Comparison of Upward and Downward Air Distribution Systems

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Introduction

At present there is a conflict between various sources of information concerning the specification of the parameters to ensure a good quality airmovement system. Independent surveys of office conditions in the USA and Europe show that stagnant air conditions are quoted by more than half the subjects as contributing towards lethargy and low productivity (Woods 1985). Laboratory experiments carried out by Fanger and Christensen (1987) claim that the turbulence of the airflow makes people more sensitive to draught than was found in previous studies and then propose a reduction of velocity limits in the present standards in accordance with the percentage dissatisfied (PD) equation.

$$PD = 13800\left\{\left(\frac{\bar{v}-0.04}{\theta_a-13.7}+0.0293\right)^2-0.000857\right\}$$

For 5% to be dissatisfied this results in a mean velocity ($\bar{v}$) of 0.08m/s at 20°C (θ_a) rising to 0.1m/s at 26°C. Field surveys and everyday experience tend to contradict these proposals. Clark (1985) using Schlieren techniques has shown that convection velocities around the body especially above the head are 0.2 - 0.4 m/s ; these figures are independently supported by measurements taken by van Gunst in the de Doelan concert hall in Rotterdam described by Croome and Roberts (1981). Airstreams with velocities of 0.10 m/s can easily be deflected by the body convection currents thus rendering the airflow system ineffective from the freshness point of view.

The present standards define mean velocities only. This is insufficient and finer details of the airmovement pattern need defining especially at head and foot levels. Mayer (1985) has defined the **standard deviation** for the airmovement fluctuations as:

$$S = Tu.V_{50\%} = V_{84\%} - V_{50\%}$$

where Tu = turbulence coefficient (0 - 0.6)

$V_{50\%}$ = velocity exceeded for 50% of the time (mean velocity)

$V_{84\%}$ = velocity exceeded for 16% of the time

Fanger and Christensen (1986) studied the velocity fluctuations around the body and derived the standard deviation in terms of $V_{50\%}$ (at the elbow, at the feet and at the back of the neck), the space air temperature and the heat input into the space. Besides turbulence, the **periodicity** (T) of the fluctuations is important where fluctuations are defined as

$$\tilde{V} = (S + V_{50\%}) \sin(2\pi t/T + \phi)$$

or
$$\tilde{V} = V_{50\%}(1 + Tu) \sin(2\pi t/T + \phi)$$

Linke (1966) and Regenscheit (1970) have described the basic fundamentals of air motion in air distribution systems. The work of Linke carried out in a lecture theatre with 500 seats at the Technical University in Aachen has revealed the patterns of airmovement and temperature distribution resulting from conditioned air being supplied at floor level. Because the main direction of airmovement corresponds with that induced by heat released from the occupants, air supplied from floor level produced an even pattern of air flow throughout the auditorium and the vertical temperature gradients were negligible above head level but the temperature from foot to head level varied from 3 to 5°C (Croome and Roberts, 1981), high differentials occurring when the occupancy was high. The advantages of upward systems are mainly in the use of reduced air supply rates by using (i) higher temperature differentials and (ii) occupancy rather than total space volumes for overall heating and cooling requirements. Extraction of the heat from the upward moving air via the lighting troffers, can easily be achieved; alternatively high temperature stratification air can be recirculated to the occupancy zone. Downward systems showed large circular currents with temperature differentials of 4 to 9°C with full occupancy but in this case most of the temperature gradient occurs within 1.5m of the ceiling and the temperature drop from head to foot level was about 1°C.

Characteristics of Upward and Downward Systems

According to Linke, the Archimedes number is a decisive factor influencing the air movement patterns. For downward air flow distribution systems, the Archimedes number should be $Ar \leq 46$; for upward air flow distribution system, the Archimedes number should be $Ar \leq 360$.

The Archimedes number is defined as

$$Ar = \frac{g\,\beta\,\Delta\theta\,H}{v^2} \qquad (1)$$

Where g = acceleration due to gravity m/s^2
β = thermal expansion factor ($^{-1}$ K)
$\Delta\theta$ = supply to return air temperature differential (°C)
H = height of the auditorium, (m)
v = air velocity, (m/s)

Equation (1) can also be defined in the form of

$$Ar = \frac{g\,\beta\,q}{\rho\,c\,n^3 H^2} \qquad (2)$$

Where q = heat load (W/m^2)
ρ = air density (ρ = 1.2 Kg/m^3)
c = specific heat capacity of the air (c = 1006 J/Kg°C)
n = air change rate (ach/h)
g = acceleration due to gravity

Using equation (2) and the Archimedes conditions stated above the minimum air change rate ensuring a stable air movement pattern in the room for a downward system is

$$n > 30.4\sqrt{q/H^2} \qquad (3)$$

and for an upward system is

$$n > 15.3\sqrt{g/H^2} \tag{4}$$

Comparing equations (3) and (4), it can be concluded that the required air change rate to acquire a stable air movement in a space for a downward system is twice as large as that of an upward system. This is another reason why the upward system is suitable for spaces with large heat gain ($>$ 140W/m²) such as auditoria, where the floor level is heavily occupied by people.

The supply to return air temperature differential, $\Delta\theta$, for upward systems could be 10-12°C while for downward systems is 8-10°C and hence the fan energy could be reduced by about 20-25%. The total energy saving could read 7-9%, assuming 35% of the total energy is used for delivering air.

Generally speaking, the supply air temperatures for upward systems are about 18-19°C and 14-18°C for downward systems in summer. With higher supply temperatures about 8-10% of the cooling load could be saved, resulting in reduced energy costs of about 1.5%

A comparison of upward and downward systems is shown in Table 1.

(a) The Archimedes number can be defined in terms of the outlet characteristic

$$Ar = \frac{g\, d_o\, \Delta\theta_o}{V_o^2\, \theta} \tag{5}$$

Where $\Delta\theta_o$ = temperature difference at jet entry (°C)
θ = room temperature (°K)
d_o = outlet diameter (m)
v_o = outlet velocity (m/s)
g = acceleration due to gravity (9.81 m/s²)

Ar decreases as the air outlet velocity, v_o, increases. With increasing supply to room air temperature differentials, $\Delta\theta_o$, Ar increases (ie buoyancy increases) and the more pronounced the curvature of the jet axis becomes.

(b) The trajectory is calculated by inserting Archimedes number in

$$\frac{y}{d_o} = \frac{x}{d_o}\tan\alpha + Ar\left(\frac{x}{d_o\cos\alpha}\right)^2\left(0.51\frac{t\,x}{d_o\cos\alpha} + 0.35\right) \tag{6}$$

Where y = the vertical displacement (ie deflection from horizontal axis) (m)
α = inclination angle
x = horizontal distance from outlet (m)
t = turbulence factor, for vaned outlet, $t = 0.2$

(c) The axial jet velocity profile is derived using the jet velocity decay equation, Reigenscheit's (1970)

$$\frac{V_x}{V_o} = \frac{x_o}{x} \pm \frac{Ar}{m}\left[1 + \ln\left(2\frac{x}{x_o}\right)\right]^{1/2} \tag{7}$$

Where m = mixing number = d_o/x_o (0.1 to 0.3)
x_o = core zone length (m)

This compares favourably with Koestel's work (Sofrata 1987, Koestel 1954).

CHARACTERISTICS	**DOWNWARD**	**UPWARD**
Vertical temperature gradient:		
i Foot to head level	Negligible(1to3°C): head temperatures are a little warmer	3-8°C lower temperature at foot level.
ii Above head level	4 - 8°C	Negligible
Supply temperature	In the order of 14 to 18°C	Higher (18°C minimum) than for downward system but more susceptible to draughts at ankle level.
Dust	Dust kept at floor level	Dust tends to rise - essential to avoid this in opera houses, concert halls or debating chambers.
Noise	-	Airborne sound more likely to be heard.
Maintenance	-	Supply grilles need regular cleaning.
Number of supply outlets	Can use a small number of outlets but less flexibility to control	Necessary to use a large number of small outlets
Energy	Air has to deal with lighting gains before those from people.	Air absorbs heat gains from people before lighting ones. Savings in energy can be large in high, well-insulated spaces.

Table 1: THE CHARACTERISTICS OF DOWNWARD AND UPWARD VENTILATION SYSTEMS

The axial velocity varies inversely with the distance from the outlet. The velocity at any point in the jet flow can be divided into two components:

longitudinal velocity v_x and cross-sectional velocity v_y while v_x is much larger than v_y in most cases. In practice, therefore the cross-sectional velocity, v_y, is neglected. It is safe to consider that $v = v_x$, particularly in the main zone of jets, which are mainly used in air-conditioning. Hence, Reigenscheit's equation can be modified using empirical data as expressed as

$$\frac{V_x}{V_o} = \frac{0.48}{tx/d_o + 0.145} \qquad (8)$$

The velocity decay equation, which includes the Ar, also demonstrates the importance of the Archimedes number as a design factor. Experiments on the behaviour of jets in rooms have shown that the Archimedes number correlates with the air movement patterns in space.

Supply Air Parameters

(a) Work by Linke reported in Croome and Roberts (1981) indicated that a **throw giving a velocity of 0.5 m/s at three quarters of the room length,** was a suitable criterion to ensure satisfactory room motion without the presence of high velocities in the occupied zone. On this basis the required supply air velocity and the outlet diameter may be determined.

The **air outlet velocity (v_o)** normally ranges from 5 m/s to 7 m/s for high level outlets in downward air distribution systems; a limitation on the velocity for upward systems is normally observed to avoid excessive sound emission and cold draught in the occupied zone. In the case of sedentary or light work activity, the suitable (v_o) for low level outlets is 0.5 to 1.0 lm/s. Hence, larger outlet areas are required for a given air flow rate in upward systems. The use of twist outlets gives more scope for achieving penetration of air into the space with less likelihood of draughts or noise.

(b) The allowable **supply to room air temperature differential, $\Delta\theta_o$** for downward distribution may be up to about 11°C but a smaller value, say 5°C, should be selected if air is distributed from low location outlets to meet the people's comfort. This does not mean that a larger amount of supply air volume is needed in upward systems because of the Archimedes criterion referred to previously (see equations 3 and 4) and the use of occupied zone volume instead of total space volume for heating and cooling calculations.

(c) **The temperature difference profile** is similar to that of the axial velocity, v_x, especially in the main zone of the jets. Hence, the temperature decay law can be expressed in a similar form as for the velocity decay laws;

$$\text{hence} \quad \frac{\Delta\theta_x}{\Delta\theta_o} = \frac{T_x - T_n}{T_o - T_n} = \frac{0.35}{tx/d_o + 0.145.} \qquad (9)$$

Where T_x = core temperature of the jet flow
T_n = room air temperature
T_o = temperature at jet entry

Microclimate Air Distribution System

A microclimate or task air distribution system is suitable for buildings such as auditoria and lecture theatres, where seats are fixed in permanent locations. The air distributed from the airconditioning system is supplied through the ducts beneath the seats and is delivered through outlets located at the back of each seat, supply air jets being formed at seated head level. When an air stream travels from an outlet, the kinetic energy is increased in creating turbulence due to the entrainment of secondary room air into the jet stream, (convection currents in the head region due to the mixing effect of the secondary air plus the effects of buoyancy). It is sensible psychologically to provide occupants with some control of air flow direction and/or velocity, usually a simple manual damper control, thus avoiding draughts or increasing freshness as required. The inclination angle of the vane ranges from 0° to 20° from the vertical axis. The inclination angle is usually adjustable and therefore settings could be selected to meet the various preferences of the individuals. The velocity of air at the outlet should not exceed about 1.5 m/s; Sodec (1984) shows that the front of the face can enjoy short intermittent velocity amplitudes of 0.6 m/s.

An alternative form of the micro-climate air distribution system is a system built in to the seating structure. As shown in Figures 1 and 2, the conditioned primary air is generally fed from a pressure chamber accommodated in the chair mounting supports. Indoor air mixes with the primary air and the supply air emerges at the top of the back-rest, at an angle of 0-20° to the vertical axis. In each case, the direction of discharge is selected in such a way that the head of the seated individual is located not in the direct path of the jet, but rather within its induction zone. The momentum of the jet is set so as to ensure stability of the jet direction within the occupied zone and this can be varied in all load situations. Sodec (1984) reported that systems of this type serve to meet the following demands:

- Stability of air distribution within the occupied zone without the substantial circular room air patterns that develop in downward systems.
- Provision of the human respiratory system with a direct supply of conditioned air.
- Provision of adequate convection within the seating area by means of the primary-secondary air intermixing action.

The features of the micro-climate system on which the outlets are mounted to the top of the back-rest are:

- The cold supply air is discharged into the upper half of the occupied area.
- The lower half is conditioned by induction of the secondary air.
- The air discharge velocity at the air outlet is approximately 1.5 m/s; this caters for adequate induction of the indoor air.
- No formation of stagnant patches of cold air.
- Direct discharge of fresh (draught-free) supply air into the occupied zone without having first to enter from floor outlets.
- Effective air distribution in the occupied area.

Typical design parameters are

- air volume flow rate:	8.5-10 l/s per outlet
- minimum supply air temperature:	18°C
- induced secondary air:	3.5-5 l/s
- sound power level:	18-26 db (A)
- pressure loss:	30-50 Pa
- air velocity in hollowed seat pedestal:	1.5-2.6 m/s

The return air temperature underneath the ceiling can be as high as 30°C, which will not cause discomfort in theatres because of the extensive room height. Within the occupied area itself the air temperature is 23-24°C owing to the direct arrangement of the seats, an air volume flow rate of 8.5-10 l/s per outlet at $\Delta\theta$ = 12K will suffice to meet the requirements of the room. The proportion of induced secondary air is approximately 40-60% of the primary air volume flow rate.

Compared with downward air supply systems the microclimate air distribution system has several advantages:

- Direct supply of air to the immediate vicinity of the occupants.
- Greater temperature differences between return and supply air of up to 12°C, hence a lower air volume flow rate is necessary.
- Lower pressure losses.
- Lower refrigeration consumption on account of higher supply air temperature (18°C instead of 14-16°C).
- Use occupancy space volumes in heating calculations.
- Reduced investment costs due to
  smaller dimensioned central units,
  smaller refrigeration units
  fewer ducts.

A saving of about 20% is power costs is effected, apart from the investment expenses bill being cut by 30-40%.

Combined Air Distribution System

Conditioned air can be supplied through grilles located beneath the auditorium seatings and outlets at the top of the back-rests (see Figures 3 and 4). The system normally needs double-deck floors, which form a plenum pressure chamber or space for under floor ducts. In the case of double-deck floors the supply air is distributed evenly through the plenum to the various outlets which are inserted in the raised floor. Normally the depth of the plenum should be at least 200mm. Floor ducts are suitable for conveying the supply air to specific areas of the auditorium. Each individual outlet is made to form a direct link with the supply air branches - usually by means of flexible ducting. One of the advantages of the combined air distribution system is that it creates a basic conditioned environment for the extensive areas in the auditorium besides a micro-climate for the individual's comfort.

Types of Floor Mounted Outlets

The recommended outlets suitable for upward air distribution system assume the following forms:

Slot Plates: The flow emitted from a slot plate is similar to that encountered in a perforated plate system. The small individual jets - except the outer ones - do not induce the room air, but rather the adjacent jets of supply air. The reduction of jet velocity takes place by means of the diffuser effect and not by exchange of energy with the environment.

Free Jet Outlets: These produce round, non-twist type air jets; the diameter of the free outlets ranges from 150 to 250mm
No diffuser current develops and the induction of room air is more intensive than for slot plates.

Floor-mounted Twist Outlets: On the floor-mounted twist outlets the air jets are emitted in a swirl corkscrew pattern. As a result of the higher degree of turbulence, a more intensive induction effect of the indoor air is brought about; the air jet is stable and less sensitive to cross convection so that jet penetration is improved. Owing to the larger amount of small inclined jet with swirl effect, intensive exchange of energy with the ambient air is attained. The reduction in jet velocity and adjustment of the supply air temperature to the temperature of the room air proceed at a faster rate than is the case with slot plates and free jet outlets. Due to the geometry of the outlet the noise emission is reduced.

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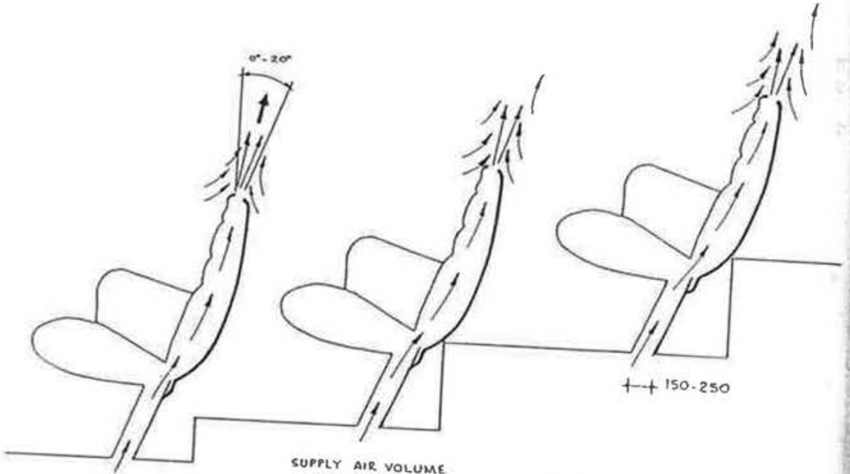
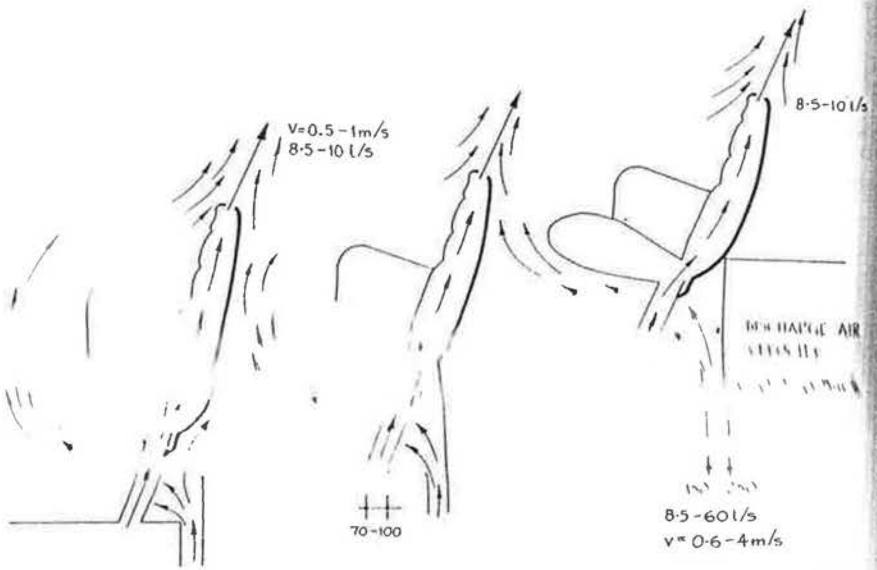
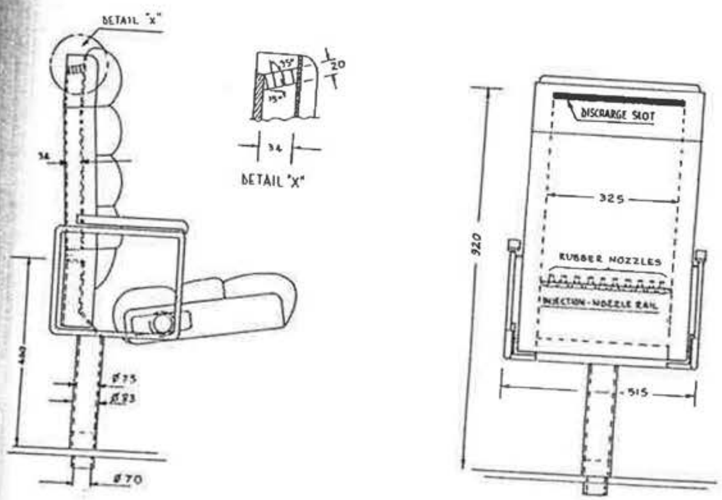


FIG 1

BUILT_IN SEAT MICRO_CLIMATE AIR DISTRIBUTION SYSTEMS



BUILT_IN SEAT MICRO_CLIMATE AIR DISTRIBUTION SYSTEMS FIG.2



PERFORATED PEDESTAL TYPE OF SUPPLY OUTLET

FIG.3

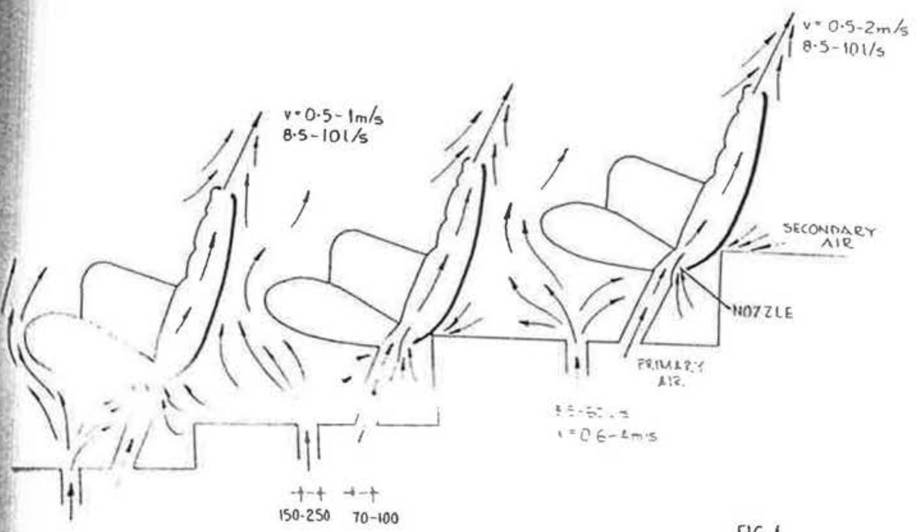


FIG.4

COMBINED MICRO_CLIMATE AIR DISTRIBUTION SYSTEMS