What A (room) Load

Dan Int-Hout
Chief Engineer, Krueger
Richardson, Texas

KRUEGER
Excellence in Air Distribution

Where We Are Today:

What Happened To Building Loads?

- Building loads are dropping all the time, both internal and through the envelope.
- Meanwhile, Architects are challenging Mechanical Engineers to achieve energy efficiency 30% better than the base systems in ASHRAE 90.1, which for large buildings is VAV / Overhead (well mixed).
- It seems, however, that engineers are still basing load assumptions on what they used 20 years ago.
- In addition, most Energy Calculation programs have not been validated for systems other than VAV overhead.
- The result is that many buildings don’t perform as calculated, resulting in distrust of energy use predictions.
- The LEED rating system is losing support as a result.

So What is the result?

- BOMA continues to report that the #1 reason for occupants not renewing the lease is dissatisfaction with the environment we create.
- Spaces are often too cold
- Acoustics are poor (typically too quiet)
- Energy bills are higher than predicted
- Productivity suffers.

Spaces are often too cold

- In order to control humidity, and still meet minimum ventilation requirements, spaces are often sub cooled.
- Reheat, while a solution to the comfort issue, is expensive.
- Perimeter zones, in winter, are often severely stratified, with it being very cold near the floor.
- All too often, space heaters are found throughout the interior, increasing significantly loads.

Acoustics are poor

- In order to achieve a good acoustical environment, it is almost always necessary to add background sound masking.
- Asking the HVAC system to provide that noise is very expensive.
- Poor acoustical treatment of surfaces compound the issue. Open ceilings, while often desired by architects, often result in poor acoustical privacy.

Energy bills are higher than predicted

- Systems have difficulty controlling humidity at low loads
- Poorly adjusted or selected air outlets don’t allow thermostats to sense room temperatures correctly
- Operators don’t know how to run the system efficiently
- Opportunities for free cooling or heat removal aren’t utilized.

Productivity suffers

- Occupants have high absenteeism from poor environmental controls
- Lack of Acoustical Privacy reduces ability to concentrate
- Inability of management to provide comfort reduces employee morale

Let’s look at some of the issues:

- Interior loads: the reality vs the “rule of thumb”.
- The cost of comfort
- The need to select and adjust air outlets properly
- Designing for low interior loads
- Taking advantage of “free cooling”
- Making overhead heating work. (and meet code)
- Predicting and Controlling Acoustics

Interior loads:
The Reality vs the "Rule of Thumb"

- I find almost all interior designs assume 1 cfm/sf (23BTUH/SF) for the air loading.
- Actual loads, as we have seen in many research projects, is closer to 0.3cfm/sf (7 BTUH/sf)
- When asked, many engineers respond with "I've never been sued for having too much capacity".

Thermal Comfort Economics

- **ASHRAE Journal, June 2008**

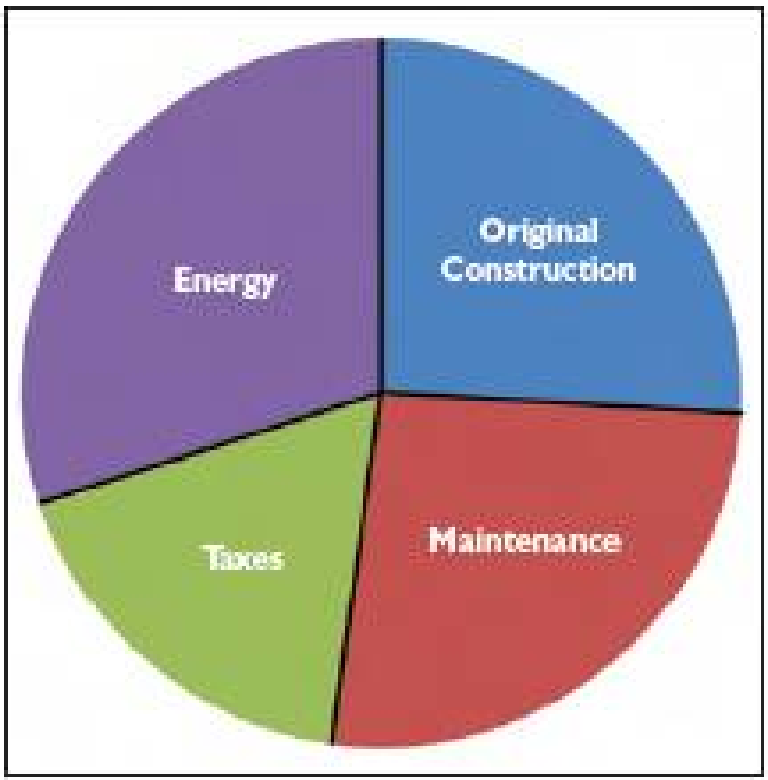


Figure 1: Life-cycle building costs breakdown.

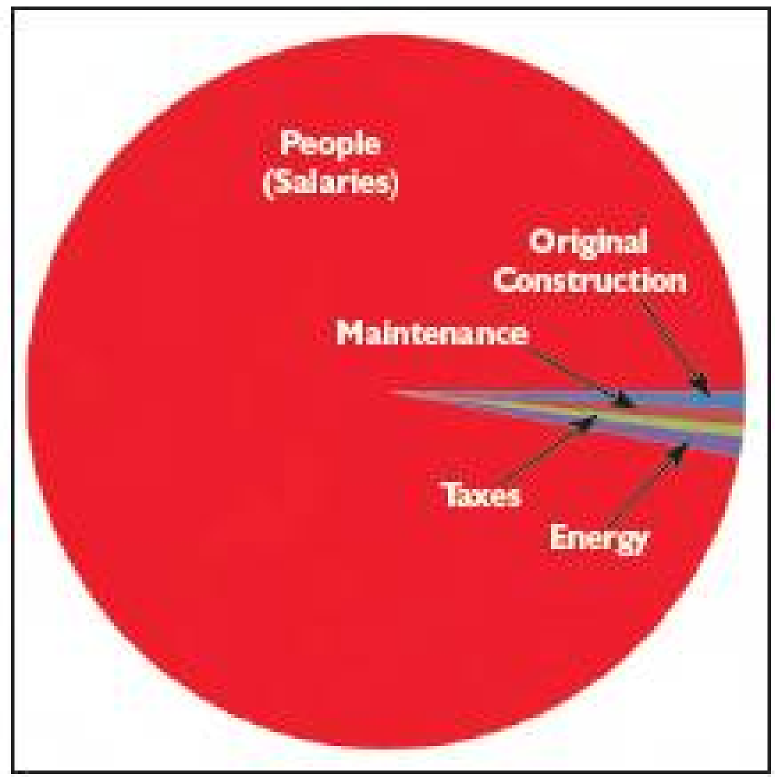


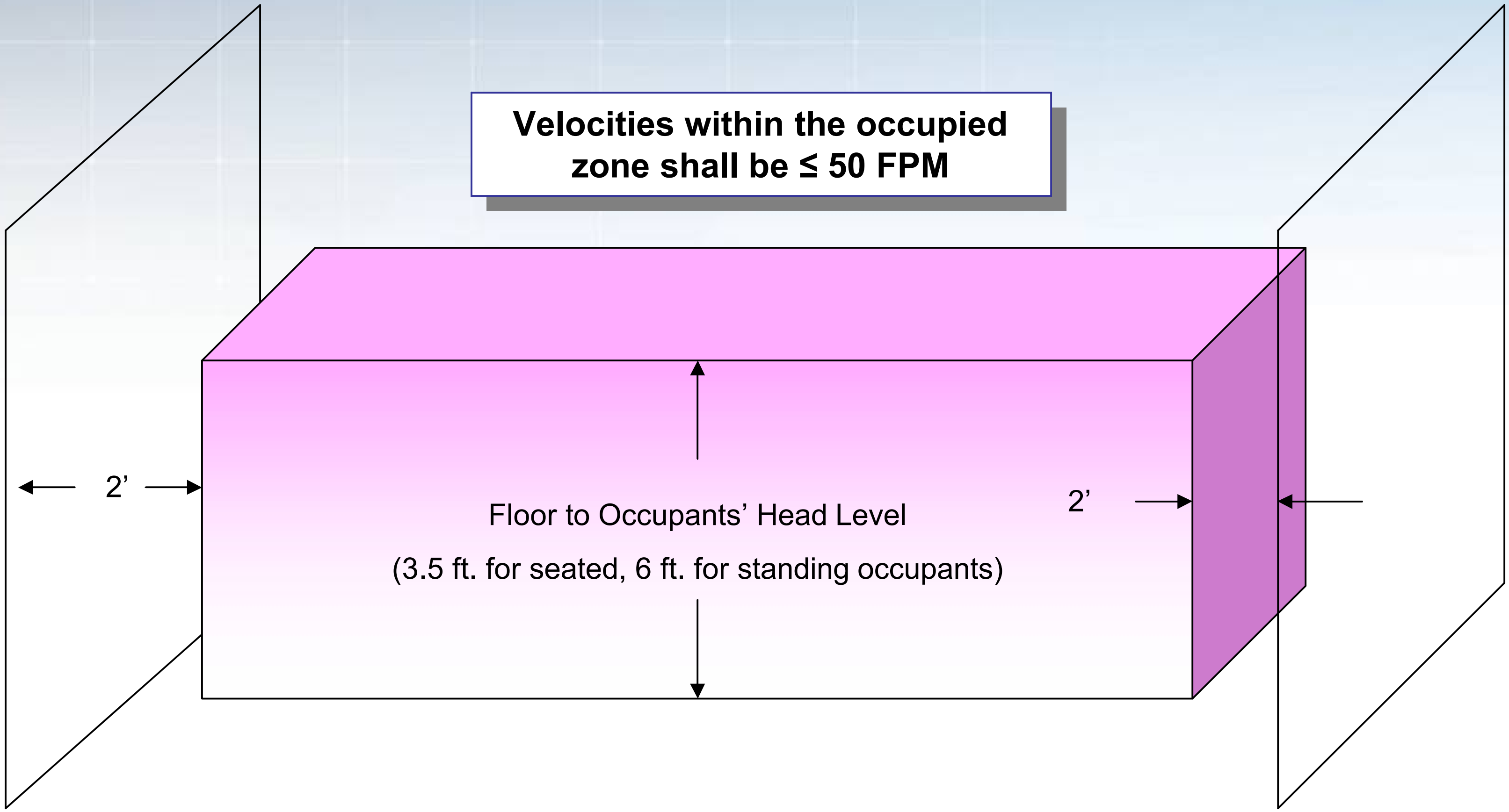
Figure 2: Life-cycle building costs breakdown with people (salaries).

The cost of comfort

- Most buildings cost about $2/ sfy to heat and cool.
- At 150SF/employee, salaries are about $200/sfy.
- Moving thermostats can save about 5% of heating and cooling costs. That is $100,000/year in a million sf building!
- Salaries in that building are $200 Million / year.

Thermal Comfort

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



The need to properly select and adjust air outlets

- The ASHRAE handbook suggests selecting diffuser to achieve an 80% ADPI at all air flow rates.
- Most diffuser layouts have diffusers too close together, resulting in drafts at full flow
- Some diffuser designs work better than others at low flows – a plaque-type is a good choice. Perforated and swirl diffusers not so much.

ADPI

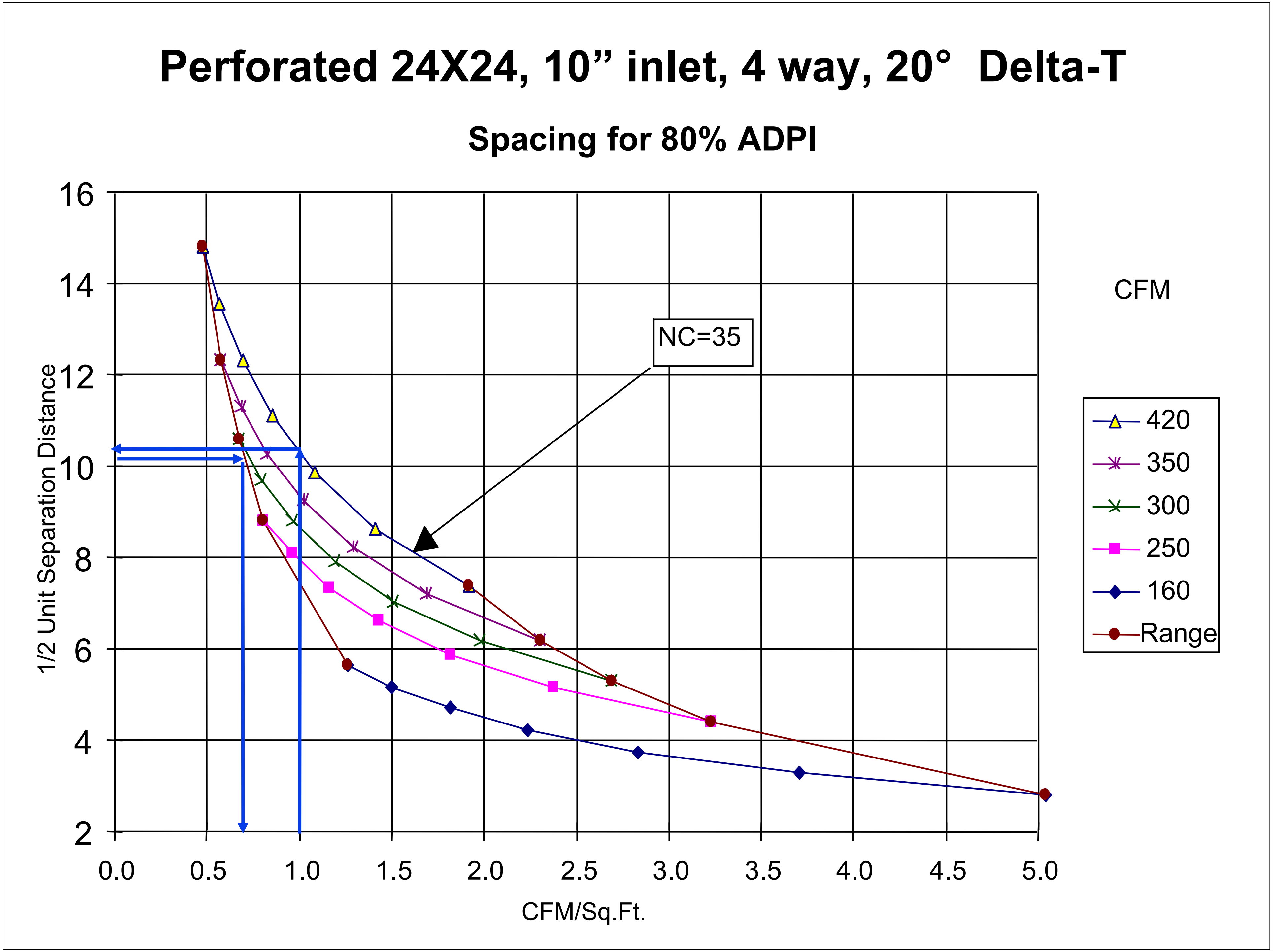
- ADPI is the percentage of points within the occupied zone having a range of calculated “effective draft temperatures” of -3° to +2° of average room temperature at a coincident air velocity less than 70 FPM.
- At 75F, and 30 fpm, 1 met, 0.9 clo, 0°F Draft temperature = 0 PMV (from ASHRAE Standard 55).
- 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.

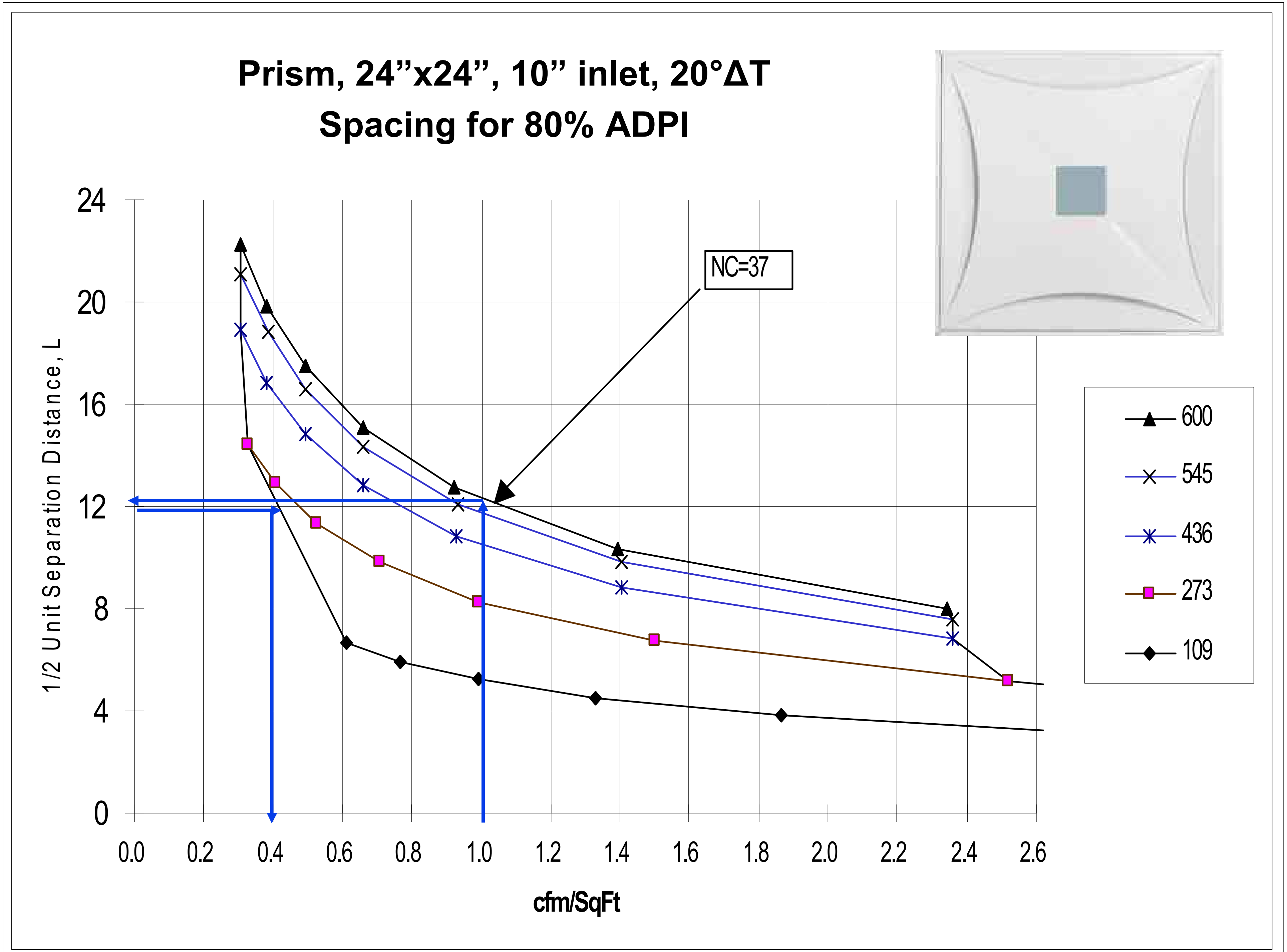
ASHRAE Fundamentals Chapter 20, Table 3

- 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. See Fundamentals Chapter 20 table 2 for more details on definition of L.
- A relationship was found between 50 FPM/min isothermal throw and cooling throw, and built into the selection charts included in ASHRAE Fundamentals, Chapter 20, table 3.
- Using this table engineers can assure clients that diffuser selections will provide acceptable mixing and air change effectiveness.

Air Diffusion Performance Index (ADPI) Selection Guide

Terminal Device	Room Load, Btu/h·ft²	T_{50}/L for Maximum ADPI	Maximum ADPI	For ADPI Greater than	Range of T_{50}/L
High sidewall grilles	80	1.8	68	—	—
	60	1.8	72	70	1.5–2.2
	40	1.6	78	70	1.2–2.3
	20	1.5	85	80	1.0–1.9
Circular ceiling diffusers	80	0.8	76	70	0.7–1.3
	60	0.8	83	80	0.7–1.2
	40	0.8	88	80	0.5–1.5
	20	0.8	93	90	0.7–1.3
Sill grille, straight vanes	80	1.7	61	60	1.5–1.7
	60	1.7	72	70	1.4–1.7
	40	1.3	86	80	1.2–1.8
	20	0.9	95	90	0.8–1.3
Sill grille, spread vanes	80	0.7	94	90	0.6–1.5
	60	0.7	94	80	0.6–1.7
	40	0.7	94	—	—
	20	0.7	94	—	—
Ceiling slot diffusers (for T_{100}/L)	80	0.3	85	80	0.3–0.7
	60	0.3	88	80	0.3–0.8
	40	0.3	91	80	0.3–1.1
	20	0.3	92	80	0.3–1.5
Light troffer diffusers	60	2.5	86	80	<3.8
	40	1.0	92	90	<3.0
	20	1.0	95	90	<4.5
Perforated, louvered ceiling diffusers	11–50	2.0	96	90	1.4–2.7
				80	1.0–3.4

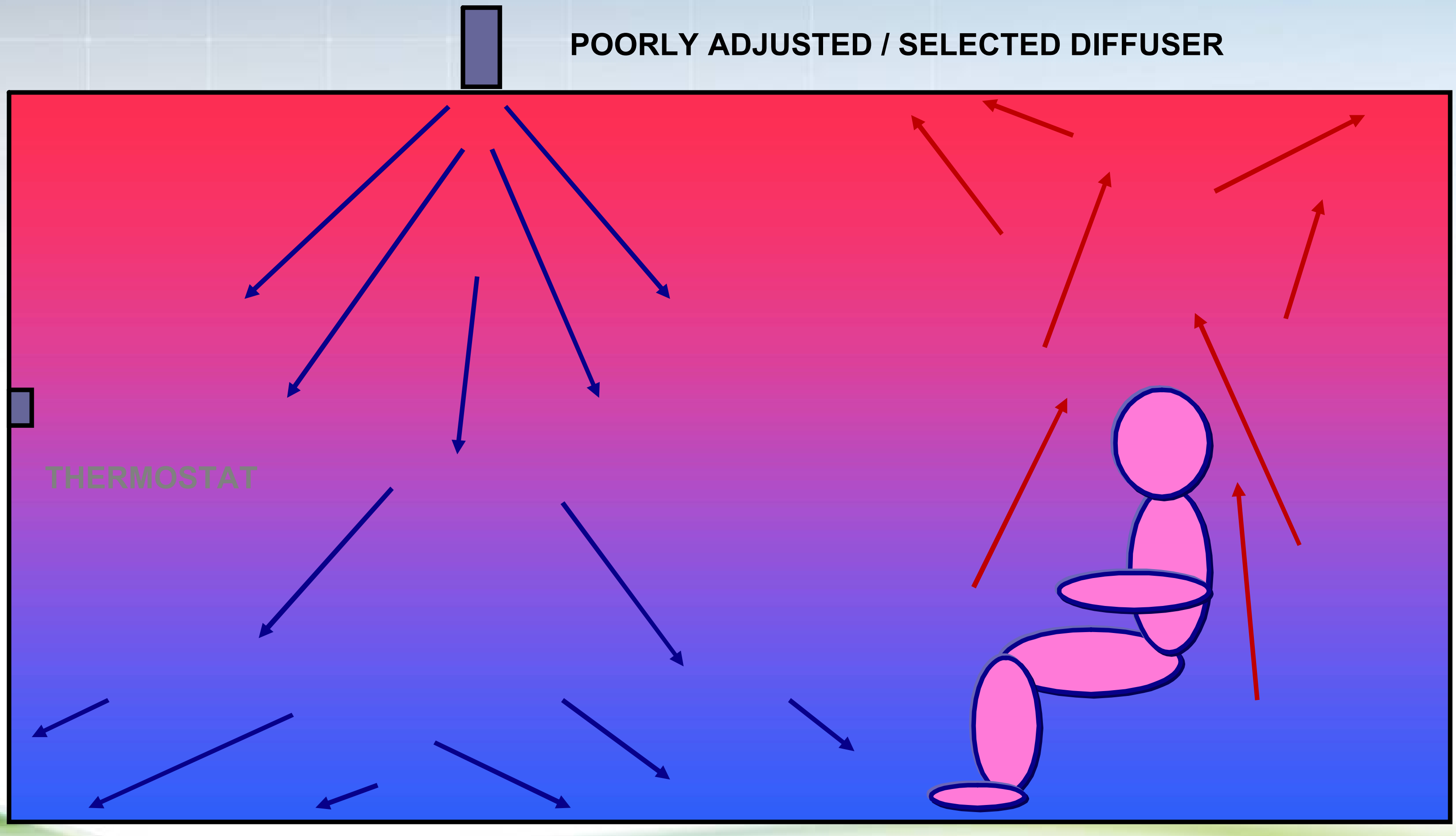

Perforated 24X24, 10” inlet, 4 way, 20° Delta-T
Spacing for 80% ADPI
NC=35
CFM
420
350
300
250
160
Range
1/2 Unit Separation Distance
CFM/Sq.Ft.
16
14
12
10
8
6
4
2
0.0
0.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0
4.5
5.0


Prism, 24”x24”, 10” inlet, 20°ΔT
Spacing for 80% ADPI
NC=37
1/2 Unit Separation Distance, L
cfm/SqFt
0
4
8
12
16
20
24
0.0
0.2
0.4
0.6
0.8
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
2.6
600
545
436
273
109

Linear diffusers

- Linears are an excellent perimeter choice – select two way blow with the diffuser located a couple feet from the window, with 2-way blow for best heating and cooling performance.
- Most linear diffusers MUST be adjusted. This requires the engineer to specify adjustments and that they be adjusted.
- Most linears, however, are not adjusted, and as a result, typically blow down, resulting in stratification, slow thermostat response, occupant complaints, and poor ventilation control.

Air Distribution, Poor Pattern



Designing for low interior loads

- The default minimum ventilation rate in offices is 17 cfm/person. With 150sf/person (per above example) that equals 0.22 cfm/sf
- A person generates about 350BTU of cooling demand (100w).
- At a supply air temperature low enough to provide latent heat removal (<55^{o}F), the minimum ventilation rate provides 370 BTUH of cooling.
- Not much more is required for today's laptops and flat panel monitors.
- 1 cfm/sf is probably 5X too much cooling!

Taking advantage of “free cooling”

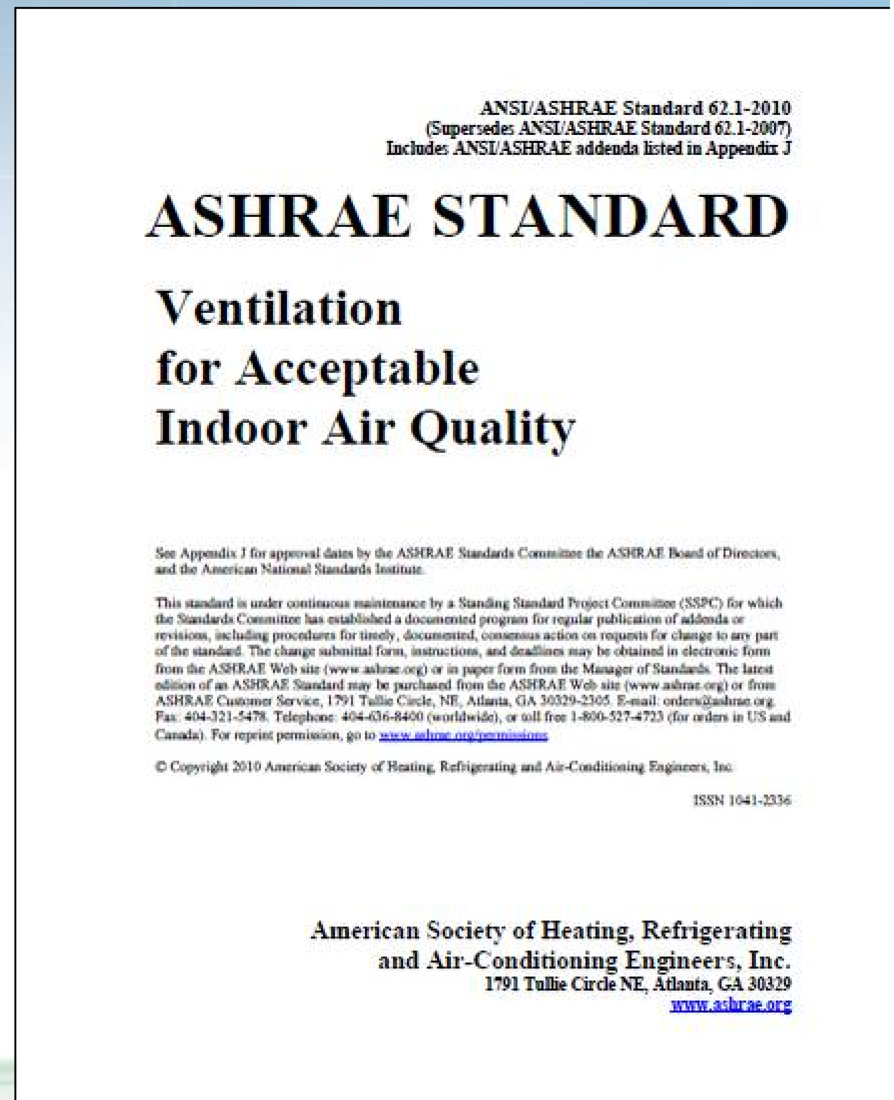
- Where is the highest cooling demand load found in most offices?
- At the perimeter, of course.
- Locate a return opening above the window. Warm air will pass into the plenum without any work by the HVAC system (hot air rises!)
- At low loads, most of the air in the plenum is discarded (to maintain building pressures) taking perimeter heat with it.

Making overhead heating work. (and meet code)

- Overhead heating is the most popular means of heating an office space.
- A number of technical papers were presented in the late 70's outlining the relationship between diffuser location, outlet temperature and perimeter heat loss.
- The results were included in the 1981 ASHRAE Fundamentals handbook.
- They were also included in ASHRAE Standard 62.1, table 6.2.

Indoor Air Quality

- Standing Standard Project Committee 62.1
- Residential Committee is 62.2
- Current Standard is 62.1-’10
- Several addenda for the ’10 version have already been approved



ANSI/ASHRAE Standard 62.1-2010
(Supersedes ANSI/ASHRAE Standard 62.1-2007)
Includes ANSI/ASHRAE addenda listed in Appendix J

ASHRAE STANDARD

Ventilation for Acceptable Indoor Air Quality

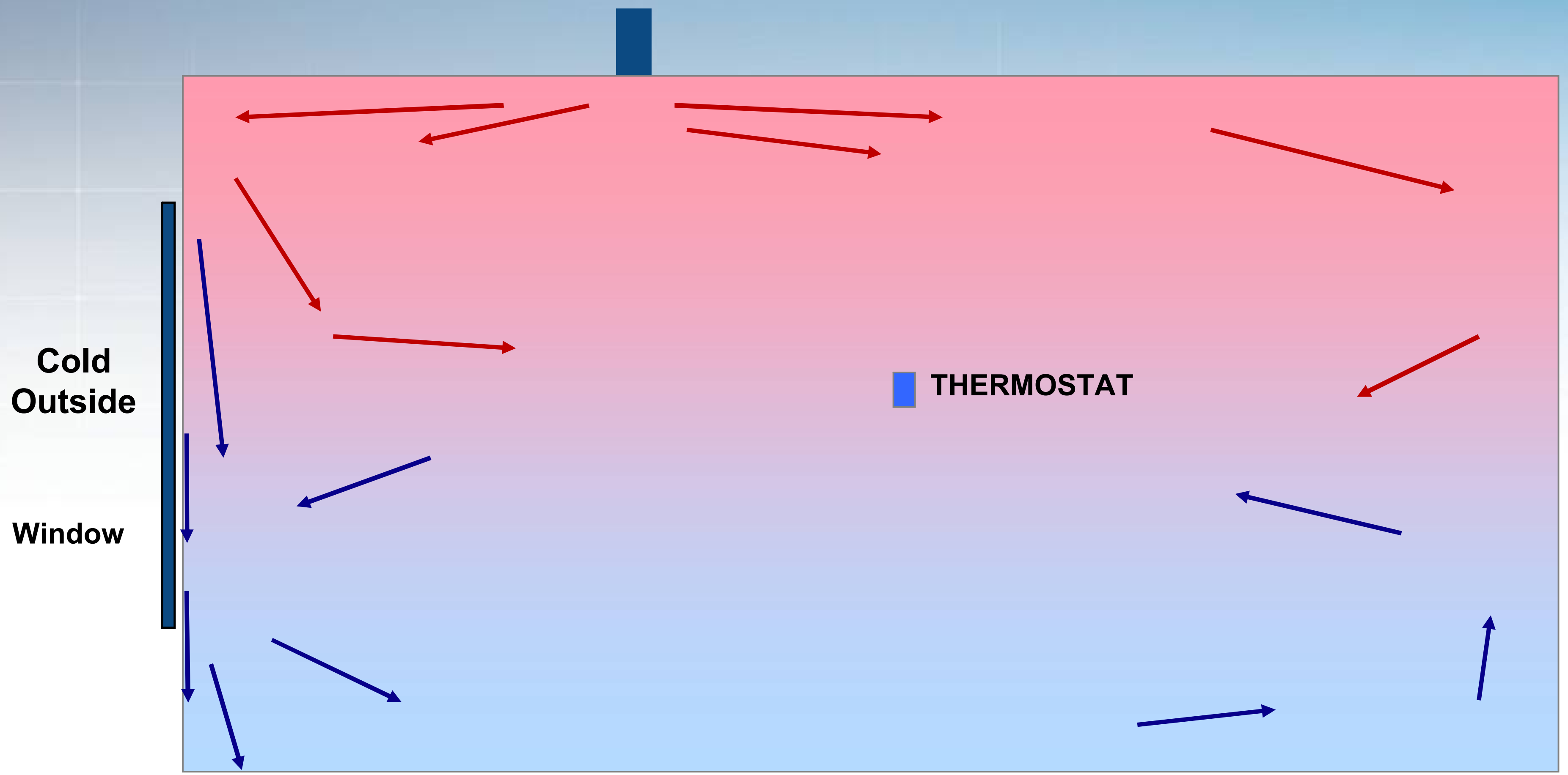
See Appendix J for approval dates by the ASHRAE Standards Committee the ASHRAE Board of Directors, and the American National Standards Institute.

This standard is under continuous maintenance by a Standing Standard Project Committee (SSPC) for which the Standards Committee has established a documented program for regular publication of addenda or revisions, including procedures for timely, documented, consensus action on requests for change to any part of the standard. The change submittal form, instructions, and deadlines may be obtained in electronic form from the ASHRAE Web site (www.ashrae.org) or in paper form from the Manager of Standards. The latest edition of an ASHRAE Standard may be purchased from the ASHRAE Web site (www.ashrae.org) or from ASHRAE Customer Service, 1791 Tullie Circle, NE, Atlanta, GA 30329-2305. E-mail: orders@ashrae.org. Fax: 404-321-5478. Telephone: 404-636-8400 (worldwide), or toll free 1-800-527-4723 (for orders in US and Canada). For reprint permission, go to www.ashrae.org/permissions.

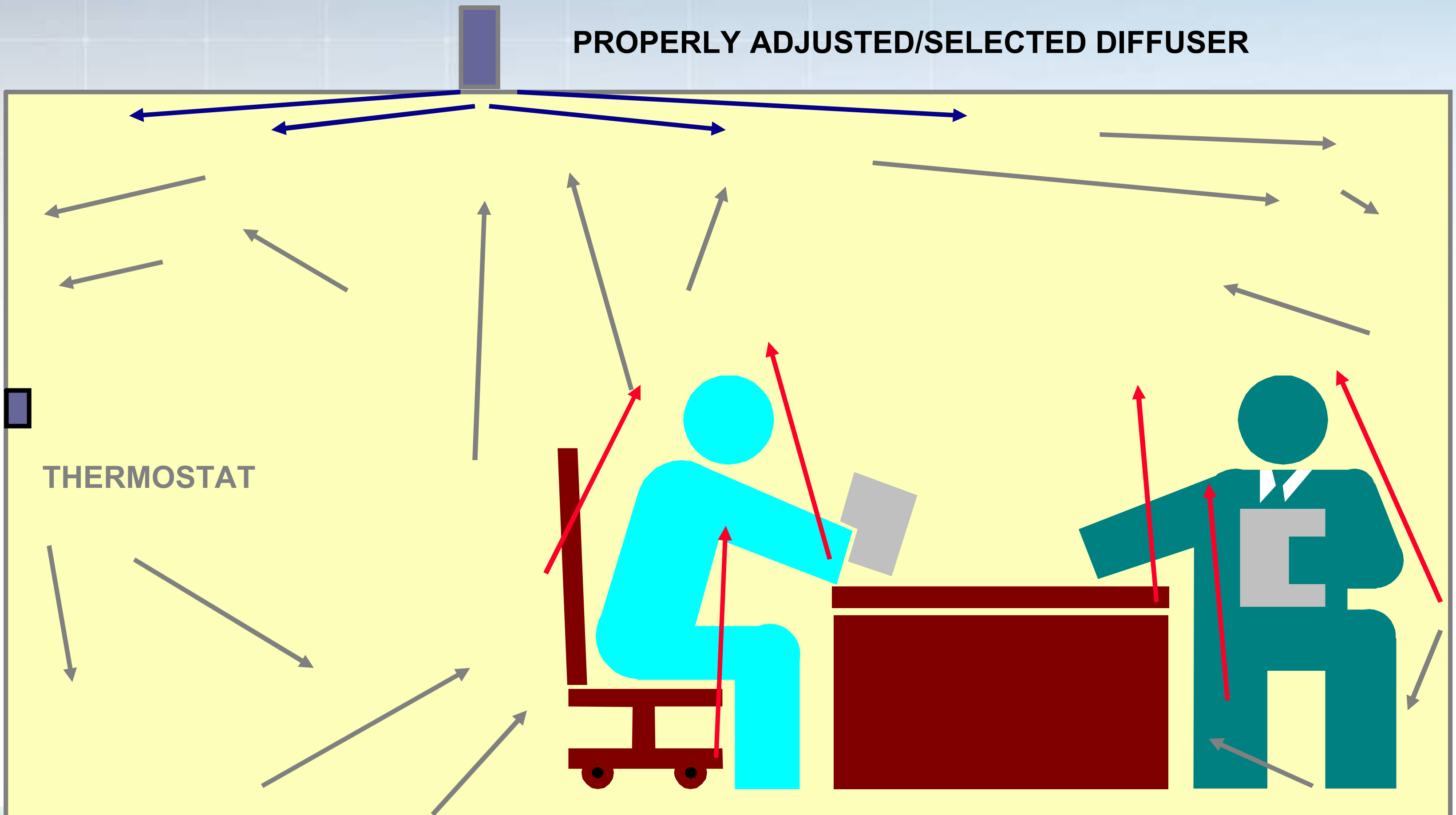
ISSN 1041-2336

American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.
1791 Tullie Circle NE, Atlanta, GA 30329
www.ashrae.org

Common Overhead Heating Design



Air Distribution, Good Pattern



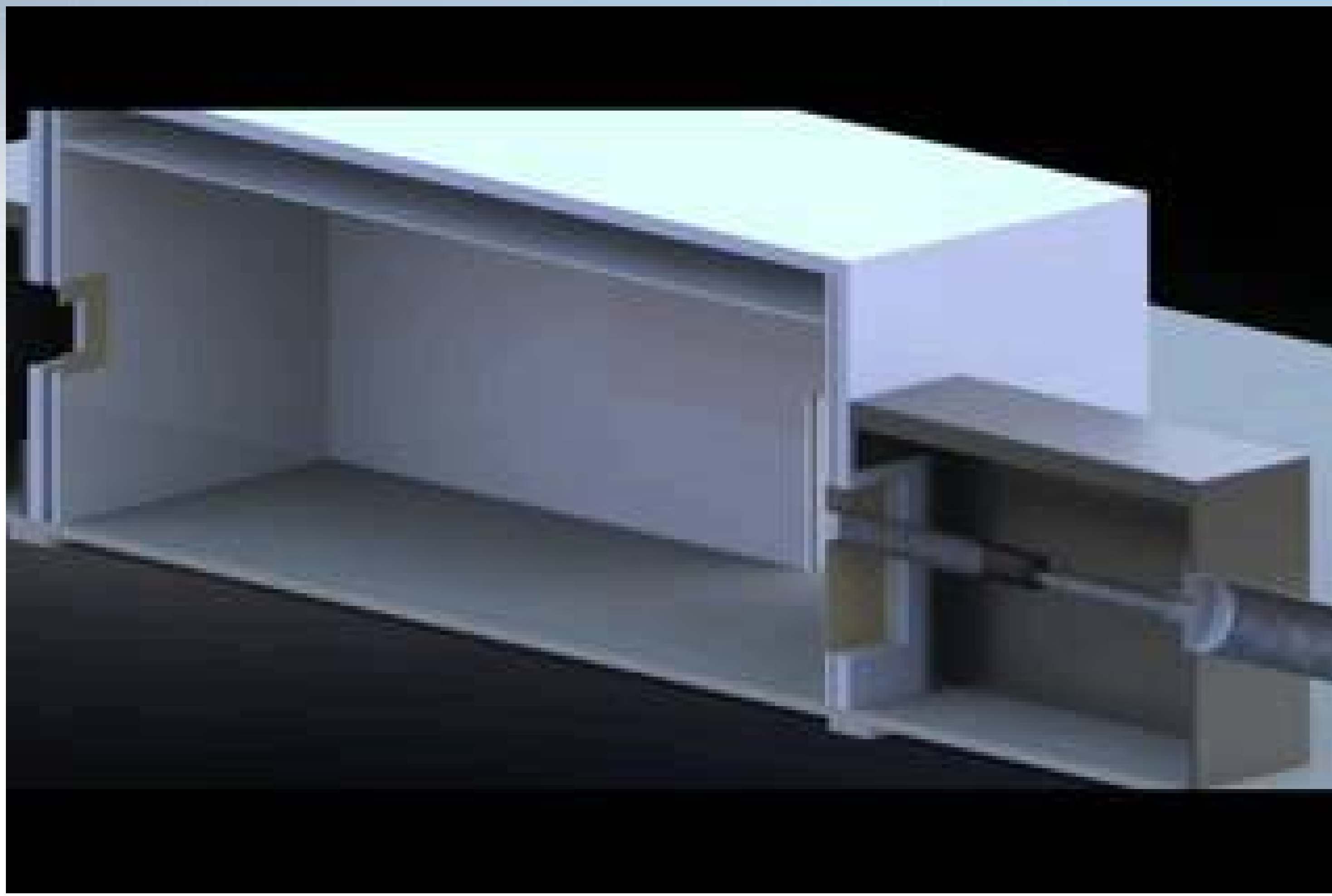
Overhead Heating Perimeter Considerations:

- **Maximum delta-t for effective mixing when heating from overhead, per ASHRAE handbook = ?.**
- **= 15°F (90°F discharge), continuous operation.**
- **Throw toward and away from glass.**
- **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.**
- **Typical perimeters require only 8°F Delta-t**
  @ 1cfm/sq.Ft.

Predicting and Controlling 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 2013 will include acoustical credits and requirements.
- A new ruling by AHRI will change everyone's discharge sound ratings considerably.

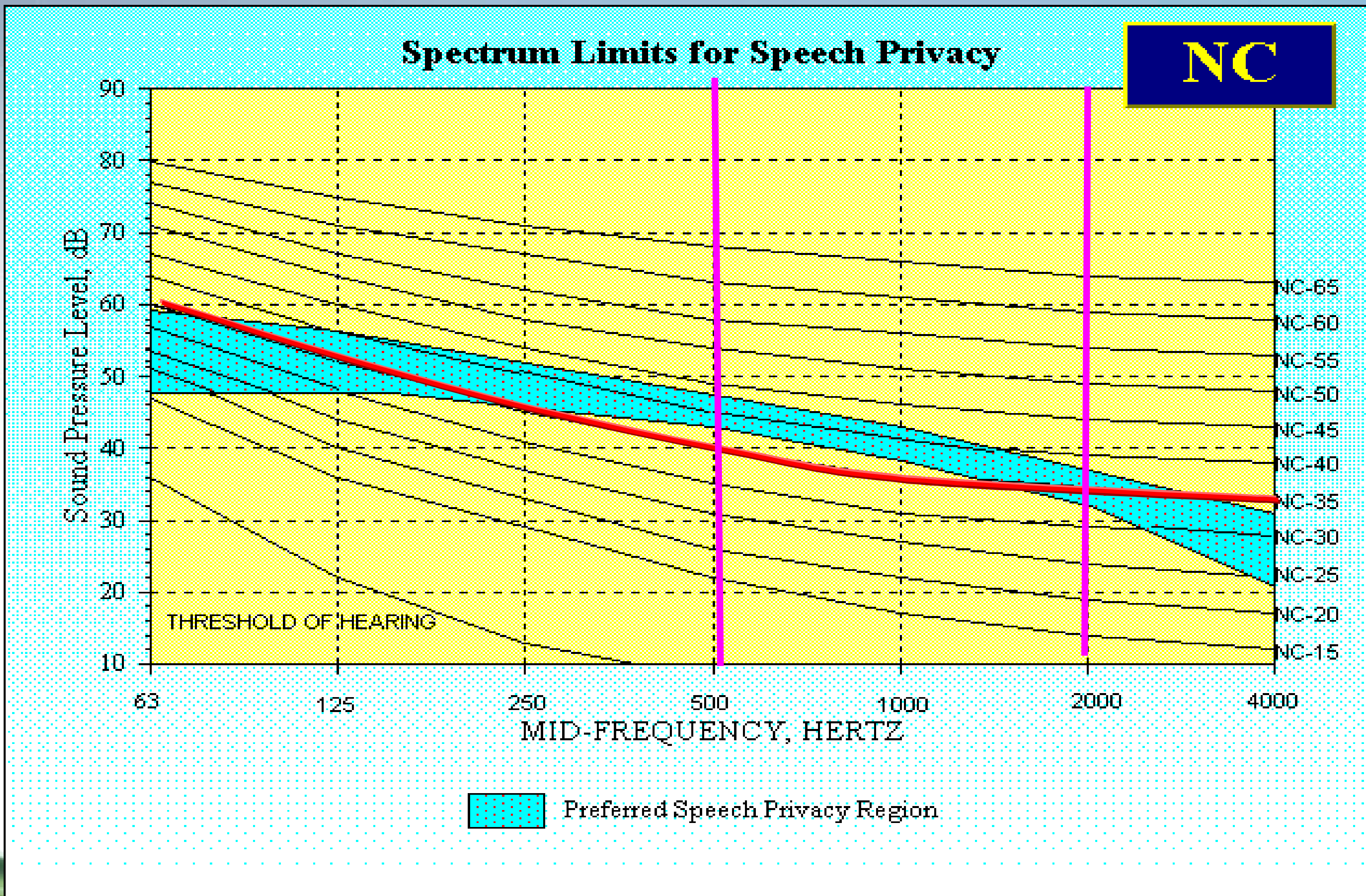
End Reflection



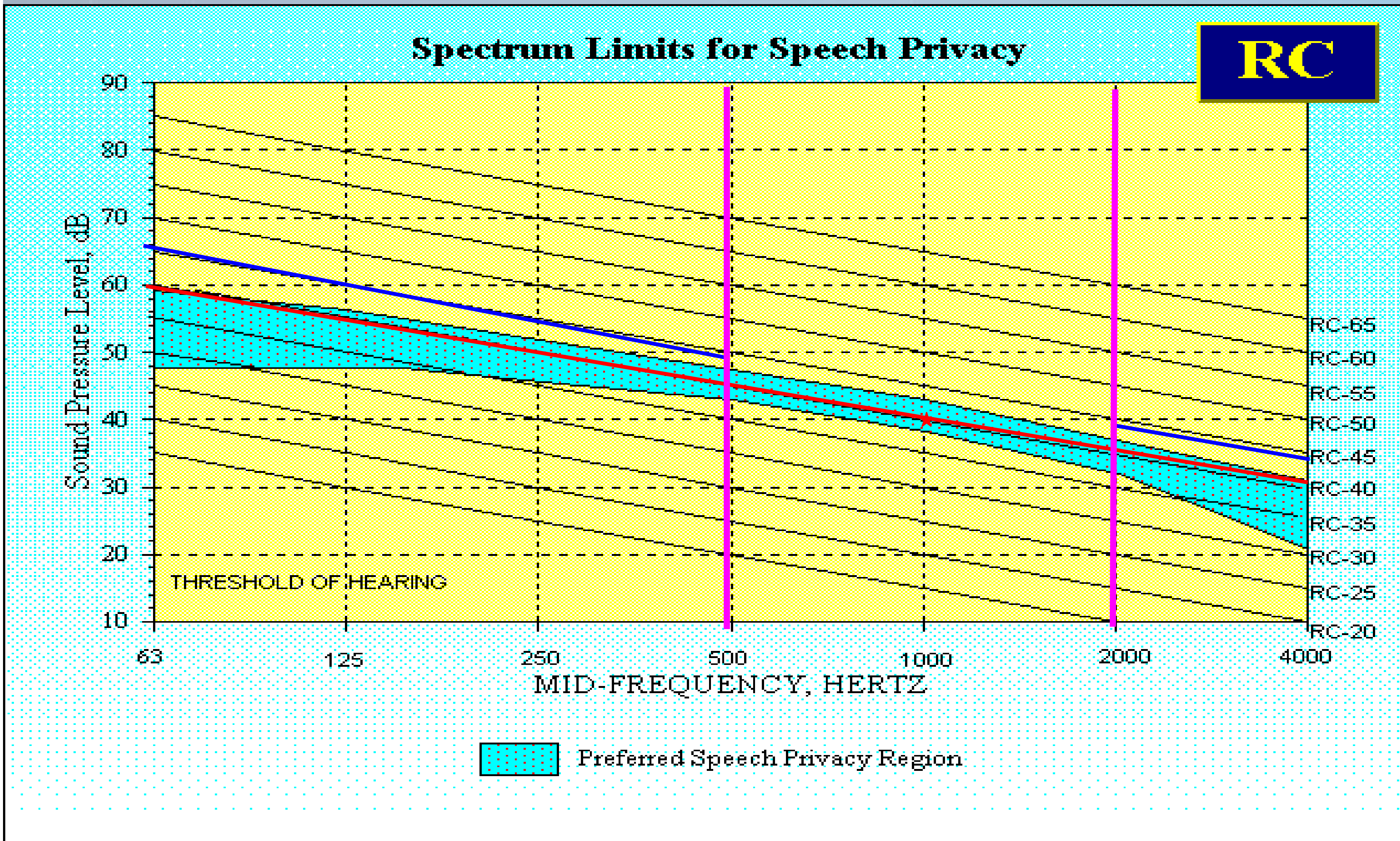
End Reflection

- 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.

NC & SPEECH PRIVACY



RC Speech Privacy



Sound Specifications

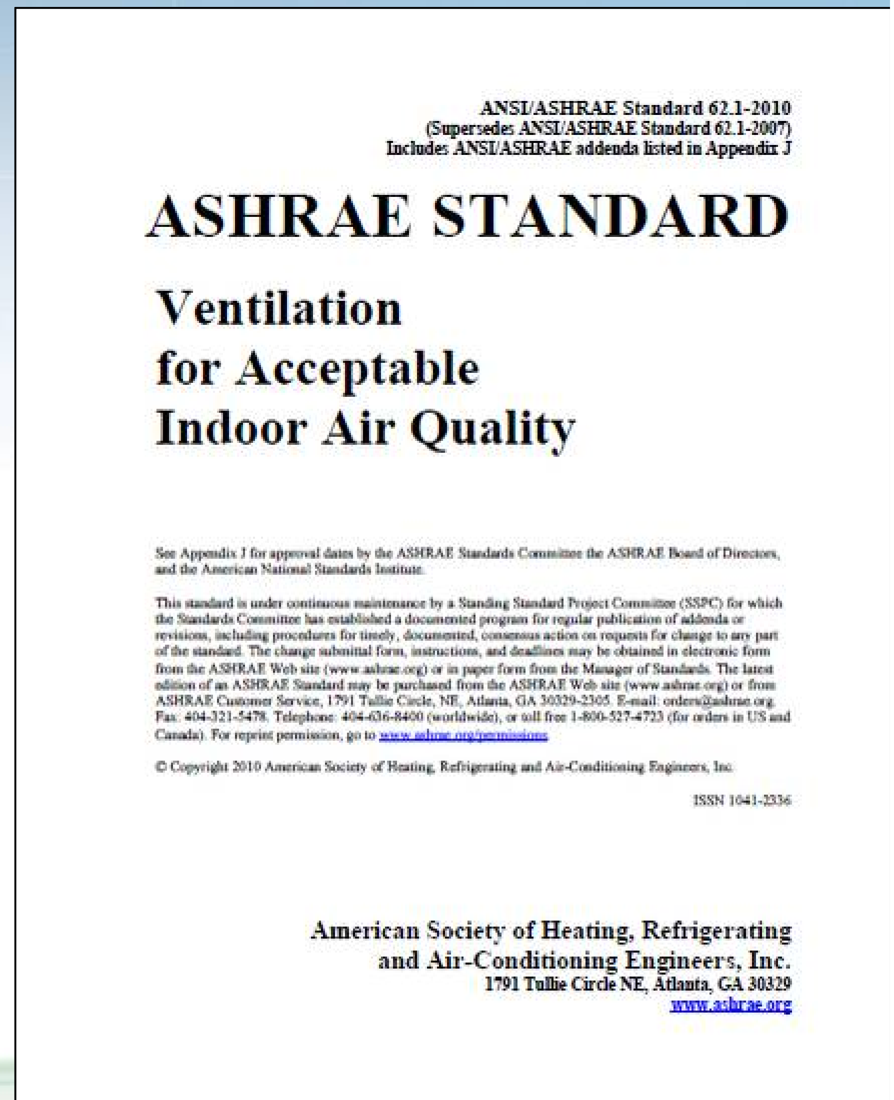
- Should be based on clearly stated assumptions.
- Should reflect real project needs, not any manufacturer's data and use currently accepted application factors.
- If duct lining is used – require"NC shall be determined in accordance with ARI 885-08, Appendix E"
- A meaningful specification will specify octave band.
- Over-silencing increases both initial costs and operating costs, and may hinder proper IAQ performance.
- Manufacturer's air terminal selection software and generic specification makers such as (SounSpec.EXE) are available online!

Acoustics:

- Acoustical quality suggests the use of RC (or newer measures) rather than NC.
- ANSI 12.60 requires 26NC (35dBA) in schools. (LEED currently requires 45 dBA)
- LEED 2012 for Schools will have 40dDBa as an HVAC noise prerequisite, using AHRI 885 calculations, with reverb time specified by construction.

Indoor Air Quality

- Standing Standard Project Committee 62.1
- Residential Committee is 62.2
- Current Standard is 62.1-’10
- Several addenda for the ’10 version have already been approved



ANSI/ASHRAE Standard 62.1-2010
(Supersedes ANSI/ASHRAE Standard 62.1-2007)
Includes ANSI/ASHRAE addenda listed in Appendix J

ASHRAE STANDARD

Ventilation
for Acceptable
Indoor Air Quality

See Appendix J for approval dates by the ASHRAE Standards Committee the ASHRAE Board of Directors, and the American National Standards Institute.

This standard is under continuous maintenance by a Standing Standard Project Committee (SSPC) for which the Standards Committee has established a documented program for regular publication of addenda or revisions, including procedures for timely, documented, consensus action on requests for change to any part of the standard. The change submittal form, instructions, and deadlines may be obtained in electronic form from the ASHRAE Web site (www.ashrae.org) or in paper form from the Manager of Standards. The latest edition of an ASHRAE Standard may be purchased from the ASHRAE Web site (www.ashrae.org) or from ASHRAE Customer Service, 1791 Tullie Circle, NE, Atlanta, GA 30329-2305. E-mail: orders@ashrae.org. Fax: 404-321-5478. Telephone: 404-636-8400 (worldwide), or toll free 1-800-527-4723 (for orders in US and Canada). For reprint permission, go to www.ashrae.org/permissions.

ISSN 1041-2336

American Society of Heating, Refrigerating
and Air-Conditioning Engineers, Inc.
1791 Tullie Circle NE, Atlanta, GA 30329
www.ashrae.org

Minimum Ventilation Rates

- Occupied Minimums are the sum, of occupant and floor area
- Unoccupied (but available) is just based on floor area
- Not Occupied means "off".
- So, we need to have three ventilation rates, not one.
- When one zone changes, all the others need to adjust .

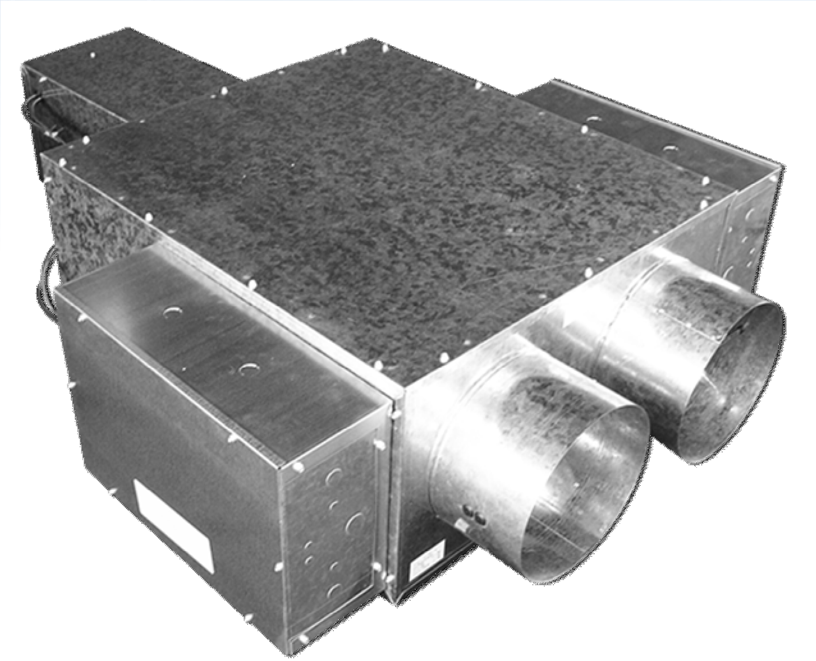
So, What Can We Do?

- The “Rules” are becoming code, or at least the “Acceptable Standard of Care”.
- Loads are dropping as Ventilation rates are increasing.
- Perimeter loads are becoming less.
- Ventilation rates are not constant.
- The solution starts at the input of outside air.

Dedicated Outdoor Air Systems

DOAS Dual Duct

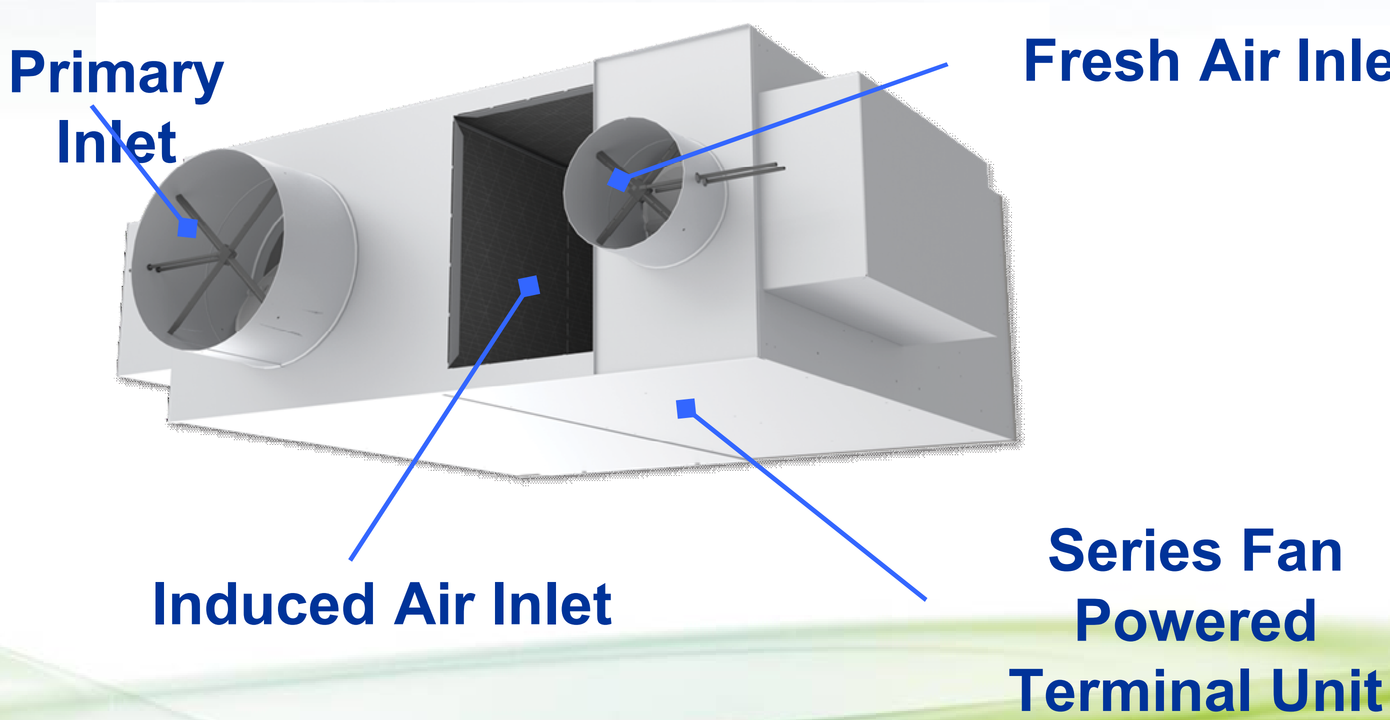
- One inlet provides 100% outside air, dehumidified, typically cold.
- Other duct provides 100% return air, either warm or cold, depending on the season.
- Supplemental reheat coils and even a sensible cooling coil have been considered
- A good mixing baffle should be employed (20:1 Mixing Ratio recommended)



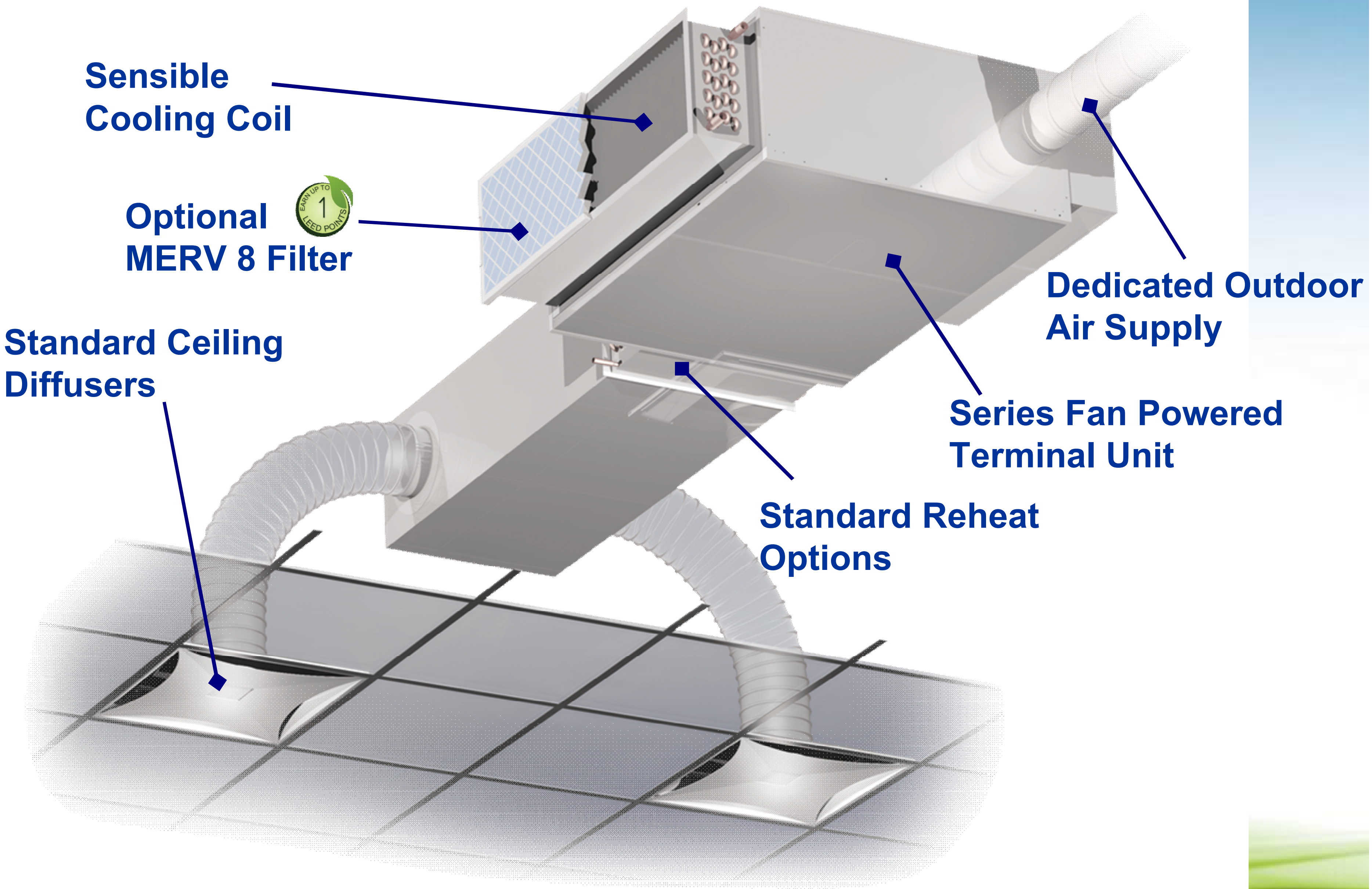
Mixing Dual Duct

Fresh Air Terminal Unit

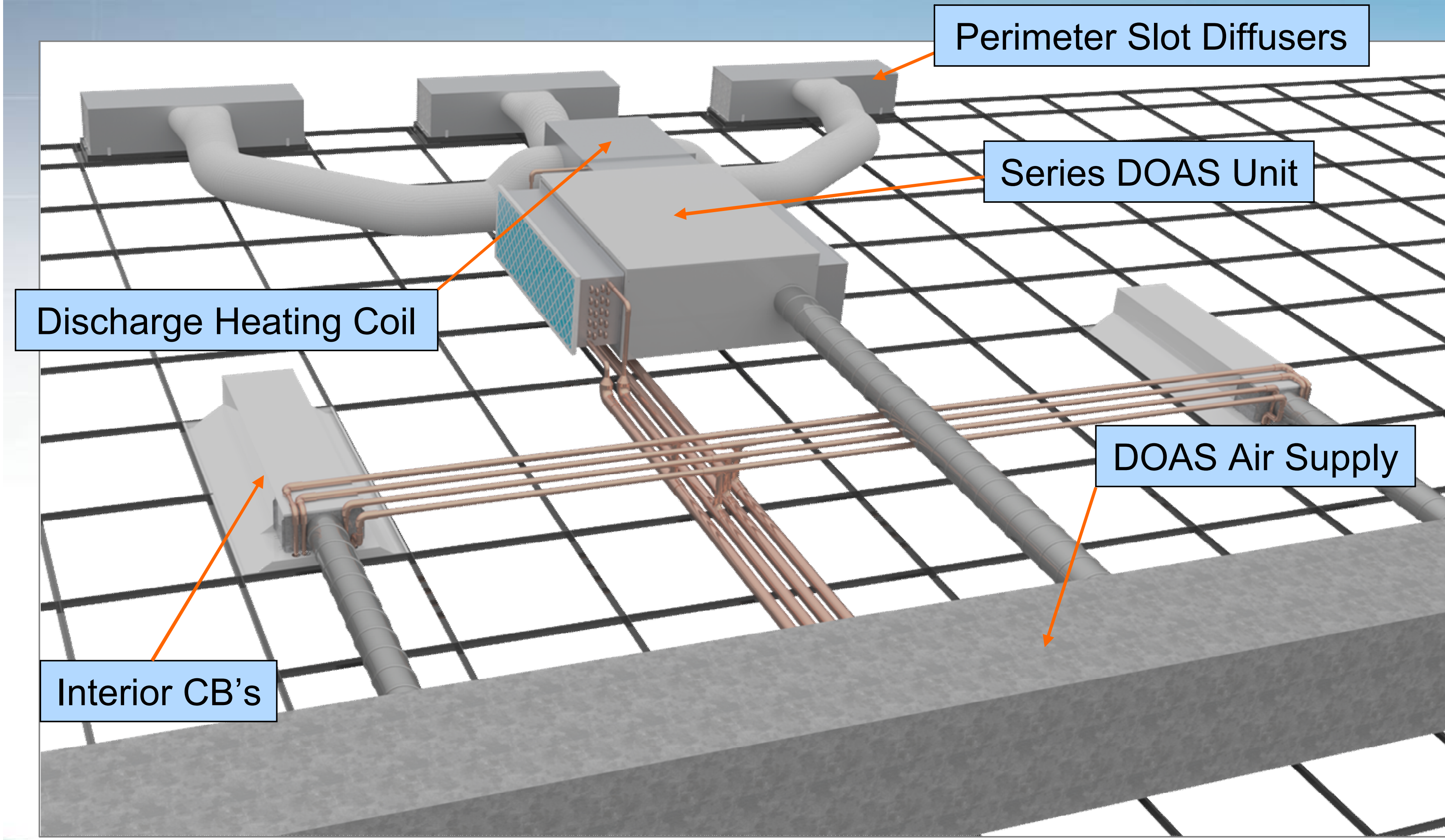
- Outside air can be supplied to a Series Fan Terminal through a second ducted system.
- This requires two duct systems, but separates ventilation and recirculated air.
- The system allows monitoring of ventilation rates into each zone.
- A stand-alone Pressure Independent controller on the DOAS inlet, tied to the fan relay, will allow standard DDC zone controls.



The Chilled Box



Series Chilled Box at the Perimeter

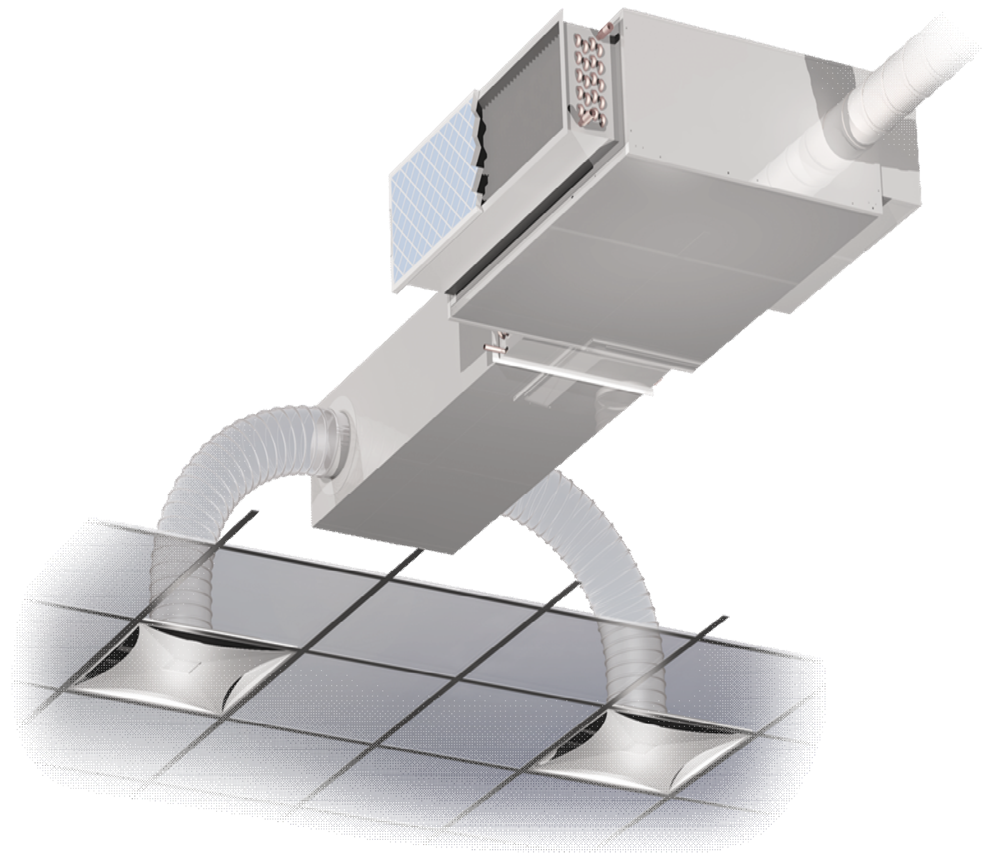


Control Strategies

- Take advantage of the ECM variable Volume feature, maintaining as low a total flow as necessary.
- DOAS Ventilation air should typically be cold and dry.
- Increase fan CFM to avoid sub cooling.
- Increase Primary (ventilation) air during periods of high perimeter cooling demand.

Chilled Box Summary

- Increased air distribution flexibility.
- Eliminated the central air handler, and uses a similar mechanical system as with Chilled Beams.
- Utilizing VAV fan control, space loads can be maintained at a low energy cost.
- Minimal Contractor Training
- Guaranteed Performance



So What have we learned?

- Spaces are often too cold because of stratification and poor humidity control
- Acoustics are poor (typically too quiet), but we can predict HVAC noise, and manage the acoustical environment.
- Energy bills are higher than predicted because designs don’t take advantage of the low loads and free cooling opportunities.
- Productivity can be increased, likely saving both time and money.

Summary

- LEED 2009 (V3) requires meeting Standard 62.1
- ADPI can assure compliance to 55 in the design phase, but only for overhead systems in cooling mode.
- Reheat needs to be carefully considered in terms of discharge temperatures and velocities.
- Acoustics should be specified as maximum allowed octave band sound power.
- DV, UFAD and Beams can supply quiet air in classrooms
- Software is available to assist in selecting the best mix of products.
- Energy Use Calculations need to be validated.
- The rules are dynamic - pay attention.

Net Zero Energy Building

Contact

KRUEGER

EXCELLENCE IN AIR DISTRIBUTION

dint-hout@krueger-hvac.com

www.krueger-hvac.com

Read my Air Distribution Blog!