Basics of Room Air Distribution

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Study Objectives

- Main issues of thermal comfort
- Principles of air distribution as they relate to human comfort
- Principles of space air distribution
- Functions of the different types of air distribution devices
Where We Are Today:
Thermal Interchange Between People and Environment

- **RADIATION**
- **CONVECTION**
- **EVAPORATION**

*HIGHLY AIRFLOW DEPENDENT*
*SIMILAR IN MAGNITUDE*

**EVAPORATION DRIVEN BY**
- \( P_{\text{skin}} - P_{\text{dewpoint}} \)
- **Air Velocity**

**CONVECTION DRIVEN BY**
- \( t_{\text{skin}} - t_{\text{room}} \)
- **Air Velocity**
Temperature and Humidity Comfort Zone

- Dry-bulb temperature, °F
- Relative humidity, rh
- Thermal radiation
- Air movement, fpm
- Insulation value of clothing, clo
- Activity level, met
- Direct contact with surfaces not at body temperature
ASHRAE Standard 55-2004

Thermal
Environmental Conditions for Human Occupancy
For 80% Thermal Acceptance

- Activity levels are between 1 and 1.3 met.
- Clothing is near 0.5 or 1 clo.
- Average Air speeds are below 40 fpm.
Avoid in humid climate to avoid mold
Ideal target conditions

Humidity Ratio

Dew Point Temperature, °F

Operative Temperature, °F

Humidity Limit 0.012 humidity ratio

1.0 Clo

0.5 Clo

No Recommended
Lower Humidity Limit

10% RH

PMV Limits
Low humidity a health issue
Graphical Solution Program
Thermal Comfort

ASHRAE Standard 55-2010 mandates a maximum 5.4°F vertical temperature stratification in Occupied Zone

Velocities within the occupied zone shall be ≤ 50 FPM

Floor to Occupants’ Head Level
(3.5 ft. for seated, 6 ft. for standing occupants)
Comfort Economics

• ASHRAE Journal, June 2008

Figure 1: Life-cycle building costs breakdown.

Figure 2: Life-cycle building costs breakdown with people (salaries).
Local Discomfort Effects

Verify that local discomfort effects have been considered and are not likely to exceed Standard 55 limits. When local discomfort effects are likely to occur, verify that calculations were performed to demonstrate that local discomfort effects are predicted to result in less than 10% dissatisfied occupants.

<table>
<thead>
<tr>
<th>Local Discomfort Effect</th>
<th>Not Likely</th>
<th>Calculations Performed</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radiant Temperature Asymmetry</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Vertical Air Temperature Difference</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Floor Surface Temperature</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Draft</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

* Operative temperature includes radiant effects. See Standard 55.
Indoor Air Quality
Indoor Air Quality

- Standing Standard Project Committee 62.1
- Residential Committee is 62.2
- Current Standard is 62.1-'13
- Several addenda for the ’13 version have already been approved, and will be published in a compiled addenda, every 18 months (or so).
“air in which there are no known contaminants at harmful concentrations as determined by cognizant authorities and with which a substantial majority (80% or more) of the people exposed do not express dissatisfaction”
Two Compliance Paths:

• Ventilation rate procedure
  – Prescribed rates in Table 6-1: cfm/person plus cfm/square foot
  – Office: 5 cfm/person plus 0.06 cfm/ft²
  – Default = 17 cfm/person in an office
  – Only 5.5 cfm/person in Hotel multi purpose space

• Indoor air quality (IAQ) procedure
IAQ Standard

- Standard 62 is on continuous maintenance.
- Continuous and incremental changes are in progress.
- It will attempt to be in coordination with building codes.
- A Guideline document for designing systems above minimum requirements is being created.
- Users Manual is available now.
- The IMC has referenced 62.1 in the 09 release of the mechanical code.
- There seems to be minimal public awareness of the dynamic nature of the Standard.
Acoustics
Acoustics:

- AHRI 885-08 acoustical application standard.
- AHRI 880-08 air terminal test standard.
- AHRI 260-01 ducted equipment except air terminals.
- ASHRAE 70, air diffuser performance.
- Acoustical quality suggests the use of RC (or newer measures) rather than NC. Many acousticians are heading back to dBA!
- LEED V4 includes acoustical credits and requirements, and references AHRI 885.
- A new ruling by AHRI will change everyone’s discharge sound ratings considerably.
End Reflection

- Low frequency sound traveling down a duct will partially reflect back when encountering a rapid change in area.
- The smaller the duct, the greater the effect.
- It can be as much as 10dB at 125Hz. It is much less at higher frequencies.
- Since NC is usually set at 125Hz, reported NC can go up as much as 10NC.
- Most importantly, Specifying Engineers should be modifying their discharge sound requirements to reflect the new data.
End Reflection
Sound Specifications

• Should be based on clearly stated assumptions.
• Should reflect real project needs, not any manufacturer’s data and use currently accepted application factors.
• If duct lining is used – require: ”NC shall be determined in accordance with AHRI 885-08, Appendix E”, otherwise specify octave band sound power.
• Specifications need to be modified to account for the new reported data.
• Over-silencing increases both initial costs and operating costs, and may hinder proper IAQ performance.
Air Distribution Principles
Outlet Types

- **Grille:** movable or fixed blades to direct the air and adjust width of jet.
  - Works well in floor, sill and wall.
- **Slot diffuser:** long, narrow slot producing rapid diffusion; use a grille for the ceiling.
- **Ceiling diffuser:** rings of pressed metal to supply air across the ceiling.
Outlet Accessories

- Dampers: opposed blades adjust flow while parallel blades also adjust direction.
  - Be careful of significant flow adjustment at damper due to noise. Quieter, less noticeable air noise, when flow is adjusted at branch.

- Directional vanes to collect or direct airflow.
Outlet Selection

- Understand the space, esthetics, and air supply access.
- Determine quantity of air to be supplied.
- Select type: air quantity, throw, obstructions in the space, and architecture.
- Locate outlets for uniform distribution.
- Select proper size.
Return or Exhaust Inlets

- Any style of outlet. Free area for air transfer is required.
- No effect on general flow around the space.
- Position so primary air is not collected.
- Good in stagnant zone.
Understanding The Terminology

**Primary Air Jets** - Air jets from free round openings, grilles, perforated panels, ceiling diffusers and other outlets can be defined by three variables.

- Throw
- Drop
- Spread
Understanding The Terminology

DROP

THROW

Primary Air

Primary Air
Understanding The Terminology

Spread - is defined as the divergence of the airstream in a horizontal or vertical plane after it leaves the outlet.
Understanding The Terminology

Coanda Effect - a negative or low pressure area is created between the moving air mass and the ceiling at or near the supply air outlet. This low pressure area causes the moving air mass to cling to and flow close to the ceiling surface and increases the throw.
Understanding The Terminology

Understanding primary air jet variables enables

• Accurate prediction of room air flow
• Improvement of thermal comfort
• Proper selection of grilles, registers and diffusers
• Adherence with ASHRAE Ventilation Std 62.1 as a Leed PREREQUISITE, and in the 2009 IMC
Understanding The Terminology

The Basis of Catalog Performance Data

• Throw – The horizontal or vertical axial distance an airstream travels after leaving an air outlet, usually assumes a surface adjacent to the air outlet

• Pressure – Can be total pressure or static pressure

• Sound – Can be either NC or Octave Band data
Throw

- Throws are cataloged for 150, 100 and 50 fpm terminal velocities.
- Throws should be selected so that jets do not collide, but have sufficient projection for the area to be served.

<table>
<thead>
<tr>
<th>Neck Vel FPM</th>
<th>Air Flow CFM</th>
<th>Pt &quot;WG&quot;</th>
<th>Ps &quot;WG&quot;</th>
<th>Throw ft</th>
<th>NC</th>
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</thead>
<tbody>
<tr>
<td>200</td>
<td>39</td>
<td>0.004</td>
<td>0.002</td>
<td>1 - 1 - 4</td>
<td>-</td>
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<tr>
<td>300</td>
<td>59</td>
<td>0.009</td>
<td>0.004</td>
<td>1 - 3 - 6</td>
<td>-</td>
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<tr>
<td>500</td>
<td>98</td>
<td>0.026</td>
<td>0.010</td>
<td>3 - 5 - 8</td>
<td>-</td>
</tr>
<tr>
<td>600</td>
<td>118</td>
<td>0.037</td>
<td>0.015</td>
<td>4 - 6 - 9</td>
<td>-</td>
</tr>
<tr>
<td>700</td>
<td>137</td>
<td>0.050</td>
<td>0.020</td>
<td>4 - 6 - 10</td>
<td>-</td>
</tr>
<tr>
<td>800</td>
<td>157</td>
<td>0.066</td>
<td>0.026</td>
<td>5 - 7 - 10</td>
<td>12</td>
</tr>
<tr>
<td>900</td>
<td>177</td>
<td>0.083</td>
<td>0.033</td>
<td>6 - 8 - 11</td>
<td>16</td>
</tr>
<tr>
<td>1000</td>
<td>196</td>
<td>0.103</td>
<td>0.041</td>
<td>6 - 8 - 11</td>
<td>19</td>
</tr>
<tr>
<td>1100</td>
<td>216</td>
<td>0.124</td>
<td>0.049</td>
<td>7 - 8 - 12</td>
<td>22</td>
</tr>
</tbody>
</table>
Pressure – Air outlet pressure data is required to properly size the air delivery system within a building.

• **Static Pressure** – The outward force of air within a duct, measured in inches of water column.

• **Velocity Pressure** – The forward moving force of air within a duct, measured in inches of water column.

• **Total Pressure** – The sum of the velocity and static pressures, expressed in inches of water column and can be obtained by use of a pitot tube.

\[ P_T = P_V + P_S \]
Sound levels reported for diffusers are conducted in accordance with ASHRAE Standard 70.

- Catalog sound data assumes 10 diameters of straight duct.
- Room absorption is assumed to be 10dB in all bands.
- In practice however, room sound levels are probably 5 NC higher than reported.
Air Distribution, Poor Pattern
Air Distribution, Good Pattern
Understanding ADPI
ADPI

- ADPI is the percentage of points within the occupied zone having a range of effective draft temperatures of -3° to +2° of average room temperature at a coincident air velocity less than 70 FPM.
- ADPI is essentially a measure of the degree of mixing in zones served by overhead cooling systems.
- When air distribution is designed with a minimum ADPI of 80% the probability of vertical temperature stratification or horizontal temperature non-uniformity is low and conformance with ASHRAE Standard 55 (Thermal Comfort) requirements is high.
- ADPI does not apply to heating situations or ventilation-related mixing.
• ADPI selection using $T_{50} / L$ was developed in the ‘60s where $L$ is the distance to the nearest wall or halfway to the nearest air outlet.

• A relationship was found between 50 FPM/min isothermal throw and cooling throw,

• Using this table engineers can assure clients that diffuser selections will provide acceptable mixing and air change effectiveness.
Perforated 24X24, 10” inlet, 4 way, 20° Delta-T

Spacing for 80% ADPI

CFM/Sq.Ft.

1/2 Unit Separation Distance

NC=35

CFM

Range

420
350
300
250
160

0.0 0.5 1.0 1.5 2.0 2.5 3.0 3.5 4.0 4.5 5.0

CFM/Sq.Ft.
Prism, 24”x24”, 10” inlet, 20°ΔT
Spacing for 80% ADPI

1/2 Unit Separation Distance, L

cfm/SqFt
Room Air Speed
Issues and Factors

• Standard 55 says thermal comfort can be achieved with 0 fpm air motion.
• Uniform air temperatures indicate good mixing when loads are present.
• With conventional (well-mixed) systems, room air speed is driven primarily by room loads when air supply is below 0.9 CFM/sq. ft. and air diffusion is adequate per ASHRAE sponsored research
• Partitions can provide excellent comfort with ceiling diffusers when cooling.
ASHRAE Journal, 2004, on overhead air distribution selection

Legal Issues

Protection Against Liability For Poor Diffuser Selection

By Daniel In-Hoult III, Member ASHRAE

The manner of achieving an acceptable Air Diffusion Performance Index (ADPI) has been well understood for more than 30 years. Unfortunately, complaints of discomfort abound. It is not uncommon for building occupants who work in the same space to complain that they are "too hot" and "too cold."

Many unconventional designs and new technologies have been used to correct this apparent problem, including displacement ventilation, underfloor pressurized plenum air distribution, occupant level conditioning systems, etc. While all of these strategies assume that conventional overhead air distributions is unlikely to provide acceptable environments, this appears to be a false premise.

According to this and to technical papers as early as 1976, given a properly designed and well-placed overhead air diffuser, and an HVAC supply system capable of meeting the loads in the space, it is possible to achieve a system of either VAV or constant volume air distribution that can respond to variations in localized loads from 20% to 100% of designed maximum loads at a variation in space temperatures that most occupants will not notice. This system also will provide a ventilation effectiveness of 100% if the ventilation air supplied at the ceiling will be delivered at the occupant's breathing zone.

The way wall plate diffusers are also well documented.

So why are building occupants complaining they're uncomfortable? The likely culprit is a "poor diffuser selection" which can lead to a number of problems. The first is "shorting" - low air flow, low air velocities may not be high enough to create the "cross flow" necessary to overcome the warm air being discharged. This causes cold air to drop into the space. As a result, it's cold under the diffuser, warm at the midpoint between diffusers, and cold at the periphery. This condition creates a diagonal movement in the space. Another problem occurs at very high airflows. Air gets into the middle between diffusers, causing cold air to drop into the space (it was hot earlier). The increased induction at the recently cold spot under the diffuser now creates an unsafe, warming that location.

At the perimeter where cool executive offices may be located, even some things can happen. In winter, air is being discharged at 15% of cooling velocities at discharge temperatures of 70°F (21°C). Since the warm jet has too much buoyancy and too little projection to mix with cold air that will spill down the window, there will be an 8°F to 10°F (4.4°C to 5.6°C) temperature difference between 6 ft. and 2 ft. (0.1 m and 0.6 m) from the floor to the midpoint of the room (virtually to the midpoints of ASHRAE Standard 55, Thermal Environmental Conditions for Human Occupancy). In summer, heat from the window stratifies the air near the ceiling. Cold air from the diffuser, which is often close to blue doors, stratifies on the floor. In both cases, at 63 in. (1.6 m) above the floor, where the thermostat is located, it is 79°F (26°C).

Problems like these can lead to liability and expense.

Here is a just a few examples. In Baltimore, condominium owners with two-story living rooms, the designer supplied air from the ceiling. Since the air was too hot in cold months, it stratified at the ceiling, resulting in a temperature of 50°F (10°C) at the window and 60°F (16°C) at the floor. The condominium owners sued the engineer. The case was eventually settled, but not until after the engineer was forced to hire his own attorneys and experts. A similar problem occurred in a one-story house in Pennsylvania. This time, the case went to trial and the jury returned a sizable verdict against the engineer. The expert witnesses then exonerated the engineer in the lawsuit. Improper diffuser selection can also lead to problems for owners. For example, in an office building in New Jersey, a tenant successfully broke its lease after complaining about comfort problems relating to ceiling diffusers. Indeed, according to the Building Owners and Managers Association, thermal comfort related issues when misconstrued as IAQ problems were the No. 1 reason for non-renewal of leases in 2002.

The information contained in this article represents the opinion of the author and is not intended to, and does not, constitute legal advice, nor does it represent the opinion of ASHRAE or any of its bodies.
Common Overhead Heating Design

Cold Outside

Window

THERMOSTAT
Overhead Heating Perimeter Considerations:

- Maximum delta-t for effective mixing when heating from overhead, per ASHRAE handbook = \(?\).
- \( = 15^\circ F \) (90°F discharge), continuous operation.
- 150 FPM should reach 4-5 feet from the floor.
- ASHRAE 62.1 requires that ventilation be increased by 25% when heating, if the above rules are not followed.
- Throw toward and away from glass.
- Typical perimeters require only 8°F Delta-t @ 1cfm/sq.Ft.
Proper Perimeter Design
Perimeter Considerations:

See March 2007 ASHRAE Journal:
• In LEED 2009 and later, in order to get ANY LEED points, one must fully meet the Ventilation Rate Procedure calculation in ASHRAE Standards 62.1 (Ventilation)
• ASHRAE 62.1-2013 VRP requires that if heating air supplied from the ceiling is less than 15° above room temperature but does not reach within 4.5 feet of the floor at 150FPM the outdoor air supply must be increased by 25%.
• ASHRAE 62.1-2013 VRP requires that if the heating air supplied from the ceiling is greater than 15° above room temperature the outdoor air supply must also be increased by 25%.
• LEED V4 Ventilation points are again gained by increased ventilation 30% beyond 62.1 minimums.
Non-Inductive Air Distribution
Laminar and Radial Flow Outlets

- Hospital Operating Suites
- Hi-Tech Electronics and other industrial applications
- Clean Rooms
- Laboratories
UFAD: Underfloor Air Distribution

- A raised floor allows electrical and communication circuits to be easily accessed and changed.
- Air may be distributed within this space, without ductwork.

Ducted perimeter with fan powered boxes, or other techniques, depending on climate, glass load, etc..

Pressurized plenum for core
Interior Outlet Selection – 1/workstation
Perimeter outlet selection

- Perimeter solutions vary considerably
- Avoid condensing coils under the floor
- Hydronic coils often leak
- Exhaust at the perimeter to draw heat away
- Best solutions seem to be heat and cool from overhead
Displacement Systems
Select based on "adjacent zone"
Applications

Classrooms
Kitchen
Restaurant
Auditoriums
Atriums
Gyms
And more…
Non Typical Throw Analysis
Special Applications

High bay application - Ceilings over 12’ high

• Heating is a challenge due to buoyancy.
  – Take advantage of vertical stratification where possible
  – Required Heating airflow rate may exceed cooling airflow rate.
  – Keep heating supply air temperature to room temperature ΔT to a minimum

• If supplying air distribution from the ceiling, consider using round diffusers, drum louvers, or diffusers with some vertical projection.

• One cannot use ADPI to predict heating performance.
• Consider Displacement Ventilation for cooling applications
Diffuser Selection & Buoyancy

- ADPI isn’t always the best way to analyze, select and place diffusers, especially with heating and high bay applications.
- One can estimate Throw as a function of ΔT and buoyancy.
- Simple rule: Distance to 75ft/min is affected by 1%/degree(F) ΔT.

Example:
1. 20°ΔT Cooling, Vertical Down = +20% projection
2. 20°ΔT Heating, Vertical Down = -20% projection
3. 20°ΔT Heating, Along Ceiling = +20% projection
Side Wall Register Selection & Buoyancy

- Horizontal free jet:
- Vertical change @ 75ft/min is affected by 1% of 75fpm throw/F⁰ ΔT.

Example: 15°F Delta T heating

Note: T₁₅₀ is not affected by Delta-t
Entrained vs. Free Jets

- Most catalog throw data assumes jet is along a surface.

- Exceptions include drum louvers, duct mounted grilles and vertical linear diffusers.

- A free jet will be shorter than an entrained jet because it has more surface area to induce surrounding air, which shortens throw.
Special Applications

Continuous Duct Application Suggestions:

• Use multiple drum louvers, duct mounted grilles and continuous linear applications (longer than 10’).
• Size duct as large as possible (Duct inlet velocity < 1000fpm).
• If inlet velocities are less than 1000fpm, maintain constant duct size through entire length of run and balancing will be minimal.
Returns

- Typically, returns are located in the ceiling in offices.
- Returns have an almost immeasurable effect on room air flows below 1.5 cfm/sf.
- Suspended ceilings typically leak 1 cfm/sf at 0.1” differential pressure.
- Spaces with high airflow rates can benefit from low returns.
Air Distribution Summary

• LEED 2009 requires meeting Standard 62.1 for project approval—**No compliance=No Leed project designation!**
• Documented use of ADPI is the ONLY way to assure compliance to ASHRAE Std. 55 in the design phase for cooling.
• Reheat needs to be carefully considered in terms of discharge temperatures and velocities. High heating supply temperatures will void meeting Standard 55 (and loose a potential LEED point).
• Software is available to assist in selecting the best mix of products.
• LEED V4 was released last October.
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Read My Air Distribution Blog!