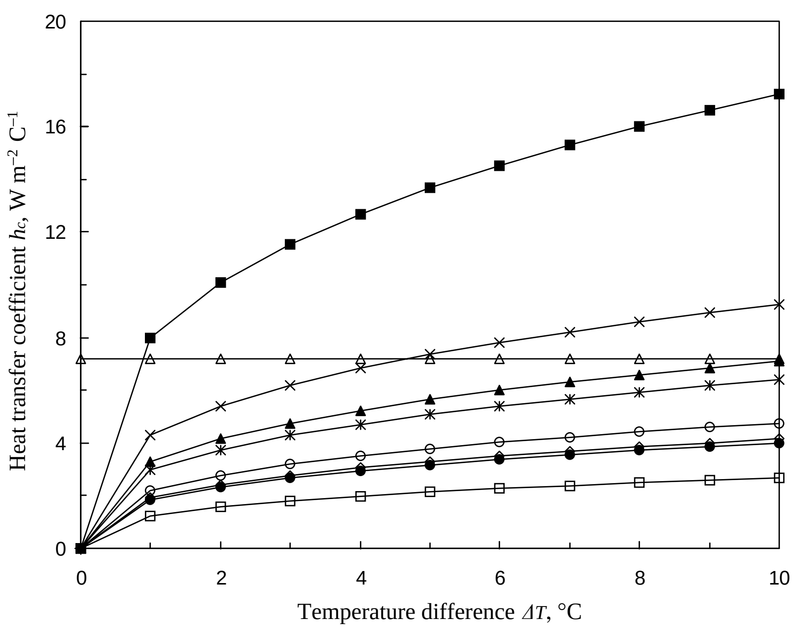

Fig. 2. Variation of the convective heat transfer coefficient h_c for the inside cover surface as a function of the temperature difference between the cover and the air according to different authors: -◊-, Kanthak (1970); -□-, Tantau (1975); -▲-, Stoffers (1985); -×-, Kittas (1986); -Δ-, Garzoli and Blackwell (1987); -●-, De Halleux (1989); -Δ-, Papadakis et al. (1992); -✳-, Miguel et al. (1998); -■-, Lamrani et al. (2001)

Table 6
List of empirical formulae for convective heat transfer coefficients between soil surface and air in greenhouses according to different authors

Heat transfer coefficient h_c ($W m^{-2} K^{-1}$)	*Conditions*	*Source*
$3{\cdot}4\Delta T^{0{\cdot}33}$	Screened greenhouse	Stoffers (1985)
$10{\cdot}0\Delta T^{0{\cdot}33}$	Bare soil	Silva (1988)
$1{\cdot}86\Delta T^{0{\cdot}33}$	Large-scale greenhouse	De Halleux (1989)
$5{\cdot}2\Delta T^{0{\cdot}33}$	Heated floor surface	Lamrani *et al.* (2001)

Note: ΔT, characteristic temperature difference.

soil surface has to be taken into account. Table 6 shows several empirical models obtained from experiments in greenhouses. These models are fitted by free convection and turbulent flow equations with different coefficients.

3.1.3. *Plant–air exchanges*

Heat convection and transpiration processes play a crucial role in the energy balance of leaves. Heat convection between vegetation and interior air is largely dependent on the value of crop aerodynamic resistance. The evaporation rate depends on two important factors: net radiation and the vapour pressure deficit of the air. To pass from the leaf into air, the vapour has to overcome stomatal and aerodynamic resistances.

Strictly speaking, natural crops cannot be considered homogeneous in full-scale greenhouses. However, a number of studies on vegetation transpiration show that treating greenhouse crops as homogeneous porous volumes is acceptable and helpful in understanding the complex exchange process between plants and the environment (Bot, 1983; Stanghellini, 1987; Yang *et al.*, 1990). This simplification is supported by Yang *et al.* (1990) who found that the temperature difference between the top and bottom cucumber leaves was small and by Boulard *et al.* (1991) who found that both single and a multi-layer models gave similar results for canopy transpiration. The convective heat transfer density between vegetation (subscript v) and interior air (subscript in) is given by

$$q_{v,in} = \frac{\rho C_p \Delta T}{r_a} \tag{11}$$

where C_p is the specific heat of air at constant pressure and r_a the aerodynamic resistance. Aerodynamic resistance is related to the convective heat transfer coefficient and is thus a function of the Nusselt number. Table 7 gives a list of Nusselt numbers Nu for heat convection between vegetation and air obtained by different researchers. The aerodynamic resistance obtained from Nu values varied significantly, probably due to different experimental conditions. The heat transfer process between vegetation and air may be free, forced or mixed convection modes depending on the vegetation schemes and the interior air speed around the leaves.

Crop transpiration density E is deduced from the aerodynamic resistance r_a and the stomatal resistance r_l using either the direct (physical) formula

$$E = \frac{\mathrm{Le}^{1/3}\, P_v l_l \rho (c_v^* - c_{in})}{r_a + r_l} \tag{12}$$

or the Penman–Monteith formula:

$$E = \frac{R_g r_a \delta + 2 l_l \rho C_p D_a}{\lambda((\delta + \gamma) r_a + 2\gamma r_l)} \tag{13}$$

where c_{in} is the specific humidity of the interior air, c_v^* is the specific humidity of saturated air at the temperature of the vegetation, R_g is the global solar radiation density inside the greenhouse, l_l is the leaf area index expressed as leaf area per unit ground area, Le is the non-dimensional Lewis number defined in Table 1, P_v is the shading factor of the vegetation (proportion of greenhouse ground area covered by the vegetation), γ is the psychrometric constant, δ is the slope of saturated vapour pressure to temperature, D_a is the saturation pressure deficit of air and λ the latent heat of water evaporation.

Table 7
List of empirical formulae for the Nusselt number for heat convection between the vegetation and the air according to different authors

Nusselt number Nu	*Conditions*	*Source*
$0{\cdot}37\mathrm{Gr}^{0{\cdot}25}$	Artificial and horizontal leaves	Parkhurst *et al.* (1968)
$0{\cdot}139\mathrm{Gr}^{0{\cdot}25}$	Artificial porous discs	Morrison and Barfield (1981)
$0{\cdot}291\mathrm{Re}^{0{\cdot}5}$	Artificial porous discs for forced flow	Morrison and Barfield (1981)
$0{\cdot}25\mathrm{Gr}^{0{\cdot}3}$	At average greenhouse climates	Stanghellini (1987)
$0{\cdot}14\mathrm{Gr}^{0{\cdot}33}$	Greenhouse tomato	De Halleux (1989)

Note: Gr, Grashof number; Re, Reynolds number.

The stomatal resistance of tomato crops under greenhouse conditions had been measured and modelled by several authors (Stanghellini, 1987; Boulard *et al.*, 1991; Jolliet & Bailey, 1994, Papadakis *et al.*, 1994). Stomatal resistance varied mainly as a function of internal global solar radiation and partly as a function of greenhouse air temperatures above 30°C and of saturation deficits above 10^3 Pa. In spring, when the models were tested, air temperature and saturation deficit sometimes exceeded these threshold values.

3.2. *Measuring ventilation rate*

The rate of air exchange between the inside and the outside of the greenhouse is measured by the decay in any one of the variables transported (enthalpy, concentration of a chemical substance) by the airflow.

3.2.1. *Heat balance*

The energy balance method of determining ventilation rates uses either static or dynamic models. The first static models approximated energy consumption to compensate for total thermal losses (Morris, 1964; Chiapale *et al.*, 1981). These models had limited accuracy but their virtue lay in their simplicity. Later Bailey (1977), Hurd and Sheard (1981) as well as Breuer and Short (1985) improved these static models by taking into account the contribution of solar energy but these later models were not more accurate. An example of a recent improved static model used to estimate the ventilation rate of a greenhouse under stationary conditions is given by Fernandez and Bailey (1992). Comparison with ventilation rates measured using tracer gas (Fernandez & Bailey, 1992) shows that calculated rates were higher, due to an underestimation of the energy stored in the system and that the precision of measurement increased with the length of the time scale and a minimization of the transient energy exchange term.

The dynamic models for greenhouses can predict both energy needs and the greenhouse interior climate. A number of dynamic models, each better than the preceding one, have been developed (Takakura *et al.*, 1971; Kindelan, 1980; Bot, 1983; De Halleux *et al.*, 1991). Wang (1998) exploited these two last dynamic model recently to deduce the ventilation rate of a large Venlo-type greenhouse and Teitel and Tanny (1999) used the transient behaviour of temperature and humidity over a short period of time (less than 1 h) to measure the ventilation rate.

3.2.2. *Mass balance*

The tracer gas techniques used for ventilation measurements are based on the mass balance of natural and artificial constituents of greenhouse air. Assuming a uniform gas distribution in the greenhouse and a perfect mixing with air, the following relation holds:

$$V\frac{dc_{in}}{dt} = -Q(t)(c_{in}(t) - c_{out}) \pm F_{in}(t) \qquad (14)$$

where Q is the ventilation flow rate, V the greenhouse volume, t the time, c_{in} and c_{out} the inside and outside concentration of tracer gas and $F_{in}(t)$ the rate of supply or removal of the tracer gas within the greenhouse.

Selection of the tracer gas is very important. The gas should be inert, non-toxic, non-flammable with a molecular weight close to the average weight of air components, and easy to measure at low concentrations. Many gases such as SF_6, CH_4, CO_2, H_2, N_2O, argon 41 and krypton 85 have been used as tracer gases. The two most frequently used are CO_2 and N_2O. The latter meets all the above requirements. CO_2 may be used in a non-cropped greenhouse, but it is necessary to measure the CO_2 concentration in external air and its release rate from the soil. In a cropped greenhouse, N_2O is not influenced by photosynthesis nor by plant respiration.

Water vapour may also be used as a tracer gas (Demrati *et al.*, 2001). The air exchange rate (measured by the N_2O decay rate method or the CO_2 continuous enrichment method) was compared with the water balance of a 400 m^2 bi-span greenhouse by Boulard and Draoui (1995).

3.2.3. *Decay method using nitrous oxide*

The decay or dynamic tracer gas method is certainly the most common (Okada & Takakura, 1973; Ruther, 1985; De Jong, 1990; Fernandez & Bailey, 1992; Boulard & Draoui, 1995; Baptista *et al.*, 1999) and requires very simple devices. After an initial release of the tracer gas and the homogenization and stabilization of inside content, the decrease in concentration $c_{in}(t)$ is monitored. As $F_{in}(t)$ and c_{out} are equal to zero (the natural N_2O concentration in the atmosphere $\approx 0{\cdot}3$ p.p.m.), integration of Eqn (14) gives

$$c_{in}(t) = c_{in}(t_0)e^{-N(t-t_0)} \qquad (15)$$

where $N = 3600\ (Q/V)$ is the air exchange rate.

If the tracer gas is well mixed throughout the volume and also if the ventilation rate is constant during the measurement period, a plot of concentration *versus* time on a semi-log graph gives a straight line as shown by the example of *Fig. 3*. A single determination lasts several minutes with internal tracer concentration measurements performed every second. The slope of the lines gives the ventilation rate.

The different phases of this method are given in *Fig. 4* which illustrates the ventilation rate measurement for a very large scale (5600 m^2) Canarian-type greenhouse

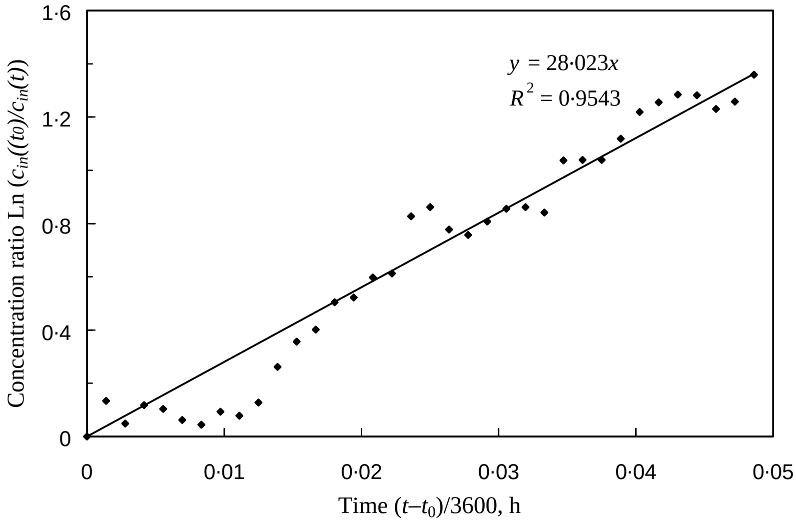

Fig. 3. Development of the logarithm of the ratio of the N_2O concentration $Ln(c_{in}(t)/c_{in}(t_0))$ versus time $(t - t_0)/3600$ during the measurement of air renewal in a large-scale (5600 m^2) canarian-type greenhouse with 100% of side vents opening (windward and leeward); R^2, coefficient of determination (after Fatnassi et al., 2002)

(Fatnassi *et al.*, 2002). Under natural ventilation conditions, it can take several minutes before getting a reasonable homogeneous mixture of the gas in space, the time scale of the decay rate method lasting between 3 min (*Fig. 4*) for a completely open greenhouse with a strong wind ($4 < U < 7\,\mathrm{m\,s^{-1}}$) and 30 min for a closed greenhouse.

3.2.4. *Water vapour and carbon dioxide balances*

3.2.4.1. Water vapour balance. In this method, water vapour is used as a tracer gas. Inside $c_{in}(t)$ and outside $c_{out}(t)$ greenhouse air-specific humidities and greenhouse crop transpiration rate $E'(t)$ are measured to calculate the greenhouse air exchange rate. Assuming uniform humidity conditions in the whole greenhouse volume and considering that evaporation loss from the crop substrate and the soil are negligible, Eqn (14) can be rewritten as follows:

$$\rho\frac{V\mathrm{d}c_{in}(t)}{\mathrm{d}t} = -\rho Q(t)(c_{in}(t) - c_{out}(t)) + E'(t) \qquad (16)$$

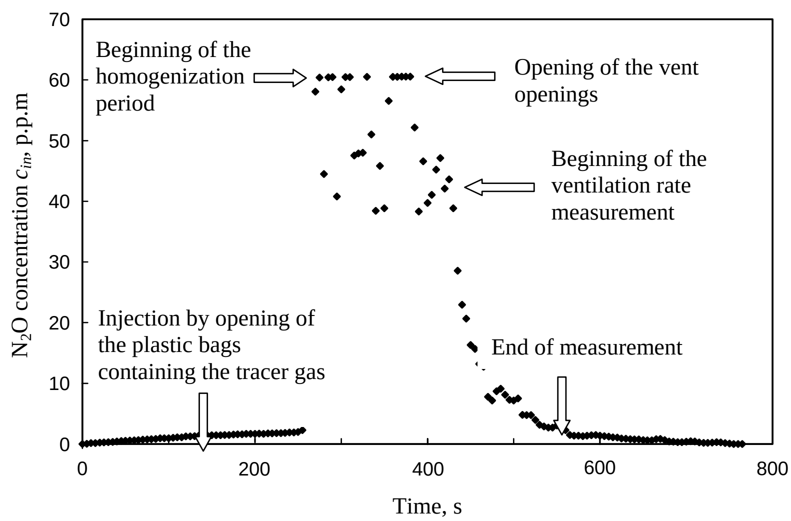

Fig. 4. Development of N_2O concentration versus time during the various phases of the greenhouse air renewal measurement in a very large greenhouse (5600 m^2) where, in order to reduce the injection duration, the gas release is performed by opening large plastic bags inflated with N_2O gas (after Fatnassi et al., 2002)

The ventilation airflow may be directly deduced from

$$Q(t) = \frac{\rho V \, \mathrm{d}c_{in}(t)/\mathrm{d}t - E'(t)}{\rho(c_{out}(t) - c_{in}(t))} \tag{17}$$

3.2.4.2. Carbon dioxide balance. As CO_2 concentration is influenced by photosynthesis and vegetation respiration, the use of CO_2 as a tracer gas requires a non-cropped greenhouse or the use of a correction factor (Nederhoff *et al.*, 1984).

3.2.5. *Constant gas supply method using N_2O*

An alternative method is to supply the tracer gas at a constant rate F_{in}. Under this condition, integration of Eqn (14) from time t_0 to time t_1 gives the mean ventilation rate Q over the time interval as

$$Q = \frac{F_{in}}{c_{in}(t_0) - c_{out}(t_0)} - \frac{V}{(t_1 - t_0)} \mathrm{Ln}\left(\frac{c_{in}(t_1) - c_{out}(t_1)}{c_{in}(t_0) - c_{out}(t_0)}\right) \tag{18}$$

In this method the greenhouse volume V only appears in a correction term that accounts for changes in the amount of tracer gas in the greenhouse air. However, the instrumentation requirement is greater as the rate of tracer gas supply must be measured, but the method is more suitable for automated monitoring. The substantially constant concentration of the tracer gas improves the accuracy compared to the decay method.

3.2.6. *Measurement problems*

Sherman (1990) has analysed tracer gas techniques for measuring ventilation in a single zone like a greenhouse volume. He has shown that mixing problems present a major source of potential error if the tracer gas method is used. As the tracer gas must be uniformly mixed throughout the greenhouse volume, this condition can hardly be verified when the greenhouse is large and when the number of sampling points is limited. In order to minimize errors, auxiliary fans must be used to mix the tracer gas, and air samples from several locations in the greenhouse volume must be analysed. It must also be remembered that air transfer from the sampling points to the infra-red gas analyser can require several tenths of seconds.

Ducarme *et al.* (1994) estimate that the accuracy of the analysis by the tracer gas is about 30% and that the main errors are caused by

(i) exfiltration air re-entering the measurement space at another location;
(ii) non-uniform concentration of the tracer gas due to mixing problems;
(iii) changes in external wind conditions, air temperature and window opening during the measurement process; and
(iv) measurement errors in the equipment itself (10% of full-scale reading).

3.3. *Ventilation models*

3.3.1. *Basic ventilation mechanisms*

The quantity of fluid flowing through a vent can be calculated from the head losses with the help of the Bernoulli equation. If the air speed u is constant across an opening, the pressure drop ΔP across this opening is given by Eqn (5) and leads to

$$\Delta P = \tfrac{1}{2}\zeta\rho u^2 \tag{19}$$

The discharge coefficient C_d is directly defined from the pressure drop coefficient by

$$C_d = \zeta^{-0{\cdot}5} \tag{20}$$

This coefficient is very useful in determining flow conditions in ventilation situations: if the pressure drop ΔP is determined between two points on both sides of a vent, the mean velocity u may be directly deduced from

$$u = \frac{|\Delta P|}{\Delta P} C_d \left(\frac{2}{\rho}|\Delta P|\right)^{0{\cdot}5} \tag{21}$$

where the sign of the ratio $|\Delta P|/\Delta P$ gives the direction of the airflow through the opening.

All ventilation phenomena may be modelled using Eqn (21). Pressure differences that produce the ventilation fluxes may be caused (i) either by a temperature difference between inside and outside air that creates a pressure difference, commonly known as the chimney (stack) effect or (ii) by the wind which creates a pressure difference over the greenhouse (Bailey, 2000).

3.3.2. *Chimney effect*

When considering a non-uniform temperature field in the domain of interest characterized by an opening between both parts of the domain, a flow will occur between the hot and the cold parts of the fluid even in the absence of a pressure gradient due to external conditions (wind). This flow is driven by the fluid density gradient in the domain, leading to vertical buoyancy forces, *i.e.* momentum sources in the momentum equation for the fluid. Homogeneous fluid flow in the macro-model definition cannot be assumed. Nevertheless, the spatial variation of fluid density may be approached with the Boussinesq approximation. Then, considering β the thermal expansion coefficient of an ideal gas,

$$\beta = -\frac{1}{\rho}\left(\frac{\mathrm{d}\rho}{\mathrm{d}t}\right) = \frac{1}{T_0} \tag{22}$$

with T_0 the reference temperature at the bottom of the opening ($z = 0$), the Boussinesq approximation leads to

$$\rho(T(z)) = \rho(T_0)\left(1 - \frac{T(z) - T_0}{T_0}\right) \tag{23}$$

Thus, when considering a vertical temperature profile $T(z)$ on each side of the vent, the pressure difference for each vertical position z in the vent is determined with the hydrostatic formula

$$\Delta P(z) = \Delta P_0 - \int_0^z \rho(T_0)\frac{T(z) - T_0}{T_0} g\,\mathrm{d}z \tag{24}$$

where ΔP_0 is the pressure difference at the bottom of the opening ($z = 0$). The integration of this equation is possible when the temperature profile $T(z)$ is known.

3.3.2.1. Case of a single opening. If we assume, in order to simplify the problem, that the temperature field is homogeneous inside and outside (Bruce, 1982; Timmons *et al.*, 1984; Zhang *et al.*, 1989), then

$$\Delta P(z) = \Delta P_0 - \rho_0 g\frac{\Delta T}{T_0}z \tag{25}$$

where ρ_0 is the fluid density at T_0 and ΔT is the temperature difference between inside and outside. Then, Eqn (21) becomes

$$u = \frac{|\Delta P(z)|}{\Delta P(z)}C_d\left(\frac{2}{\rho}\left(\Delta P_0 - \rho g\frac{\Delta T}{T_0}z\right)\right)^{0\cdot5} \tag{26}$$

Integration of $u(z)$ along the inflow or outflow surface gives Q, the airflow rate.

Two important hypotheses concerning airflow across openings may be accepted: (i) average air inflow and outflow speeds are assumed constant (*i.e.* $u(z) =$ *constant*) in order to simplify the model and (ii) the expression of $u(z)$ in Eqn (26) is integrated for the full height of the opening. These two approaches are called the second- and first-order approaches respectively (Boulard & Baille, 1995).

3.3.2.2. First-order approach. In the case of a single opening, Eqn (26) may be simplified if constant air speed is assumed, equal to u over the whole surface of inflow and equal to $-u$ for outflow (*Fig. 5*). Then, if h is the height of the geometric centre of the surface of inflow,

$$u = \frac{|\Delta P(z)|}{\Delta P(z)}C_d\left(\frac{2}{\rho}\left(\Delta P_0 - \rho g\frac{\Delta T}{T_0}h\right)\right)^{0\cdot5} \tag{27}$$

If h_0 is the height of the neutral plane corresponding to the height for which the inside and outside pressures are equal, then $\Delta P = 0$ and $u = 0$.

Equation (26) gives for $u = 0$:

$$\Delta P_0 = \rho g\frac{\Delta T}{T_0}h_0 \tag{28}$$

substituting the new expression of ΔP_0 into Eqn (27) yields

$$u = \frac{|\Delta P(z)|}{\Delta P(z)}C_d\left(2g\frac{\Delta T}{T_0}(h - h_0)\right)^{0\cdot5} \tag{29}$$

Then, integrating u along 0 to $H/2$ (surface of inflow) or $H/2$ to H (surface of outflow) yields

$$Q = \frac{A}{2}C_d\left(2g\frac{\Delta T}{T_0}\frac{H}{4}\right)^{0\cdot5} \tag{30}$$

where A is the area of the opening.

3.3.2.3. Second-order approach. Integrating the speed of air flux $u(z)$ given by Eqn (26) over the inward surface (lower part of the opening) or the outward

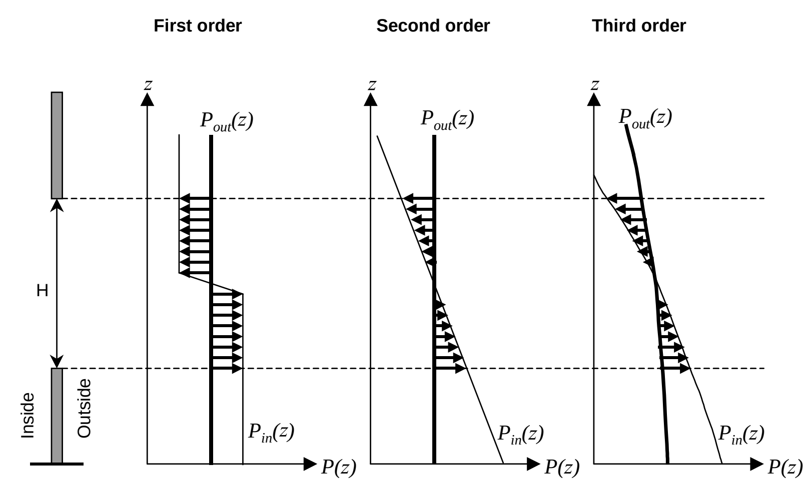

Fig. 5. Scheme of the distribution of inside pressure $P_{in}(z)$ (thin lines), outside pressure $P_{out}(z)$ (thick lines) and air speed (vectors) in an opening of height H (adapted from Boulard & Baille, 1995)

surface (upper part of the opening) yields

$$Q = \frac{wH}{3} C_d \left(2g \frac{\Delta T}{T_0} \frac{H}{2} \right)^{0.5} \tag{31}$$

where w is the width of the ventilator and H the vertical height of the opening.

3.3.2.4. Third-order approach. For mixed convection situations where the effect of external wind speed and the chimney effect are of the same magnitude, the determination of the temperature difference on each side of the opening is necessary in order to accurately model the airflow in the opening. The assumption of a linear temperature profile leads to a new approach where both temperature profiles are described by the following relations.

The inlet profile:

$$T_{in}(z) = T_{0,in} + b_{in} z \tag{32}$$

where $T_{0,in}$ is the inlet temperature for $z = 0$ and b_{in} the slope of the inlet profile.

The outlet profile:

$$T_{out}(z) = T_{0,out} + b_{out} z \tag{33}$$

where $T_{0,out}$ is the outlet temperature for $z = 0$ and b_{out} the slope of the outlet profile.

Recalling the Boussinesq approximation [Eqn (23)], Eqn (21) for wind speed u becomes

$$u = \frac{|\Delta P(z)|}{\Delta P(z)} C_d \left(\frac{2\Delta P_0}{\rho} + \frac{2g}{T_0} \left((T_{0,out} - T_{0,in}) z + (b_{out} - b_{in}) \frac{z^2}{2} \right) \right)^{0.5} \tag{34}$$

The neutral height h_0 can be determined by setting Eqn (34) at zero. Flow repartition at the opening is determined by specifying the forced convection flow rate (Kirkpatrick & Hill, 1988).

3.3.2.5. Case of two openings. Similarly, when considering two openings with areas A_{bot} (lower one) and A_{top} (upper one) separated by a vertical height h' and an identical discharge coefficient C_d, the Bernoulli equation and the continuity equation with the hypothesis of a second-order type approach help to determine the ventilation rate Q exchanged between A_{bot} and A_{top} induced by a temperature difference ΔT between inside and outside air (Bot, 1983; Boulard, 1993; Kittas *et al.*, 1997):

$$Q = \frac{A_{bot} A_{top}}{(A_{bot}^2 + A_{top}^2)^{0.5}} C_d \left(2g \frac{\Delta T}{T} h' \right)^{0.5} \tag{35}$$

3.3.3. *Wind effect*

The wind action on structures such as buildings or greenhouses results in a pressure distribution around these obstacles. Wind effects are usually split into two components: a mean component linked to the average wind speed through the average wind coefficient $\bar{C}$:

$$\Delta \bar{P} = \tfrac{1}{2} \rho \bar{C} u^2 \tag{36}$$

and a turbulent one, characterized by C' which expresses the fluctuating characteristics of the wind pressure:

$$\Delta P' = \tfrac{1}{2} \rho C' u^2 \tag{37}$$

As the measurements cannot determine the relative contribution of the mean and turbulent components, most authors (Boulard & Baille, 1995; Kittas *et al.*, 1995, 1996; Papadakis *et al.*, 1996; Baptista *et al.*, 1999; Bailey, 2000; Fatnassi *et al.*, 2002) assume an overall wind effect coefficient C_w that indifferently integrates both contributions:

$$\Delta P_w = \tfrac{1}{2} \rho C_w u^2 \tag{38}$$

Substituting the expression of the pressure drop given in relation (38) into Eqn (21) and integrating the flux over the area of inflow or outflow ($A/2$), the following simplified relation for ventilation flow can be deduced (Boulard & Baille, 1995; Papadakis *et al.*, 1996):

$$Q_w = \frac{A}{2} C_d C_w^{0.5} u \tag{39}$$

This relation is similar to the empirical equation proposed by Albright (1990) and Hellickson and Walker (1983) that links the ventilation rate linearly to wind speed:

$$Q_w = \frac{A}{2} C_t u \tag{40}$$

In that case C_t is the overall wind effect coefficient and $C_t = C_d C_w^{0.5}$.

Relation (38) may also be compared to the widely used non-dimensional ventilation function $G(\alpha)$ which linearly relates the ventilation rate to wind speed and the opening angle α of the specific ventilators of Venlo-type greenhouses (Bot, 1983; De Jong, 1990; Wang & Deltour, 1999; Bailey, 2000):

$$Q_w = G(\alpha) A' u \tag{41}$$

where A' is the area of the ventilator.

For a greenhouse with roof vent aeration, Boulard and Baille (1995) have more specifically analysed the correspondence between the parameters of relations (38) and (40).

3.3.4. *Combination of wind and chimney effect*

Greenhouse ventilation is usually a combined result of wind and buoyancy forces. Most authors (Walker & Wilson, 1993) assume a pressure field resulting from the sum of a pressure field due to both wind and thermal effects, *i.e.* $\Delta P = \Delta P_w + \Delta P_{th}$ and this hypothesis leads to the vectorial sum of individual wind and temperature fluxes according to the following formula:

$$Q = (Q_w^2 + Q_{th}^2)^{0 \cdot 5} \tag{42}$$

According to Walker and Wilson (1993), the hypothesis of an addition of flows instead of individual pressure differences leads to an error of the order of 10% in the estimation of the combined flow.

Boulard and Baille (1995) proposed the following relation for a greenhouse equipped with only a roof or only side openings for adding the individual pressure differences:

$$Q = \frac{A}{2} C_d \left(2g \frac{\Delta T}{T_0} \frac{H}{4} + C_w u^2 \right)^{0 \cdot 5} \tag{43}$$

For a greenhouse equipped with a roof (opening area A_r) and side openings (opening area A_s), Kittas *et al.* (1997) derived the following equation for calculating the combined effect:

$$Q = C_d \left(2g \frac{\Delta T}{T_0} \left(\frac{(A_r A_s)^2}{A_r^2 + A_s^2} \right) + \left(\frac{A_r + A_s}{2} \right)^2 C_w u^2 \right)^{0 \cdot 5} \tag{44}$$

Generally temperature-driven ventilation is only significant at low velocities, that is at wind speeds below $1\,\mathrm{m\,s^{-1}}$ for greenhouses equipped only with roof vents (Baptista *et al.*, 1999) or below approximately $2\,\mathrm{m\,s^{-1}}$ according to Bot (1983), Boulard (1993) and Papadakis *et al.* (1996). For a greenhouse with roof and side vents, Kittas *et al.* (1997) considered that temperature-driven ventilation is only significant if $u/\Delta T^{0 \cdot 5} < 1$.

3.4. *Determining the model parameters*

The wind effect coefficient C_w and the discharge coefficient C_d are the two main experimental coefficients involved in these semi-empirical models to be determined for each vent and greenhouse type. The airflow rate Q is generally determined from measurements of volumetric flow rates and climate parameters ($\Delta T, T, u$) are measured together with the vent opening area A. Using the models described in the theory, the unknown parameters can be deduced. Linear and non-linear regression techniques (such as Marquardt's algorithm for example) are used for the determination of the values of the discharge coefficient C_d and wind effect coefficient C_w. These minimize the error between the Q values measured and the ones calculated by using the models described by Eqn (43) or (44).

3.4.1. *Discharge coefficient*

The discharge coefficient C_d is a function of the window characteristics and is usually determined by model or full-scale experiments. Table 8 presents C_d values found in the literature, mainly for vertical continuous buildings or greenhouse openings. Generally, C_d values are between 0·6 and 0·8 with an average of 0·66. For greenhouse continuous openings, different authors do not assume any variation of the opening angle whereas for windows used with Venlo greenhouses, Bot (1983) has shown that the discharge coefficient C_d depends on the opening angle α of the window according to the relation

$$C_d = (1 \cdot 75 + 0 \cdot 7 e^{-(w/32)(1/\sin \alpha)})^{-0 \cdot 5} \tag{45}$$

where w is the width of the ventilator and H the height of the ventilator, measured in the plane of the greenhouse roof surface.

Discharge coefficient values refer to openings and greenhouses without any obstacles to air circulation. If there are obstructions, the C_d value can be decreased greatly as in the case of tall crops (Sase, 1989) or in the case of insect-proof and shading nets (Fatnassi *et al.*, 2002). Air discharge through the plant cover and nets is analysed in the second part of this review study (Boulard

Table 8
Discharge coefficient C_d determined by some authors

Discharge coefficient C_d	*Conditions*	*Source*
0·6–0·8	Building, rectangular	Brown and Solvason (1963)
0·67	Building, rectangular	De Gids (1978)
0·6	Building, rectangular	Bruce (1982)
0·65–0·7	Greenhouse, roof vents	Bot (1983)
0·63	Building, vertical and rectangular	Hellickson and Walker (1983)
0·6–0·7	Building, vertical and rectangular	Timmons *et al.* (1984)
0·65	Building, vertical	Bois *et al.* (1988)
0·61	Building, vertical and rectangular	Zhang *et al.* (1989)
0·65–0·75	Greenhouse, roof vents	De Jong (1990)
0·65	Building, rectangular	Randall and Patal (1994)
0·61	Building, rectangular	Vandaele and Wouters (1994)
0·644	Greenhouse, continuous vents	Boulard and Baille (1995)

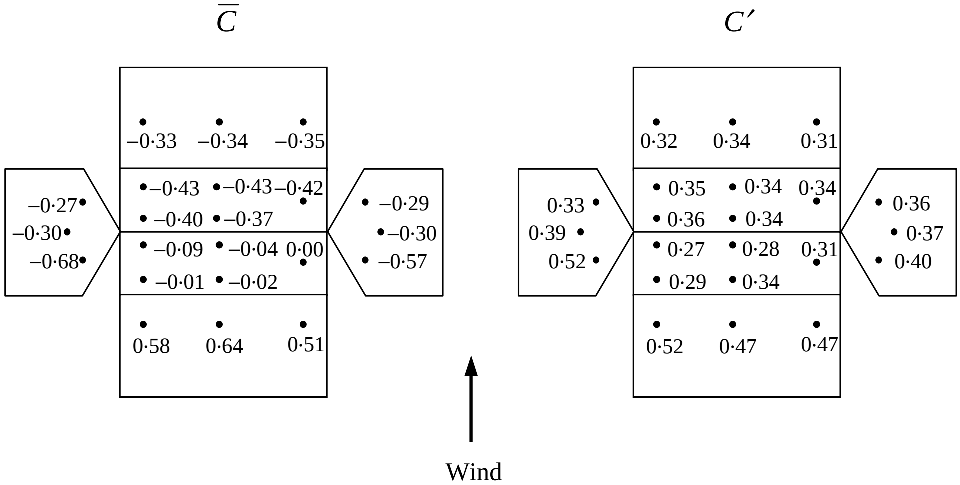

Fig. 6. Distribution of average wind effect coefficient $\bar{C}$ and turbulent wind effect coefficient C' (after Gandemer & Bietry, 1989)

et al., 2002). A detailed study of the combined effects of the discharge coefficients of the opening itself and of a net or a crop cover is given by Fatnassi *et al.* (2002).

3.4.2. *Wind effect coefficient*

In situ measurements of $\bar{C}$ and C' [see Eqns (36) and (37)] on buildings (Gandemer & Bietry, 1989) show that they have the same order of magnitude and $\bar{C}$ is very dependent on wind direction whereas C' is independent of it. Few values of C' are available in the existing literature. An example of $\bar{C}$ and C' values, recorded for a building with large wind tunnel facilities (Gandemer & Bietry, 1989) is given in *Fig. 6*.

An attempt to relate the wind effect coefficient C_w to $\bar{C}$ and C' was made with the help of sonic anemometry and fast-response pressure measurements by Boulard *et al.* (1996) in a greenhouse with a single roof opening. This showed that the turbulent characteristics of the wind were responsible for about 30–40% of the air exchange in the greenhouse. However, in practice, most measuring techniques used in the determination of ventilation rate do not help to determine the relative contribution of the steady and turbulent wind effects. Most researchers therefore consider only the coefficient C_w.

Table 9
Wind effect coefficients C_w (from Bailey, 2000)

Wind effect coefficient C_w	*Greenhouse area (m^2)*	*Source*
0·10	416 (2 spans)	Boulard and Baille (1995)
0·14	179 (1 span)	Kittas *et al.* (1995)
0·071	900 (tunnel)	Kittas *et al.* (1996)
0·13	416 (2 spans)	Papadakis *et al.* (1996)
0·09	204 (4 spans)	Baptista *et al.* (1999)
0·11	38 700 (60 spans)	Bailey (2000)

According to Bailey (2000) this coefficient seems to be independent of the surface area of the experimental greenhouse since C_w values which are very close to each other were obtained from measurements in single- and multi-span greenhouses with areas between 180 and 38 700 m^2 (Table 9).

Boulard and Baille (1995) have shown that the greatest differences seem to be due to the range of wind speed observed during the experiments for the determination of C_w. They demonstrate a significant decrease of C_w with increasing wind speed.

3.4.3. *Overall coefficient of wind efficiency on ventilation*

There are several models in which an overall wind effect coefficient including both the wind effect itself and the discharge coefficient in the vent openings is taken into account. All these models indicate a linear relation between the ventilation rate and the wind speed with a specific coefficient: the coefficient $C_d C_w^{0{\cdot}5}$ for relation (38), the coefficient C_t and $G(\alpha)$ in relations (39) and (40).

Experiments on single- and multi-span greenhouses with areas between 180 and 5000 m^2 (Kittas *et al.*, 1995, 1996; Boulard & Baille, 1995; Papadakis *et al.*, 1996; Baptista *et al.*, 1999; Bailey, 2000; Fatnassi *et al.*, 2002) have shown that $C_d C_w^{0{\cdot}5}$ could be treated as a constant (0·16–0·27) whose value depends upon the range of wind speed (Boulard & Baille, 1995) and by the wind direction with respect to the openings (Fatnassi *et al.*, 2002).

Similarly, values for $G(\alpha)$ determined for different greenhouses by Fernandez and Bailey (1992), Boulard

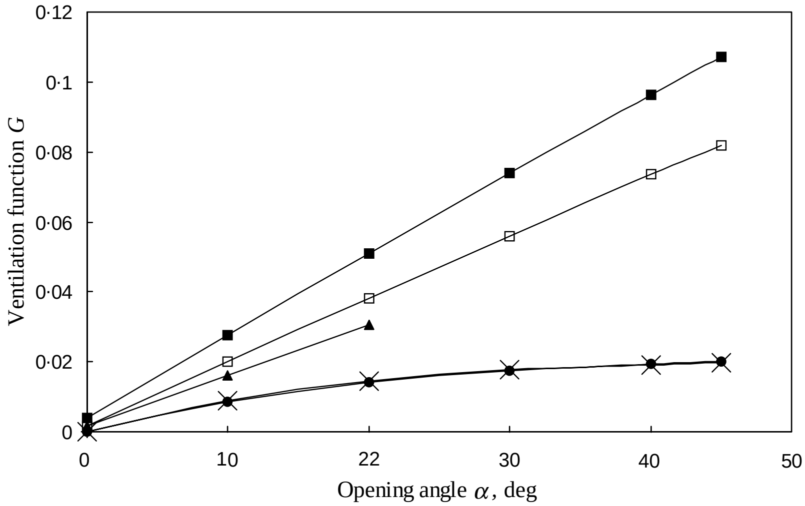

Fig. 7. Non-dimensional ventilation function G as a function of the opening angle α, according to different authors: -×-, Bot (1983); -●-, De Jong (1990); -▲- Fernandez and Bailey (1992); -□-, Boulard and Draoui (1995); -■-, Kittas et al. (1996)

and Draoui (1995), Kittas *et al.* (1995) and Wang (1998) are approximately equal (*Fig. 7*). However, Bot's (1983) and De Jong's (1990) results showed a distinct non-linearity between $G(\alpha)$ and angle of opening and far lower values for $G(\alpha)$ which contrast with the results of the other authors.

4. Conclusions

The main body of scientific literature relative to convection and ventilation processes involving simplified schemes such as homogeneity of the greenhouses climate has been presented in this review. Two main steps with different objectives have been identified, (i) the determination of energy consumptions in glasshouses was first performed during the 1960s and 1970s and (ii) the study of climate control and particularly ventilation performances, during the 1980s and 1990s. Only sensible heat transfers were studied in the first case whereas both heat and mass transfers as well as the transpiration rate were considered in the second case.

Two semi-empirical models were used in these studies for the determination of heat transfer and ventilation transfers in greenhouses. The measurement techniques associated with these approaches are also rather simple: temperature measurements in the first case, air temperature and humidity measurements together with air exchange rate measurements pertaining to the whole greenhouse volume in the second case.

From a practical point of view, the combination of simple semi-empirical models describing the convective exchanges and the greenhouse ventilation is often sufficient in solving most classical engineering problems and now, little new research is carried out in these domains.

The present challenge is to be able to exploit these models for improving the design of the greenhouse and greenhouse control devices (Martin-Clouaire *et al.*, 1996), or to derive more efficient control algorithms (Bailey & Chalabi, 1994; Ferentinos *et al.*, 2000) or strategies (Tchamitchian *et al.*, 1997).

Yet, there is always the need for further research on greenhouse micrometeorology, to permit for example realistic descriptions of the complexity of the climate in the greenhouse and at plant level. This is presented in the second part of this review study (Boulard *et al.*, 2002) dedicated to the greenhouse distributed climate.

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