Practical Diffuser Selection and Layout Procedure

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The selection and layout of diffusers to maximize occupant thermal comfort can be complicated and time consuming. Given the sheer quantity of diffusers that must be selected and laid out on every project, developing a streamlined and consistent process can help firms remain profitable in a competitive environment. This article describes a practical selection and layout procedure that provides occupant comfort, minimizes construction cost and can be applied to most spaces quickly and effectively.

Constraints and Assumptions

No single procedure could encapsulate the engineering design required to adequately distribute air in any space that an architect can envision. Edge cases and limitations will always exist. As such, we will work with the following constraints and assumptions:

1. The investigation will be limited to spaces where overhead air supply is possible. We won't include, for example, gymnasiums, which typically require a large space between the court or field and the nearest overhead obstruction.

2. The focus will be on spaces designed primarily for human comfort. We won't include, for example, surgery suites, where the primary intent of the air distribution system is to create an envelope of cold clean air around the staff and patient.

3. A consistent diffuser look will be maintained by assuming all diffusers in a space will be the same model. This will limit our coverage options, but consistency is key when designing the aesthetics of a space.

4. The assumption will be made that windows don't need "washing." Historically, linear slot diffusers have been used to blow warm air onto windows in the winter—when the delta T across the window is highest—in an effort to keep the interior window surface temperature to within 18°F (10°C) of the space temperature to satisfy the allowable radiant temperature asymmetry in Table 5.3.4.2 of ASHRAE Standard 55-2017.[1]

With the proliferation of the International Energy Code, washing is not as important because code minimum windows have high enough R-values that the internal surface temperature stays above the allowable temperature in the coldest regions of each climate zone. *Table 1* shows the calculated window inside

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surface temperature for a select station in each climate zone assuming a 72°F (22°C) space temperature, 0.17 h·ft^2·°F/Btu (0.03 K·m^2/W) exterior air film resistance and a 0.68 h·ft^2·°F/Btu (0.12 K·m2/W) interior air film resistance. Even if the project is a renovation with old single-pane windows, "washing" windows with warm air is also tricky with a variable air volume (VAV) system. You need the most throw when your VAV is at a minimum; instead consider fin tube heating under windows.

TABLE 1 Window inside surface temperature in each climate zone.

CLIMATE ZONE	2018 IECC INOPERABLE FENESTRATION MINIMUM R-VALUE (Btu/h·ft^2·°F)	STATION	99% DESIGN TEMPERATURE (°F)	INSIDE WINDOW SURFACE TEMPERATURE (°F)	WINDOW COOLER THAN INDOOR AIR (°F)
Zone 1	2.0	Miami, Fla.	49.6	66.7	5.3
Zone 2	2.0	Midland Intl, Texas	24.6	60.7	11.3
Zone 3	2.2	Vance AFB, Okla.	15.6	59.3	12.7
Zone 4	2.6	Kansas City Intl, Mo.	7.2	59.3	12.7
Zone 5	2.6	North Omaha, Neb.	-0.1	57.9	14.1
Zone 6	2.8	Sioux Falls, S.D.	-6.3	57.3	14.7
Zone 7	3.4	Minot AFB, N.D.	-17.2	57.9	14.1
Zone 8	3.4	Fairbanks, Alaska	-37.6	54.7	17.3

5. Diffusers will be selected and laid out based on the comfort criteria T_{50}/L method, where T_{50} is the diffuser throw length to a terminal velocity of 50 fpm (0.25 m/s) and L is the characteristic length of the space. This method is used to predict the level of comfort in a space by predicting its air distribution performance index (ADPI). A high ADPI indicates that a space will be well mixed and a high percentage of occupants will be comfortable. Each diffuser type will have an ideal range of T_{50}/L, provided by the diffuser manufacturer or by the *2015 ASHRAE Handbook—HVAC Applications*, Chapter 57.[2] Each space will have an actual T_{50}/L, based on the dimensions of the space and the T_{50} of the selected diffuser. When selecting and placing diffusers, we will attempt to drive the actual T_{50}/L as close as we can to the recommended T_{50}/L. However, this article will show that forcing the ideal and actual T_{50}/L to match is unlikely.

The Characteristic Length, L

The characteristic length is defined by the *2015 ASHRAE Handbook—HVAC Applications*, Chapter 57 for perforated and louvered ceiling diffusers as the "distance to wall or midplane between outlets."[2] It is a single number for the entire space, so an effort must be made to keep all diffusers in a space equidistant from walls and other outlets. To accomplish this, let's look at a few examples. First, a square 20 ft × 20 ft (6 m × 6 m) space with one diffuser as shown in *Figure 1a*. If the diffuser is placed directly in the center of the space, the space has a characteristic length of 10 ft (3 m). If the characteristic length needs to be decreased, the diffusers should be placed in a 2 × 2 grid as shown in *Figure 1b*.

With a non-square 20 ft × 40 ft (6 m × 12 m) rectangular space as shown in *Figure 2*, a single diffuser will not give a consistent characteristic length. The characteristic length in the direction of the 40 ft (12 m) wall is 20 ft (6 m), while the characteristic length in the direction of the 20 ft (6 m) wall is 10 ft (3 m). This deviation in the characteristic length has real-world implications.

FIGURE 1A (LEFT) AND 1B (RIGHT) left: L = 10 ft; right: L = 5 ft.

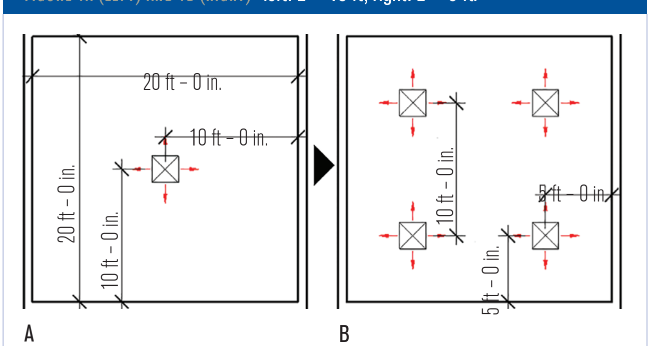


FIGURE 2 L = 10 ft and 20 ft.

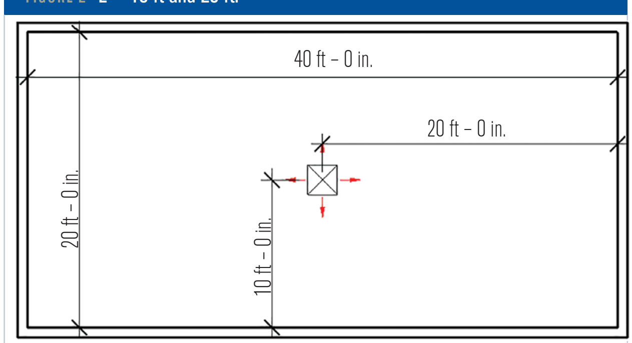


FIGURE 3 L = 10 ft.

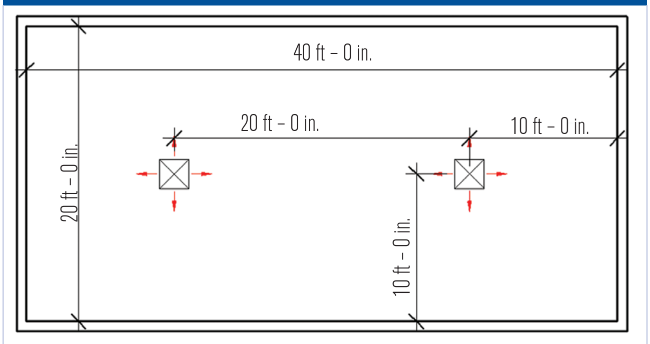


FIGURE 4 L = 5 ft.

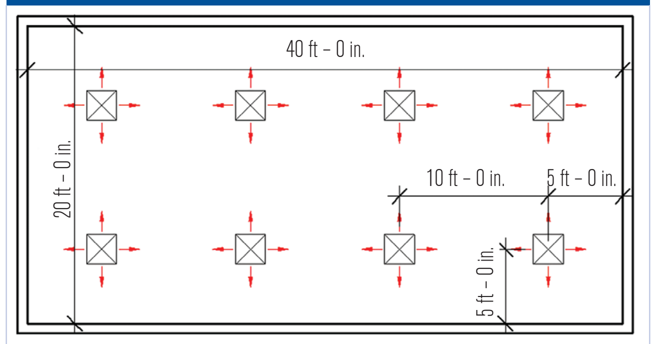
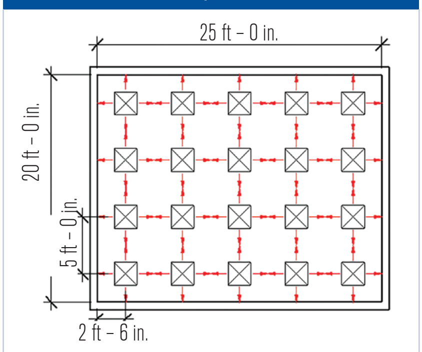


FIGURE 5 4 × 5 diffuser pattern.

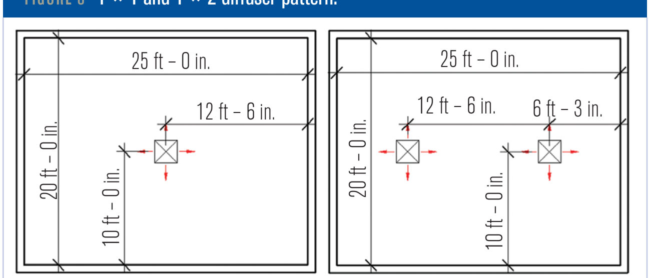


Depending on the throw of the selected diffuser, air may crash against the 40 ft (12 m) wall and leave occupants along the 20 ft (6 m) wall with stagnant air. If a 1 × 2 grid of diffusers is used (as shown in *Figure 3*), the characteristic length becomes a consistent 10 ft (3 m). If the characteristic length needs to be decreased, the grid must increase to 2 × 4, as depicted in *Figure 4*. A mathematical definition of this pattern would be to create a grid of $AT_W \times (AT_W \times S_R)$, where S_R is the aspect ratio of the space and AT_W is the grid count in the width (shorter) dimension, which can be increased in order to reduce the characteristic length.

In this 20 ft × 40 ft (6 m × 12 m) space (*Figure 4*), the number of diffusers increased from two diffusers to eight diffusers. This is a significant construction cost increase because in addition to the diffuser cost, additional flex duct, runouts, dampers and taps are required.

While spaces come in various aspect ratios, diffusers can only be placed in discrete grids, i.e., 1 × 2, 2 × 4. Therefore, we must accept that characteristic lengths may deviate. For instance, consider a 20 ft × 25 ft (6 m × 8 m) space (S_R = 1.25), which would require 20 diffusers in a 4 × 5 grid (diffuser aspect ratio, AT_R = 1.25) to achieve a consistent characteristic length of 2 ft, 6 in. (0.8 m). *Figure 5* demonstrates how impractical this is. In addition to the cost of the diffusers, the design does not allow space for other ceiling devices such as lights, sprinkler heads or occupancy sensors.

FIGURE 6 1 × 1 and 1 × 2 diffuser pattern.



A 1 × 1 or 1 × 2 grid is much more reasonable for this space at the cost of having the characteristic length of the space slightly deviate. The characteristic space will be 10 ft (3 m) vs. 12 ft, 6 in. (4 m) in the case of the 1 × 1 grid, and 6 ft, 3 in. (2 m) vs. 10 ft (3 m) in the case of a 1 × 2 grid (*Figure 6*).

In addition to the space shape causing inconsistencies in the characteristic length, ceiling grids rarely line up perfectly with the space. It is also not uncommon for the ideal diffuser location to be taken by a higher priority ceiling element, such as a light. This causes the diffuser to shift even further, causing additional deviations to the characteristic length.

Because the characteristic lengths will vary slightly from diffuser to diffuser regardless, the constraint of keeping a perfectly consistent characteristic length can be relaxed and the $AT_W \times (AT_W \times S_R)$ layout grid can be improved by incrementally adding diffuser counts in the longer dimension using the following pseudocode algorithm:

```
S_R = S_L / S_W
AT_W = 1
AT_L = AT_W × [S_R]
AT_R = AT_L / AT_W
Loop:
        If |AT_R – S_R| 0.5
                AT_L = AT_L + 1
                AT_R = AT_L / AT_W
        Else If |AT_R – S_R| > 0.5
```

$$AT_W = AT_W + 1$$
$$AT_L = AT_W \times [R_S]$$
$$AT_R = AT_L / AT_W$$

Until: desired diffuser count, $AT_W \times AT_L$ is reached
where S_R is the space aspect ratio, calculated by dividing the space length, S_L, by the space width, S_W.

In this example it is assumed that S_L is the larger dimension. AT_R is the diffuser aspect ratio, calculated by dividing the diffuser count in the larger dimension (AT_L, by AT_W), which is the diffuser count along the shorter dimension. To ensure the diffuser grid follows the space shape as closely as possible, initially and every time we add a diffuser in the shorter dimension, we multiply the grid count in the shorter dimension (AT_W) by the rounded space aspect ratio, $[S_R]$.

When the diffuser aspect ratio diverges too significantly from the space aspect ratio, as indicated by the absolute value of the difference of the two aspect ratios being greater than 0.5 ($|AT_R - S_R| > 0.5$), then a diffuser in the shorter dimension must be added and the grid reset to match the space aspect ratio as closely as possible.

This algorithm allows the user to add diffusers in the smallest increment possible while maintaining the tightest characteristic length. In the case of our 20 ft × 40 ft (6 m × 12 m) space, the diffuser counts would progress from 2 to 3 to 4 to 4 to 6 to 8 as shown in *Figure 7*.

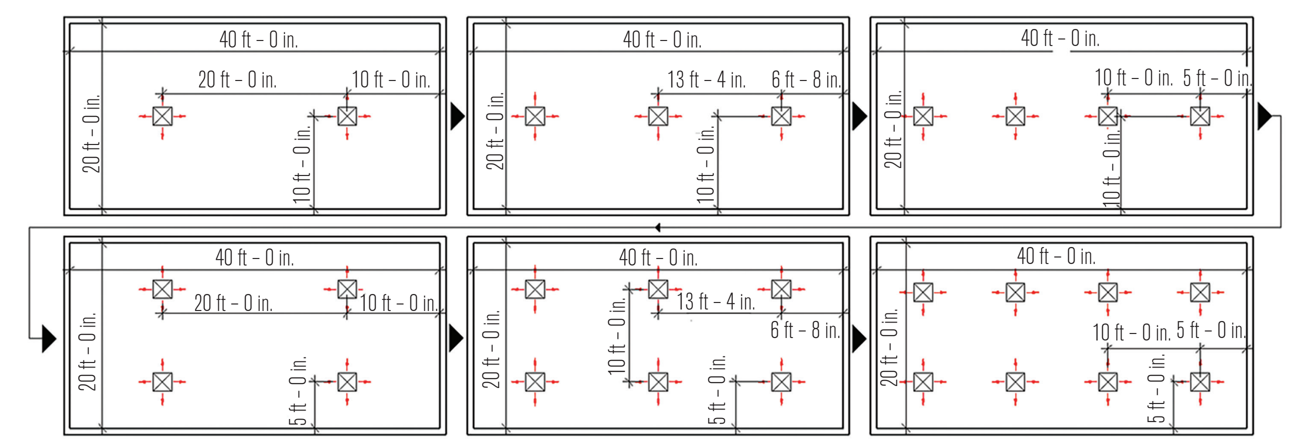
Length to 50 fpm Terminal Velocity, T_{50}

As previously mentioned, T_{50} is the diffuser throw length to a terminal velocity of 50 fpm (0.25 m/s). These values can be determined for a given diffuser size and throw pattern at a given cfm by reviewing the catalog data. An example for a louvered face diffuser is given in *Figure 8*.[3]

Regardless of what the ideal T_{50}/L is, each diffuser is going to ideally "cover" some area for a given cfm based on the T_{50} throw of the diffuser. If the ideal T_{50}/L is 1, then the area covered will be ($T_{50} \times 2/1$).[2] If the desired T_{50}/L is 2, then the area covered is ($T_{50} \times 2 / 2$).[2] For an example from *Figure 8*, an 8 in. (203 mm) four-way louvered face diffuser with 314 cfm (148 L/s) of supply air

Advertisement formerly in this space.

FIGURE 7 Air terminal layout pattern progression for the first six loops.



would have a T_{50} throw of 14 ft (4 m) in all four directions, and with an ideal T_{50}/L of 1, would cover a 28 ft × 28 ft (9 m × 9 m) area, or 784 ft^2 (73 m^2).

These areas are calculated for 6 in. (152 mm) and 8 in. (203 mm) in *Table 2*. If the coverage areas vs. the cfm across all neck sizes are plotted as shown in *Figure 9*, it is clear that the area "covered" by each diffuser is linear with the cfm applied. This holds true for many popular diffuser types, including radial plaque and perforated.

Because the coverage area decreases linearly with the cfm applied (even across multiple neck sizes), coverage area cannot be increased simply by adding more diffusers. If the diffuser count is increased from one to four, the cfm available to assign each diffuser reduces by ¼, and the coverage area remains the same.

For example, assume 314 cfm (148 L/s) is being delivered to a 20 ft × 20 ft space (6 m × 6 m) or 0.785 cfm/ft^2 (3.99 L/s·m^2), and a louvered face diffuser from *Figure 8* with an ideal T_{50}/L of 2 is used. If one 8 in. (203 mm) neck diffuser is placed in the center of the space, the characteristic length would be 10 ft (3 m), and the actual T_{50} would be 14. So, the actual T_{50}/L is 1.4 (14/10), which is only 70% (1.4/2) of the ideal 2. If the characteristic length is reduced to 5 ft (2 m) placing four 6 in. (152 mm) neck diffusers, each diffuser would now only have 78.5 cfm (37 L/s) and a 7 ft (2 m) throw. In this case, the actual T_{50}/L is still 1.4 (7/5), which is still 70% of the ideal 2, but now with more construction cost. Because the coverage area decreases linearly with the cfm applied (even across multiple neck sizes), this is true in all situations.

This demonstrates that with the given constraints, the actual T_{50}/L is decided solely by the diffuser type and the airflow density (cfm/ft^2) of the space. With popular diffuser types and standard ranges of airflow densities at maximum flow, ideal T_{50}/L ranges are

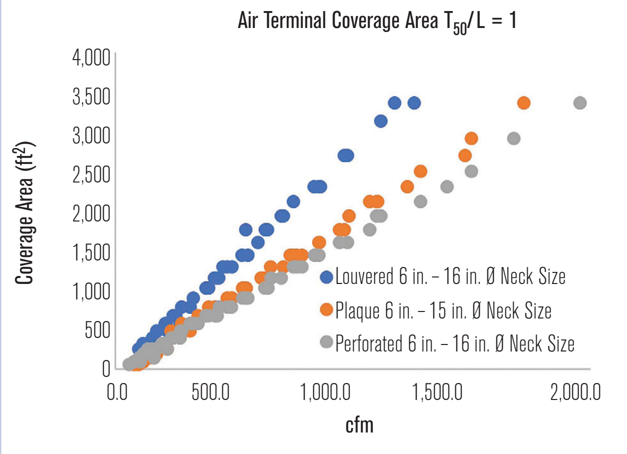
FIGURE 8 Typical diffuser catalog data.

Return Factors -SP = 1.1 TP NC + 1		Total cfm Total Pressure NC	98 0.041 -		117 0.058 14		137 0.079 18		156 0.103 21		176 0.131 25		215 0.196 30		254 0.273 34	
		Side	cfm	Throw	cfm	Throw	cfm	Throw	cfm	Throw	cfm	Throw	cfm	Throw	cfm	Throw
18	S1	X	98	6-9-18	117	7-11-20	137	9-15-22	156	10-15-23	176	11-17-25	215	20-24-34	254	17-21-30
x	S2&G2	X & Y	49	4-6-13	59	5-7-14	69	6-9-16	78	7-10-17	88	7-11-18	108	9-14-20	127	11-15-21
18	A3	X	37	3-5-9	44	4-6-10	51	5-7-10	59	6-8-11	66	6-8-12	81	7-9-13	95	8-10-14
6"		Y	25	3-5-8	29	4-6-9	34	4-7-9	39	5-7-10	44	6-7-11	54	7-8-12	64	7-9-13
Round	A4	X & Y	25	3-5-8	29	4-6-9	34	4-7-9	39	5-7-10	44	6-7-11	54	7-8-12	64	7-9-13
Return Factors -SP = 1.1 TP NC + 1		Total cfm Total Pressure NC	174 0.041 11		209 0.059 16		244 0.080 20		279 0.104 24		314 0.132 27		383 0.197 32		453 0.275 36	
		Side	cfm	Throw	cfm	Throw	cfm	Throw	cfm	Throw	cfm	Throw	cfm	Throw	cfm	Throw
18	S1	X	174	8-13-24	209	10-15-27	244	12-18-29	279	14-21-31	314	15-23-33	383	19-26-37	453	22-28-40
x	S2&G2	X & Y	87	5-8-17	105	7-10-19	122	8-12-24	140	9-14-22	157	10-15-24	192	12-19-26	227	15-20-29
18	A3	X	66	5-7-12	79	6-9-13	92	7-10-14	105	8-11-15	119	9-11-16	145	10-12-18	171	11-14-19
8"		Y	44	4-7-11	52	5-8-12	61	6-9-13	70	7-9-14	79	8-10-14	96	9-11-16	113	10-12-18
Round	A4	X & Y	44	4-7-11	52	5-8-12	61	6-9-13	70	7-9-14	79	8-10-14	96	9-11-16	113	10-12-18

TABLE 2 Louvered face data.

NECK DIAMETER	AIRFLOW (cfm)	A4 T_{50} (ft)	COVERAGE AREA $T_{50}/L = 1$ (ft^2)
6 in.	98.0	8	256
	117.0	9	324
	137.0	9	324
	156.0	10	400
	176.0	11	484
	215.0	12	576
	254.0	13	676
8 in.	174.0	11	484
	209.0	12	576
	244.0	13	676
	279.0	14	784
	314.0	14	784
	383.0	16	1,024
	453.0	18	1,296

FIGURE 9 Diffuser coverage area for popular models.



typically met. It's important to note that while catalog data is typically presented for isothermal conditions (i.e., supply air temperature equals room temperature), non-isothermal correction factors will only increase or decrease the slope of the coverage area vs. cfm lines, so the range of acceptable airflow densities will change in heating and cooling mode, but the conclusion is the same: adding more diffusers does not lead to better coverage.

As previously mentioned, with popular diffuser types and standard ranges of airflow densities at maximum flow, ideal T_{50}/L ranges are typically met. However, at VAV minimum flow, the actual T_{50}/L will typically be far below the ideal T_{50}/L range. This is the situation that most engineers refer to as "dumping," where there is not enough airflow coming out of the diffuser to cause adequate mixing and the air "dumps" onto the occupants, causing discomfort. While it may seem like "dumping" is a diffuser layout problem, it is a VAV minimum setpoint problem. Again, adding more diffusers does not alleviate the "dumping" problem because, for example, it is unclear if it is better to "dump" 40 cfm (19 L/s) out of one 8 in. (203 mm) diffuser or 10 cfm (4.7 L/s) out of four 6 in. (152 mm) diffusers.

To help the reader accept the fact that the ideal T_{50}/L will not be met at VAV minimum, we ask you to consider a 2014 ASHRAE Research Project entitled "Thermal and Air Quality Acceptability in Buildings that Reduce Energy by Reducing Minimum Airflow from Overhead Diffusers," which showed that thermal comfort significantly improved when VAV minimums were reduced from 30% to 10% of maximum flow.[4] Therefore, the lower bound of the ideal T_{50}/L ranges probably need to be reassessed.

Complex Spaces

Complex spaces are a conglomerate of simple spaces and can be addressed by being broken into simple rectangular subspaces that are treated separately.

Because architects are concerned about sight lines, complex spaces should be decomposed in a manner that leaves the largest area intact. For instance, the L-shape space in *Figure 10* could be decomposed with a vertical line shown in the left image, or a horizontal line shown in the right image. The right image would have the largest consistent diffuser grid pattern and the most aesthetically pleasing sight lines, so it is the logical choice.

As shown in the previous section, coverage area decreases linearly with cfm; if more cfm is assigned to one subspace in an attempt to get better coverage for that subspace, the other subspaces will be under covered. Therefore, cfm should be simply assigned to each subspace in proportion to its area.

There are some layouts in which uneven cfm distribution would be desired, e.g., a lab space with many hot plates in a nook or a space with uninsulated brick and single-pane glass walls.

Procedure

Now that the individual components of the procedure

FIGURE 10 Two ways of decomposing a space.

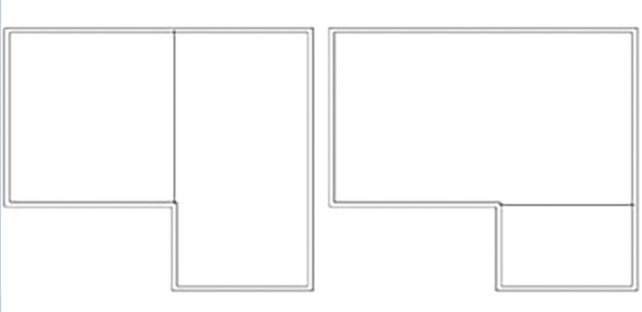


FIGURE 11 An example table of the maximum allowable cfm.

Supply Selections

Square Plaque Diffuser (SPD)

Air Terminal Neck Size	Maximum CFM	Max Noise Criteria (NC)	Max Total Pressure (in W.C.)	Max Throw at 50 fpm (ft)
06" Ø	210	25	0.074	8
08" Ø	335	25	0.106	10
10" Ø	475	25	0.136	11
12" Ø	630	25	0.166	13

have been addressed, we can formulate a streamlined and consistent process:

1. Select a diffuser manufacturer and model based on performance, material, aesthetics, cost, availability and local support.

2. Determine an acceptable noise criteria (NC) rating for the space and use the catalog to build a table of maximum allowable cfm for each neck size at the acceptable NC, such as the example in *Figure 11*.

3. Break complex spaces into rectangular subspaces leaving the largest continuous area intact.

4. Assign subspace airflow, cfm_{SS}, to each subspace in proportion to the subspace's area, SS_A, to the total space area, S_A.

5. Decrease the characteristic length by increasing the number of diffusers, following the procedure in the Characteristic Length section of this article.

6. Stop when the cfm/diffuser is below the allowable cfm of the largest neck size, cfm_{Max}. Assign the smallest

neck size where the maximum cfm for that neck size is above the cfm/diffuser.

As pseudocode:

For Each Subspace SS, in Space S

$cfm_{SS} = cfm_S \times SS_A / S_A$

$S_R = S_L / S_W$

$AT_W = 1$

$AT_L = AT_W \times [S_R]$

$AT_R = AT_L / AT_W$

Loop:

If $cfm_{SS} / (AT_W \times AT_L) < cfm_{Max}$

Break

If $|R_{AT} – R_s| \leq 0.5$

$AT_L = AT_L + 1$

Else If $|AT_R - S_R| > 0.5$

$AT_W = AT_W + 1$

$AT_L = AT_W \times [R_S]$

$AT_R = AT_L / AT_W$

End

The algorithm works well for many spaces; however, the isovel method of diffuser layout cannot be totally disregarded. In some cases, ceiling spaces can be very cluttered due to other ceiling devices such as lighting, smoke detectors and sprinkler heads, which may take precedence, so the diffusers get pushed from their ideal location closer to the perimeter or an adjacent diffuser. This may require users to map the isovels to ensure air isn't crashing on a wall or adjacent airstream at too high a velocity, which could cause uncomfortable drafts in the occupied zone.

These rules don't work well for long thin spaces, such as corridors, which may have aspect ratios of 10+. In this scenario, placing 10 diffusers down a hallway may not be worth it; two-way throw patterns should be installed instead.

Additional Diffuser Suggestions

1. Return diffusers should be placed diagonally to the supply grid to limit "short circuiting," where the supply air is blown directly into a return. The same type of diffusers can be used for returns as well as supplies to ensure a consistent look and cadence of the ceiling.

2. Round neck diffusers with flexible duct connections should be used, as this greatly increases constructability because it allows slight adjustments to the diffuser location as the ceiling grid shifts from plan. The flexible duct also absorbs noise coming down the duct from fans and dampers. This also applies to return diffusers, even if they aren't ducted. Return diffusers are frequent sources of "cross-talk" between private spaces like offices and adjacent public spaces. If too much noise is transferring back and forth, adding a short length of flexible duct to the returns is a quick and inexpensive way to mitigate the problem. Occasionally, there are also distracting visual elements in plenums, such as blinking lights from lighting controllers or glare from sheet metal, that can be hidden with a short length of flex duct attached to the round neck. If square neck diffusers are selected for returns, the fix becomes an expensive internally lined plenum box that may not fit in a crowded plenum.

3. A damper should be provided for every supply diffuser, preferably at the take-off from the main duct trunk. Dampers can produce significant noise;[3] by placing them as far away from the diffuser as possible, the ductwork and flexible duct connection can be used to dampen the noise generated by the damper.

4. Provide lay-in frames for diffusers located in drywall ceilings. This allows the diffuser to be lifted for ceiling plenum space access, which can facilitate balancing and general inspection.

5. Limit diffuser neck sizes to 12 in (305 mm). The *2021 ASHRAE Handbook—Fundamentals,* Chapter 21 recommends at least one duct diameter of straight duct.[5] As the diffuser neck size increases, the required plenum space to properly install the diffuser grows as well.

Conclusion

Most designers and engineers follow the procedure outlined in this article quickly and instinctively when laying out diffusers for most spaces. However, it is important to document our knowledge for new entrants to the industry and so that tools can be built to automate the more repetitive tasks.

References

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2. *2015 ASHRAE Handbook—HVAC Applications.*

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4. Arens, E., H. Zhang, T. Hoyt, S. Kaam, et al. 2014. "Thermal and Air Quality Acceptability in Buildings that Reduce Energy by Reducing Minimum Airflow from Overhead Diffusers." ASHRAE Research Project RP-1515, Final Report.

5. *2021 ASHRAE Handbook—Fundamentals.* ■