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COLD AIR DISTRIBUTION

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ABSTRACT:

HVAC systems with *Cold Air Distribution* (CAD) supply air from overhead at 4°C to 10°C, in comparison to "conventional" supply air temperatures of 12°C to 15°C. CAD can result in reduced mechanical system costs, lower energy consumption, enhanced comfort, improved indoor air quality (IAQ) and reduced HVAC space requirements. CAD reduces first costs because of smaller services such as power supply, air handlers, pumps, ducts and coils, though coil rows and duct insulation usually have to be increased. Enhanced dehumidification improves comfort and may reduce the threat of mould. Strategies must be adopted for proper air diffusion to ensure adequate indoor air movement and temperature distribution. High induction diffusers, such as swirl diffusers or multi-nozzle diffusers, are recommended for CAD systems to ensure high ADPI performance. High induction diffusers prevent dumping, ensure uniform temperature distribution and increase the effective air changes per hour to compensate for the lower primary airflow rates, especially in low heat load centre zones. CAD systems may loose their viability where space relative humidity must be maintained above 40% to 45%, as well as in climates where reduced supply air temperatures negate free-cooling benefits or where cooling humid outdoor air may be energy inefficient or operationally impractical. Additionally, applications where spaces with high occupant concentrations demand high fractions of minimum outdoor air, or where significant reheat has to be introduced when supply temperature reset is not able to overcome zone overcooling during turndown, may not be suitable for CAD.

KEYWORDS:

Cold air distribution, CAD, low-temperature air distribution, air diffusion, mixed flow, swirl diffuser, twist outlet, vortex diffuser, high induction diffuser, multi-nozzle diffuser, pre-induction diffuser, aspiration, induction ratio, thermal comfort, air diffusion performance index, ADPI, draught rate, draft rate, DR, predicted mean vote, PMV, percentage dissatisfied, PD, ventilation effectiveness, air change effectiveness, ACE, thermal comfort, thermal environment, indoor air quality, IAQ, energy efficiency, free cooling, economiser cycle, duct insulation, fan size, pump energy, coil rows, latent load, reheat, sick building syndrome, SBS

AIRAH Conference 2005: Energy Sustainability & Power:

This paper was presented by Sean Badenhorst at the ***AIRAH Conference 2005: Energy Sustainability & Power*** on Friday 14/10/2005 in Perth, Australia.

INTRODUCTION

Ongoing improvements in both chiller design and efficiency at low water temperatures, as well as in high induction diffuser design for low supply air temperatures, permit capital and energy savings to be achieved by lowering supply air flow rates as supply air temperature is reduced. Reduced airflow rates reduce fan, duct and electrical first costs, fan and pump energy requirements, and noise. However, costs for cooling generation and heat transfer may increase. Even so, cold air distribution technology can result not only in lower mechanical system costs, but also in reduced energy consumption, improved indoor air quality, and enhanced comfort.

1.0 COLD AIR DISTRIBUTION

In comparison to HVAC systems with "conventional" overhead supply air temperatures of 12°C to 15°C, HVAC systems with *cold air distribution* (CAD), also known as *low-temperature air distribution*, supply air from overhead at 4°C to 10°C.

Almost all aspects of the air conditioning system are affected by the reduction in supply air temperature.

As supply air temperature is reduced, so:

- Fan size and duct size decrease.
- Fan heat and fan energy decrease.
- Space required for fans and ducts decreases.
- Required chilled water supply temperature or DX suction temperature decreases.
- Chilled water temperature differential increases.
- Pump energy and pipe dimensions decrease.
- Cooling coil face area decreases and depth increases.
- Latent load and associated cooling energy increases.
- Free cooling potential decreases.
- Requirement for thermal insulation increases.
- Ventilation air fraction increases, although the absolute quantity generally remains constant.
- Requirement for high induction air diffusion with high ADPI and low DR increases.
- Requirement for reheating minimum supply air quantities may increase.
- Space relative humidity decreases.

The trade-offs between the above must be evaluated for the given project to determine the optimum supply air temperature, and must be weighed against the advantages and disadvantages of reduced supply air temperature.

1.1 Reduced Supply Air Temperature - Advantages

- *Reduced mechanical system costs*

  First cost reductions, especially of fans and ductwork, result from reduced air quantities.

- *Decreased floor-to-floor height requirements*

  Significant structural, envelope and other building system cost reductions are possible due to reduced space requirements of cold air systems.

- *Improved comfort with lower space relative humidity*

  Occupants feel cooler and more comfortable at lower humidity, and the air is judged as fresher with a more acceptable air quality[1]. Lower air quantities also provide potential to reduce noise.

- *Increased room air temperature due to lower space relative humidity*

  Decreasing dewpoint temperature by 6 K has the same effect on thermal sensation as a 0.5 K reduction in dry-bulb temperature. It is therefore possible to increase the dry-bulb temperature by 0.5 K if the space relative humidity is decreased from about 50% (typical for 13°C supply air temperature) to 35% (typical for 7°C supply air temperature).

- *Improved indoor air quality due to lower space relative humidity*

  Reduced supply air quantity does not result in a reduction in the volume of outdoor air supplied to the space. This normally remains the same (though the percentage is higher). However, lower space humidity decreases the threat of mould/mildew growth in the space, improving indoor air quality and reducing the threat of sick building syndrome (SBS).

- *Reduced fan energy consumption*

  Fan energy reduces by as much as 30% to 40% due to the lower air quantity supplied by a cold air system. This reduction is primarily in expensive on-peak demand. Where cold-air distribution is used with ice storage, any increases in cooling energy are in the form of less expensive, off-peak demand.

- *Increased cooling capacity for existing air distribution systems*

  The expense of replacing or supplementing existing air distribution equipment can often be avoided by reducing supply air temperature.

1.2 Reduced Supply Air Temperature - Disadvantages

- *Increased threat of condensation*

  The lower dew-point conditions of cold air distribution systems do not substantially increase the threat of condensation, as the cold air distribution system reduces the humidity throughout the building. However, areas where precautions against condensation need to be considered – especially in humid climates – include spaces with large infiltration (eg foyers) and ducts that pass through unconditioned spaces. Measures to reduce condensation typically comprise thicker duct insulation and diffuser selection to prevent condensation on the diffuser face.

- *Increased duct heat pickup*

  Lower supply air temperature increases the temperature differential between cold supply air in the duct and the temperature of the space through which the duct passes. Moreover, the lower airflow rate causes the temperature rise of the supply air to be greater for any given heat flow through the duct wall. As a result of these two factors, heat pickup by the supply air is increased, unless additional duct insulation is used.

- *Need for high induction, high ADPI diffuser selection*

  Supplying cold air to standard ceiling or side-wall diffusers greatly increases the threat of dumping. By selecting high induction, high ADPI diffusers that strongly mix the cold supply air with room air, cold air distribution will not increase the threat of draughts.

- *Reduced free-cooling potential*

  The lower supply air temperature of cold air distribution systems reduces the potential to benefit from economiser cycle operation. This strongly impacts energy consumption in climates where ambient temperature during HVAC operation often falls below the supply air temperature of an alternative "conventional" air distribution system (eg 13°C) but seldom falls below the selected cold air distribution supply air temperature (eg 7°C). During periods of reduced cooling demand, if free cooling is available at a slightly higher supply air temperature, then controls should reset supply air temperature to permit use of free cooling in preference to turning down fan speed – the reduced mechanical cooling costs will more than offset the increase in fan energy costs.

- *Minimum supply air reheat*

  Cold supply air may result in zone overcooling when a VAV system has fully turned down, especially in spaces not served by independent air handlers but that are subject to independent and strongly fluctuating loads, such as boardrooms and meeting rooms. Occupancy sensors or VAV box activation by means of light switches may have to be considered to deactivate such zones when they are not in use so as to prevent overcooling. Additionally, supply air reheat may be necessary, which may negatively impact on energy consumption. Reheat could be achieved by means of a heat recovery condenser fitted to the chiller, to avoid use of an external heat source.

2.0 SYSTEM COMPONENTS

Cold air distribution systems are often used with ice storage systems (though many benefits are retained even without the use of ice storage). This paper does not delve into the design of ice storage systems.

System components of cold air distribution systems differ from those of conventional air distribution systems in the following way.

2.1 Cooling Plant Requirements

For typical coil selections, the fluid temperature should be at least 3 K lower than the desired discharge air temperature. For cooling plants without ice-storage, chillers can be selected to provide any supply temperature required for cold-air distribution, and typically operate at fluid differentials of 11 K to 13 K, compared to conventional differentials of 5 K to 9 K. This increased range reduces pump flow and pipe size, as well as pump energy. With reduced fluid supply temperatures used in cold-air distribution systems, additional pipe insulation may be required if fluid temperature is below 4°C.

Chillers operating at 4°C to 6°C are commonly used with cold-air distribution systems supplying air at 8°C or more; such systems usually do not require the addition of glycol to prevent chilled water from freezing. For lower supply air temperatures, 2°C to 3°C chilled water temperature is typical; water-glycol solutions are usually used in such systems[2].

Direct expansion systems (DX) generally supply air above 7 °C to prevent coil frosting or freezing and liquid carry-over to the compressor unless special provisions are made. Discharge air temperature control is less stable than with a chilled-water system.

2.2 Cooling Coil Requirements

In comparison to conventional systems, cold air distribution coil selection has:

- Lower leaving air temperature.
- Lower entering water temperature.
- Lower face velocity.
- Closer approach between the entering chilled-water temperature and the leaving air temperature.
- Wider range between the chilled-water supply and return temperatures.

Table 1 lists common coil design parameters for conventional and cold air distribution systems.

Table 1

Common Coil Selection Parameters	Conventional	Cold Air Distribution
Leaving air temperature	12.8°C	5.6 – 10.0°C
Chilled-fluid temperature entering coil	5.6 – 7.2°C	2.2 – 5.6°C
Face velocity	2.3 – 2.8 m/s	1.5 – 2.3 m/s
Approach	5.5 – 7.2°C	2.2 – 5.5°C
Chilled-fluid temperature range	5.5 – 8.8 K	8.8 – 13.2 K

Conventional designs typically have four to six row coils with 0.3 to 0.6 fins per millimetre. For supply air temperatures of 8 °C to 10 °C, six-row coils usually provide acceptable performance, but for lower supply air temperatures eight-to-ten row coils are typically used. Fin spacing of 0.4 to 0.5 fins per millimetre is typical for cold-air systems to minimise moisture carryover.

The recommended coil face velocity for a cold-air distribution system is typically 1.5 to 2.3 m/s, which is lower than for standard designs (2.3 to 2.8 m/s) because of more moisture condensation from the air and the greater risk of water entrainment into the air stream.

The wider fin spacing and lower face velocity typically compensate for increased rows in system with a supply air temperature above 7 °C, preventing an increase in air-side pressure drop, however this does result in increased fluid-side pressure drop. Systems with a supply air temperature below 7 °C usually also have an increased air-side pressure drop. Even so, fan and pump energy are still well below those of conventional systems, due to the larger air-side and water-side temperature differentials.

When a high chilled-water supply-to-return temperature range is desired to minimise pumping energy and pipe size, more coil surface will generally be required.

2.3 Fans

In a draw-through arrangement, the fan is downstream of the coil, resulting in the fan heat being transferred to the supply air stream, typically increasing supply air temperature by 1 K to 2 K, and therefore increasing the required airflow to meet the load.

In a blow-through arrangement, the fan is upstream of the coil, resulting in the fan heat being transferred onto the coil. As a result, a lower supply air temperature can be achieved from the same coil. However, fan discharge onto the coil can result in non-uniform air distribution across the coil, which may reduce coil performance and cause moisture carryover. Moreover, fan performance may be reduced due to the less desirable fan inlet and exit conditions. Also, the threat of condensation on downstream surfaces is increased, especially if supply air temperature fluctuates.

2.4 Duct Layout

If the same pressure loss criteria are used as for traditional designs, duct size and fan energy will be reduced in proportion to the reduction in air volume. Where available space limits the height of ducts, cold-air distribution allows more favourable duct aspect ratios to be selected.

2.5 Duct Insulation

In order to prevent condensation, the insulation must maintain the duct surface temperature above the space dew-point temperature, it must cover all surfaces that may be cooled below the space dew-point temperature, and external insulation must have an external vapour barrier to prevent the diffusion of moisture into the insulation. Thorough coverage, a complete, unbroken vapour barrier, and good workmanship are important, given the lower temperature of the duct wall. Internal insulation does not require a vapour barrier, however space moisture will condense on the external duct surfaces if insulation is not continuous at the joints, and all joints and seams must be sealed to prevent moisture from entering the ducts.

The lower dewpoint temperature in the building brought about by cold air supply largely prevents an increased threat of condensation on the duct surfaces of the cold air supply system passing through conditioned spaces. The threat does increase, however, in unconditioned spaces and in spaces subject to large amounts of infiltration (eg foyers). Additionally, lower supply air temperature results in an increased supply-to-surroundings temperature differential, increasing the heat gain.

For ducts passing through conditioned spaces, the temperature difference across the duct wall may be 40% to 70% greater in a cold air distribution system when compared to a conventional system, while the surface area is typically be 14% to 40% less. As a result, with equal insulation levels, the heat gain for a given cold-air system typically varies from 15% less to 40% greater than for a conventional design. Increased insulation thickness is normally specified, reducing heat gain to between 40% and 80% of a conventional system. Insulation for ducts passing through conditioned spaces is therefore usually increased from 25 mm thickness for 13°C supply air to 50 mm for 7°C. However, because of the reduced supply air quantity, supply air temperature rise in the cold-air system will be equivalent to or slightly greater than that of a conventional system. Moreover, the total heat rise further increases as the supply airflow rate decreases at partial load in VAV systems. This increased temperature rise at low loads may help offset any reduction in diffuser performance resulting from reduced diffuser discharge velocity.

2.6 Diffusers

The reduced supply air temperature and lower airflow rate of cold air distribution systems means that there is an increased threat of diffuser dumping and of insufficient air movement, which can lead to both draughts and stagnation, causing discomfort and poor indoor air quality.

To overcome these threats, many cold air distribution systems are designed with fan-powered mixing boxes to mix recirculated room air with cold primary air, raising the supply air to a standard design temperature of about 13°C before delivering it to the diffusers. However, the many small, relatively inefficient fans and motors in this solution offset any fan energy savings from the reduced supply air temperature. Direct supply of cold air to properly selected cold air diffusers is therefore preferred.

The primary difference between conventional and cold air diffusers is that the cold air diffusers must operate over a wider range of temperatures and flow rates.

Ideally, the supply air jet from a ceiling diffuser should diffuse and mix along the ceiling above the occupied zone. At cold air temperatures, however, the greater density of the supply air stream may overcome the Coanda-effect to the ceiling, causing the supply air stream to separate from the ceiling before adequate mixing has occurred, resulting in thermal discomfort and poor indoor air quality due to draughts where dumping occurs and a lack of adequate mixing in the stagnant areas of the space. The threat of dumping and stagnation is even greater for side-blow diffusers (which typically discharge a thick bundle of cold air over a long throw) or for freely suspended diffusers (where airflow cannot attach to the ceiling).

Diffusers supplying cold air must be selected to avoid cold air dumping, including when VAV systems turn down. They should also provide increased effective air changes per hour in spaces with low airflow rates so as to counter the increased threat of stagnation. Additionally, at high airflow rates, the supply air streams should be characterised by both rapid supply air stream temperature equalisation and strong air speed decay, so as to prevent draughts by ensuring that excessive room air motion does not occur and that cold supply air streams are not deflected into the occupied space.

Cold air diffusers also need to be selected so as to prevent condensation on the diffuser surface.

For cold air distribution systems, dumping, stagnation, draughts and condensation are usually prevented by selecting diffusers with a high induction ratio compared to standard diffusers[3].

2.6.1 High Induction Diffusers

Int-Hout[4] advises “the challenge is to have sufficient induction to warm the cold air up before it loses momentum, and Coanda”. Diffusers with highly inductive discharge characteristics rise to this challenge and also provide a number of other benefits essential for proper air distribution from cold air systems:

- *Reduced risk of dumping at large $\Delta T_{supply\text{-}room}$*

  High induction of secondary room air into each primary supply air stream produces rapid supply-to-room temperature differential equalisation. As a result, the supply air streams close to the diffuser are of almost equal temperature, and hence buoyancy, to the room air, producing *stable supply air stream patterns* (ie no dumping), even where Coanda attachment to a ceiling is not possible.

- *Reduced draught risk in high specific airflow applications*

  Highly inductive diffuser discharge transfers kinetic (velocity) energy from the low mass flow rate primary air streams discharged at high velocity to the high mass flow rates of the entrained secondary air, bringing about rapid discharge velocity decay (due to the conservation of energy). Within a short distance from the diffuser the velocity of the supply air stream (primary air plus secondary air) is strongly reduced, thereby reducing the draught risk in spaces with *high specific airflow rates* (L/s/m^2), such as high heat load perimeter zones, or where *short throws* are required. Low air stream velocity also reduces the risk of air stream deflection into the space by clashing air streams, close by walls, protruding beams, or airflow onto furnishings (eg desks or shelves against walls). Highly inductive diffusers are suitable for applications with high specific airflow rates and/or closely spaced diffusers.

- *Reduced stagnation risk in low specific airflow applications*

  Hassani et al[5] caution that cold air distribution systems in which diffusers have low inlet momentum may cause early separation (ie dumping) of the cold air jets. This threat is especially great in areas with low specific airflow rates or where long throws are required. Highly inductive diffusers should, therefore, be selected to discharge primary air at a high velocity (ie high momentum). The high induction diffuser discharges this air in a multitude of highly turbulent and highly inductive supply air streams, causing large masses of secondary room air to be strongly entrained, vastly increasing the mass flow rate of each supply air stream. This achieves *long throws* despite the lower air stream velocity, since throw is a function of the high air stream momentum (momentum = mass flow rate x velocity). Strong entrainment of room air into the supply air streams also increases the effective air changes per hour, thereby preventing stagnation in spaces with *low specific airflow rates* ($L/s/m^2$). Highly inductive diffusers are suitable for applications with low specific airflow rates and/or diffusers spaced far apart.

- *Large VAV turndown range*

  The suitability of high induction diffusers to both high and low specific airflow applications, as well as the stability of the air streams of such diffusers over a wide range of discharge airflow rates, makes them an excellent choice for the large VAV turndown ratios required by cold air distribution systems.

- *Reduced diffuser condensation threat*

  Highly inductive diffuser discharge strongly increases the airflow across the diffuser surface, bringing about *raised diffuser surface temperature* and *elevated air speeds* across the diffuser surface. These two factors greatly reduce the threat of surface condensation.

- *High levels of comfort*

  Stable supply air streams of equal temperature to the room air achieve *uniform temperature distribution*, both vertically and horizontally, throughout the space, and *low air speeds*. These two factors yield high *air diffusion performance index* (ADPI) percentages and low *draught rate* (DR) percentage dissatisfied (PD) values, equating to excellent levels of *thermal comfort* and *predicted mean vote* (PMV) for the thermal environment achieved.

- *High Air Change Effectiveness*

  Whilst not conclusively proven, research data suggests that in cooling mode, high level air distribution diffusers that yield high ADPI percentages produce air change effectiveness (ACE) values approaching 1.0 (ie perfect mixing)[6]. High induction diffusers therefore can be expected to provide extremely good levels of outdoor air mixing to occupants. In heating mode, air change effectiveness values for high level air distribution systems reduce, due to stratification, resulting in less effective delivery of outdoor air to the occupants. For high induction diffusers, this air change effectiveness reduction is less than for standard diffusers, as can be seen in research by Manning et al[7]. This is due to the strong dilution of warm supply air with cooler room air, which produces reduced supply air stream buoyancy, and hence better mixing to a low level in the room.

- *High Green Star Rating*

  Both the high *air change effectiveness* (or ventilation effectiveness) and low *draught rate* (and hence high PMV for thermal environment) achieved by high induction diffusers assist in maximising points earned under the Australian Green Star Rating Design Tool for indoor air quality and thermal environment.

2.6.2 Diffuser Induction Ratio

The induction ratio is the amount of induced secondary air relative to primary airflow rate. This value, for a diffuser, is dependent on the discharge characteristics of the diffuser and on the distance from the diffuser at which the induction measurements are made. In particular:

- An induction system upstream of discharge into the space increases the induction at the point of discharge into the space.
- At the point of discharge into the space, breaking the supply air up into a multitude of individual, highly turbulent free streams significantly increases induction of room air into the supply air streams further in the space.
- Induction increases the greater the distance from the diffuser, due to entrainment of room air into the supply air streams.

A number of diffuser designs provide higher induction performance than standard diffusers. In preference to standard diffusers, such as linear diffusers, circular diffusers, four-way blow diffusers or side-way blow diffusers, three types of *high induction diffusers* are usually considered for cold air distribution systems:

- Pre-induction diffusers
- Multi-nozzle diffusers
- Swirl diffusers

<u>Pre-Induction Diffusers</u>

Pre-induction diffusers comprise multi-nozzle discharge into a pre-induction chamber (Figure 1). Secondary air, drawn from the ceiling void or directly from the room, is induced into the pre-induction chamber, before being discharged together with the primary air into the space, typically through slots or louvres. The pressure drop of these slots or louvres must be low so as not to impair the performance of the pre-induction chamber. Consequently, the slots or louvres cannot be designed to break up the discharged air into highly turbulent individual air streams. Instead, the air is discharged as bundles or sheets.

The induction ratio in the supply air stream at any distance “x” from the diffuser is a function of the induction ratio in the pre-induction chamber *plus* the induction of room air into the supply air stream up to point “x”. Induction at the point of discharge is higher than that of a diffuser without a pre-induction chamber (due to induction of room air in the pre-induction chamber) but does not increase at a high rate beyond this point.

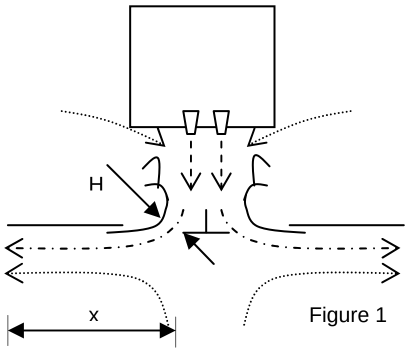


Figure 1

The following example demonstrates the induction ratio formula for a proprietary 2-slot pre-induction diffuser:

$$I = I_0 + 0.55 \bullet \sqrt{\frac{x}{H}}$$

where Io = 2.54 @ 16 L/s primary airflow @ 1.2 m nominal length

Io = 0.98 @ 75 L/s primary airflow @ 1.2 m nominal length

Io, as per manufacturer’s catalogued data

$0.55 \bullet \sqrt{\frac{x}{H}}$, based on Recknagel et al[8]

<u>Multi-Nozzle Diffusers</u>

Multi-nozzle diffusers (Figure 2) discharge air from a series of miniature nozzles, to create a multitude of individual and highly turbulent, high velocity free air streams each of which strongly induces room air, thereby using the entire room as an induction mixing chamber. Induction is often further increased by alternating discharge jets left and right, for ceiling diffusers, or by angling discharge jets at different angles, for wall diffusers, to further break up the supply air. Such diffusers are typically characterised by strongly escalating induction as distance from the diffuser increases.

The following example demonstrates the induction ratio formula for a proprietary 2-slot multi-nozzle diffuser with alternating nozzle pairs:

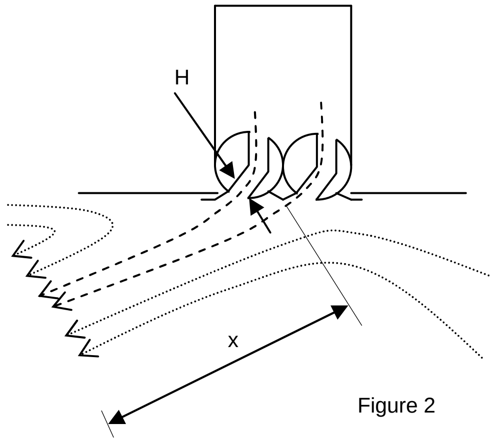


Figure 2

$$I = 0.12 \bullet \frac{x}{H}$$

where x = distance from diffuser,

H = height of 1 nozzle

Swirl Diffusers

Swirl diffusers (also known as *twist outlets* or *vortex diffusers*) produce an intense swirling motion at the diffuser discharge face, producing high aspiration of room air into the swirling supply air stream (Figure 3). Additionally, the supply air is broken up into a multitude of free streams, each one of which is highly inductive. The combination of these two factors results in extremely high diffuser induction ratios with increasing distance from the diffuser.

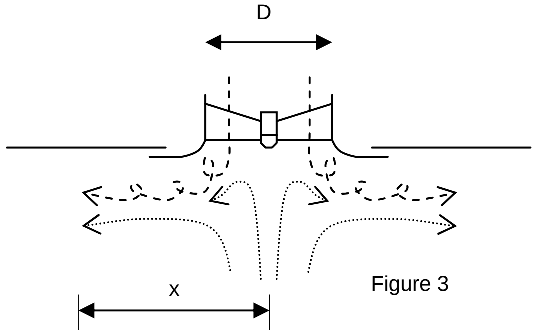


Figure 3

The following example demonstrates the induction ratio formula for a proprietary swirl diffuser:

$$I = 1.9 \bullet \frac{x}{D}$$

where x = distance from diffuser

D = nominal neck diameter

Induction Comparison

The induction ratios achieved by the above three ceiling diffusers can be compared (Table 2):

Table 2	***Pre-Induction Diffuser*** Nominal Length = 1.2 m H ≈ 15 mm I_0 = 2.54 @ 16 L/s primary air I_0 = 0.98 @ 75 L/s primary air $Pt_{75 L/s}$ = 200 Pa $Lw_{75 L/s}$ = 42 dB(A) $Lp_{75 L/s}$ = NC36 per diffuser (based on 10 dB room absorption)	***Multi-Nozzle Diffuser*** Nominal Length = 1.2 m H ≈ 10 mm Suitable for 22 to 80 L/s primary diffuser airflow $Pt_{75 L/s}$ = 30 Pa $Lw_{75 L/s}$ = 35 dB(A) $Lp_{75 L/s}$ = NC29 per diffuser (based on 10 dB room absorption)	***Swirl Diffuser*** Nominal Diameter = 180 mm D ≈ 180 mm Suitable for 21 to 80 L/s primary diffuser airflow $Pt_{75 L/s}$ = 54 Pa $Lw_{75 L/s}$ = 38 dB(A) $Lp_{75 L/s}$ = NC32 per diffuser (based on 10 dB room absorption)
x [m]	$I = I_0 + 0.55 \bullet \sqrt{\frac{x}{H}}$	$I = 0.12 \bullet \frac{x}{H}$	$I = 1.9 \bullet \frac{x}{D}$
0.0	I ≈ 1.0 to 2.5	I ≈ 0.0	I ≈ 0.0
0.5	I ≈ 4.2 to 5.7	I ≈ 6.0	I ≈ 5.3
1.0	I ≈ 5.5 to 7.0	I ≈ 12.0	I ≈ 10.6
2.0	I ≈ 7.3 to 8.9	I ≈ 24.0	I ≈ 21.1
3.0	I ≈ 8.8 to 10.3	I ≈ 36.0	I ≈ 31.7

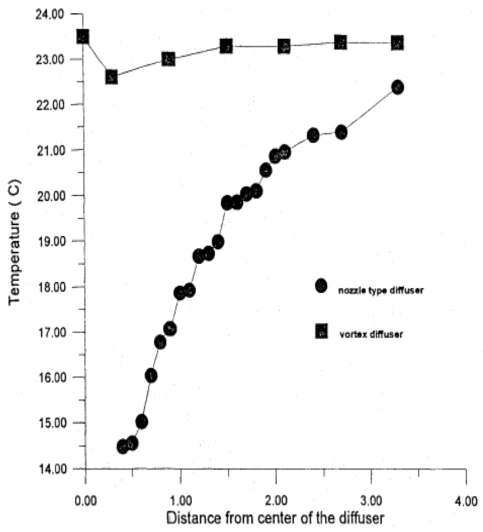
While the induction ratio (I = 1.0 to 2.5) for the pre-induction diffuser is higher at the point of discharge (x = 0 m) than for the multi-nozzle and swirl diffusers (I = 0), the induction ratios of the latter two increase sharply within a short distance and surpass that of the pre-induction diffuser for distances greater than x ≈ 0.5 m from the diffuser.

For example, at x = 3.0 m, the induction ratio of the multi-nozzle diffuser is 309% greater, and that of the swirl diffuser is 260% greater, than the induction ratio of the pre-induction diffuser operating at its maximum primary airflow rate of 75 L/s. Moreover, the high induction ratios of the multi-nozzle and swirl diffusers are each achieved at significantly lower total pressure (30 Pa and 54 Pa, respectively, vs 200 Pa for the pre-induction diffuser) and at lower sound pressure levels (NC 29 and NC 32, respectively, vs NC 36 for the pre-induction diffuser).

The operating pressure of the multi-nozzle and swirl diffusers is similar to that of standard diffusers, and will, therefore, not compromise fan energy savings. The significantly increased pressure requirement of the pre-induction diffuser, though, may more than offset the reduction in fan size[9].

<u>Induction Verification</u>

In laboratory tests, Hu et al[10] measured the air stream temperature of the above mentioned proprietary swirl diffuser (which they refer to as a "vortex" diffuser) relative to an alternative high induction pre-induction diffuser (referred to as a "nozzle-type diffuser") comprising a multi-nozzle induction system upstream of four-way blow slots. In each case, supply air temperature T_S = 8°C and room temperature T_R = 24°C (ie $\Delta T_{S\text{-}R}$ = -16 K). The significantly higher temperatures measured in the supply air stream of the swirl diffuser (Figure 4) clearly demonstrate the superior induction characteristics within the room of the swirl diffuser relative to the pre-induction diffuser.



Comparison of the vortex diffuser and the nozzle-type diffuser for case 5 and case 8: temperature profiles recorded at the locations 0.05 m (0.16 ft) below the ceiling. Figure 4

Hu et al's air stream temperature measurements may also be used to verify the above swirl diffuser induction ratio formula. Temperature in the supply air stream may be expressed as:

$$T(x) = T_R - \frac{1}{1+I} \bullet (T_R - T_S)$$

where $I = 1.9 \bullet \frac{x}{D}$ for the swirl diffuser

Table 3 compares measured and calculated air stream temperature at distance x from the diffuser.

Table 3

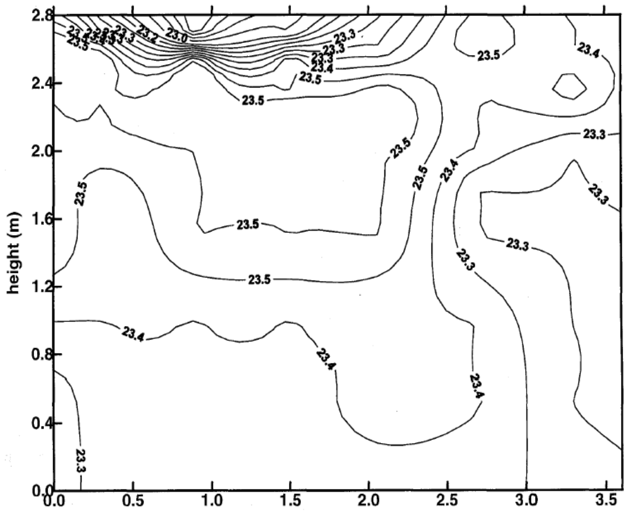
	T(x) [°C] T_S = 8°C, T_R = 24°C	
***x** [m]*	Measured by Hu et al	$T(x) = T_R - \frac{1}{1+I} \bullet (T_R - T_S)$ where $I = 1.9 \bullet \frac{x}{D}$, $D = 180mm$
0.91	22.9	22.5
1.52	23.2	23.1
2.13	23.2	23.3
2.74	23.4	23.5
3.35	23.4	23.6

The above air stream temperature comparison of Hu et al's laboratory tests and the calculated values based on the swirl diffuser induction ratio formula shows excellent correlation between the two results for distances greater than 0.91 m from the diffuser centre-line. This comparison verifies the validity of the swirl diffuser induction ratio formula

$$I = 1.9 \bullet \frac{x}{D}$$

2.6.3 Room Temperature Distribution

Figure 5, adjacent, from Hu et al represents the right-hand half of the test laboratory (the left-half can be taken as the mirror image). It can be seen that despite the large supply-to-room temperature differential of $\Delta T_{S\text{-}R}$ = -16 K, the swirl diffuser (mounted at the top of the y-axis in figure 5) provided a temperature distribution in the occupied space of 23.4°C ±0.1 K. Hu et al remarked that



0.0 = room centreline

Figure 5

the temperature distribution from the “vortex” diffuser was extremely uniform and concluded that the uniform *vertical* temperature distribution of this diffuser also makes it suitable for high rooms.

2.6.4 ADPI

The swirl diffuser in Hu et al’s tests provide an ADPI of 97% for both 47.2 L/s (specific airflow rate = 1.56 L/s/m^2) and 70.8 L/s (specific airflow rate = 2.34 L/s/m^2) primary airflow from 2.8 m discharge height at $\Delta T_{S\text{-}R}$ = -16 K. Despite the high supply-to-room temperature differential and the low specific airflow rates, the occupant comfort levels are extremely high and are maintained over a wide airflow rate range. This is due to the swirl diffuser’s high induction ratio, which creates uniform temperature distribution, increases effective air changes per hour in the space, and breaks down supply air velocity to provide draught-free air movement. In their laboratory comparison of cold air diffusers operating at $\Delta T_{supply\text{-}room}$ = -15 K, Kashirajima et al[11] also measured extremely high ADPI values for the swirl diffuser tested and showed that the swirl diffuser maintained ADPI values well above 80% even when turned down to 20%, whereas the performance of the comparative four-way blow and circular diffusers tested dropped below 80% ADPI for turndown to 30% and 37%, respectively. Both the tests from Hu et al and Kashirajima et al demonstrate that extremely high ADPI values are achieved by swirl diffusers, even when operating at a large cooling supply-to-room temperature differential and when turned down to extremely low specific airflow rates.

2.6.5 Condensation

Hu et al showed in their tests that due to the high induction, condensation does not occur on the swirl diffuser, even for 5°C supply air temperature and 36°C room temperature at 90% RH, thereby verifying that high induction swirl diffusers are well suited to preventing diffuser condensation when supplying cold air.

3.0 CONCLUSIONS

Cold air distribution systems provide potential to reduce first costs and to provide ongoing energy cost savings. This is achieved by size reductions in fans, ducts, pumps and power supply. Building size may also be reduced, due to reduced slab-to-slab heights, and nett lettable space may be increased, due to reduced riser sizes. Indoor air quality and comfort levels are often also improved. However, depending on the application, reduced free cooling potential and the potential need for reheat to prevent zone overcooling may negate these advantages. Additionally, cold air distribution may not be suitable for applications where generation of low chilled water temperatures is not practical, where space relative humidity must be maintained above 40% to 45%, or where very high ventilation volumes are required. Cold air distribution systems require increased duct insulation, to reduce the threat of duct condensation and to minimise supply air heat pickup. Particular attention must also be given to air diffuser selection, with the use of high induction diffusers, such as multi-nozzle diffusers or swirl diffusers, being preferred to ensure that occupant comfort and indoor air quality are not compromised by the increased supply-to-room temperature differential and reduced specific airflow rates.

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