WHITE PAPER:

AIR DISTRIBUTION SYSTEM DESIGN

KRUEGER

Authored by Dan Int-Hout, Chief Engineer

Krueger released a new Quick Reference Product Catalog (QRPC). In this catalog, we have introduced a new way of looking at product selections in an effort to make proper selections both easier and actually perform better. In order to understand the logic behind our selection process, we have found the need to go back a bit and ensure we are all working from the same deck. So we have prepared this document to be a more in depth supplement to a webinar (titled Air Distribution System Design), which discusses the new catalog and design suggestions.

So what is our primary objective when designing the HVAC system for a space? With the looming crisis of climate change, optimum use of scarce resources, and concerns over release of carbon, we often lose sight of the biggest challenge, making the space acceptable for the occupants of the building, especially if they are working in the space. Employee salary expense is at least 200 times greater than energy costs in almost every type of building. Proper HVAC equipment selection, location, and operation can have a significant impact on the cost of running a business.

To get a common understanding of our design parameters, we are going to discuss four elements of a system design. Tools, loads, diffuser selection, and then terminal unit selection. This discussion will focus on variable and constant volume ducted HVAC systems. The supply of system air for the described systems is a subject for quite a different volume, but will include both applied (chilled water) and unitary (compressor equipped units). The delivery of the hot and cold air to the building occupants is the subject of this paper.

PART I – DESIGN TOOLS

Krueger Tools

Krueger has a number of tools available to assist the design engineer in determining acceptable products for an application. One such tool is Krueger's K-Select selection software, which can provide custom data based on your requirements. Krueger's website (www.krueger-hvac.com) is another great resource, containing the broadest range of tools available in the air distribution industry. It includes a number of reprinted (with permission) ASHRAE Journal articles, white papers, three acoustical calculation spreadsheets, a comfort tool (executable program), an acoustical specification generator, and even a link to a blog that discusses a number of air distribution topics.

The full catalog is a three volume set that includes more detailed information on Krueger's full product offering, including octave band sound power for all products and ADPI performance graphs for all diffusers (an industry first!). Published data from the full catalog is available on the web as well, along with configurable performance sheets for products with many varied sizes, such as grilles.

The QRPC presents another industry first by way of its consolidated format and unique data presentation. It lists product information for the most commonly selected products and sizes. It also makes recommendations for open plan office air outlet separation distances (assuming 0.6 CFM/sqft and NC=30) as well as minimum recommended airflow rates. Before using it, though, it seems that some explanation is required on how we came to these recommendations.

ADPI

The Air Distribution Performance Index (ADPI) is a single number rating of the temperatures and air speeds in the occupied zone of a space. It can be measured (with some difficulty) using ASHRAE Standard 113. In a sense, ADPI is a measure of how well mixed a zone is. ADPI is limited to open spaces with multiple diffusers, such as a typical open office, with an 8 - 10 foot floor to ceiling height. It is not applicable to perimeter or closed offices or to heating evaluations. It is useful; however, to note the minimum airflow performance to see which diffusers might not be suitable for VAV in a closed office, due to "excessive drop".

The ratio of isothermal throw at 50 FPM (T50) to the mid point between diffusers (an open office is assumed), is the entering argument for an ADPI prediction, based on diffuser type and room load. It was recently validated down to ASHRAE ventilation minimums through an ASHRAE research project. An ADPI of 80% or greater is recommended for most office type spaces. In addition, an ADPI of 80% can be used to prove compliance to

ASHRAE Standard 55's vertical temperature stratification limit of 5.4°F (3°C). The ADPI at a given set of parameters can be determined using the K-Select program or from the supplied graphs in the full product catalog.

Once a diffuser's performance at a specific design load and spacing is determined, it is now possible to use the same analysis to see what the lowest possible airflow rate is that will still result in a calculated ADPI greater than 80%. In the QRPC, we have listed the optimum separation distance as well as the minimum recommended flow rate at that separation.

As a note, most acoustical specifications require NC=35. ASHRAE research confirms our recommendation that one should add 5NC to published performance data to account for real world room acoustics and less than ideal inlet conditions. We have selected NC=30 as our airflow for the different sizes of outlets we are showing in the QRPC, where we then determine the area served by simple math. For a given airflow through a 4-way outlet, the separation distance is simply the square root of the quantity (airflow divided by the airflow per unit area). Note that this calculation is independent of any performance parameter of the diffuser except for sound. The ADPI parameter, "L", is half the separation distance.

Then, knowing the separation distance, one can use ADPI graphs to see what the lowest acceptable airflow rate is at that spacing. Note that not all products can be turned down.

One can also use K-Select to obtain throw data for any outlet and perform room air motion analysis directly. An alternate to ADPI is to assure that the 50 FPM throw doesn't cause collisions at the diffuser midpoint between units. It should not exceed the half separation distance plus the distance from the ceiling to the top of the comfort zone, usually 3 feet from the ceiling.

PART II – CHANGING LOADS

'Rule of Thumb'

By the mid 1980's, VAV was becoming more and more electronically controlled (DDC) and lighting was becoming stabilized at far lower levels than the 60's, when many designs were established. A 'rule of thumb' was developed for VAV design of 1 CFM/sqft in the interior. This made design much easier and it was pretty much used everywhere in North America for the next 35 years. A second 'rule of thumb' regarded acoustics. Almost all specifications for interior spaces seem to desire an NC no greater than 35. So NC35 became another 'rule', again, for many years. Finally, pneumatic controls had a finite ability to accurately control low flow. Many VAV boxes are set with a 30% minimum, again as a 'rule of thumb'. Only recently are DDC suppliers recommending lower minimum settings.

As it turns out, nothing stays the same. Lighting loads are now less than 1 w/sqft. Computers and screens draw far less current than the old CRTs and computers. Offices are more spread out. VAV terminal unit probes have been shown to be linear, well below any controller's ability to resolve the data.

The result is that interior zone loads are likely far less than the 22 BTUH (6 w/sqft) that results with 1 CFM/sqft at 55°F. When VAV boxes are set at 30% minimum, there is a likelihood that the airflow rate will exceed the space load and that the space will become sub-cooled.

In fact, a recent ASHRAE Research Project (RP 1515) studied comfort and energy at the Yahoo campus in California. They discovered that at a minimum setting of 30% of 1 CFM/sqft, the space was being sub cooled, and occupants were quite unsatisfied with a 68°F workplace every afternoon. Reducing the minimum setting resulted in a measured load equivalent to about 0.22 CFM/sqft, very close to ventilation minimums in California. This has huge implications for many locations throughout the country, as the interior load in Sunnyvale, CA is very similar to one in Saskatoon, Canada.

So what to do?

A few design engineers are dropping the 'rule of thumb' design interior airflow to more realistic levels. Others are dropping the minimum setting to code minimum values. These will both help to reduce the incidents of sub-cooling in spaces. But there is still an issue with air outlet performance and elimination of objectionable drafts, so we need to look at air outlets.

PART III – DIFFUSER SELECTION

Diffuser Selection and Location

The one constant I have experienced over the past 40 plus years with open plan offices is the question, "Where do I place my air outlets?". The answer, of course is, "It depends!". It depends on cost, performance, and aesthetics. There is often a trade off between performance and cost. Poor performance (noisy, poor pattern control, etc.) often occurs for applications with a lower (first) cost. Better performance often comes with lower sound, increased durability, and better characterized performance. An understanding of the actual performance characteristics of air outlets can help lead one to a better choice and best value.

Interior vs Perimeter

Air distribution is different between interior open and closed offices, where all that has to be dealt with are the interior loads. For spaces that reside next to external parts of the building, such as windows, both open and closed offices must take care of both interior and perimeter loads, which can affect both diffuser selection and location.

Let's start with the interior.

Interior Zones

Locations more than 15 feet from a perimeter surface are usually considered "interior" spaces. These spaces are pretty well independent of outside temperatures and humidity, except for the conditioning of the ventilation air, a minimum of which is indicated in most building codes.

Additional recirculated air is typically provided to take care of sensible loads, including lighting loads. When VAV was first employed, in the early 70's, lighting loads were quite high, nearing 6 w/sqft. With the increase in energy costs the and addition of computers, design lighting levels dropped.

Ceiling Air Outlet Performance

Open plan offices have long been conditioned from the ceiling with ceiling plenum returns. Ceiling air outlets, properly referred to a "air diffusers", are designed to distribute cool air along a ceiling surface. They work to mix room air by induction (a jet of air with lower static pressure than the slower air around it). Air moves along the ceiling surface, preventing cold air from prematurely falling into the occupied zone, known as the Coanda Effect. Performance of these devices are characterized by throw distance, sound, and pressure over a range of airflows. It is possible to do a calculation of ADPI using throw performance data, diffuser spacing, and load information by using tables in the ASHRAE Fundamentals Handbook.

Open vs Closed Offices

In closed offices (walls extending to the ceiling), air distribution is typically served with a single diffuser that is centrally located. ADPI at design loads is not useful in single outlet closed offices, as the throw will hit the walls. If it has a window, however, that has to be considered. For air distribution in open plan offices; however, jet collisions at high flow and excessive drop at low flow have to be examined. If the suspended ceiling is near 9 ft from the floor, ADPI will be a good tool.

ADPI Performance Graph

Using device throw performance data and the data in the ASHRAE Handbook to correlate throw to ADPI, a graph can be prepared with flow per unit area on the x-axis and ½ diffuser separation on the y-axis. Geometry determines the line for a given flow rate, but the 80% ADPI limits provide acceptable performance boundaries. In between these points, ADPI will be greater than 80%.

Plotting a series of flow rates for a given inlet size results in an ADPI performance graph. One can use this both to determine spacing (2 times separation distance for 4-way or round discharge). One can also determine the lowest flow at a given separation.

We can take it a step further. Let's assume a design flow rate per floor area and a design sound level.

Let's use 0.6 CFM/sqft (instead of 1.0) and NC=30. (Remember, ASHRAE research suggests to add 5 NC to published data to account for poor inlet conditions and a more realistic room effect.)

In an open plan office, with 4-way or circular pattern outlets, the separation distance at a flow rate equal to 30 NC is determined from a simple equation.

Separation Distance = (q/r)^0.5
where q = flow at NC=30
where r = flow rate / sqft

Minimum Flow Rate

Now that we have a separation distance, we can go to the ADPI chart at that flow and half the separation distance to find the minimum flow rate. In Krueger's QRPC, this information has been presented in tables so that one can quickly evaluate acceptable minimums.

What about no ceiling open plan?

Most throw performance data is for closed ceilings, where there is a surface (unless otherwise noted). If there is no ceiling; however, throw is shortened by about 30%.

An open plan space with no ceiling is best served by round diffusers on spiral drops. The larger the diameter of the supply duct, the more uniform the discharge from a short drop. Most round diffusers have a slightly upward discharge pattern when there is no ceiling, usually compensating for the lack of Coanda to avoid excessive drop of cold air.

What about linears?

In defining the area served by linear diffusers, the end-to-end spacing must be considered. At lower design airflows, (<1 CFM/sqft), a spacing of 8 ft is typical, using 4 ft linears, results in a 12 ft wide zone. Dividing this into (q/r) will yield the length of the space to be conditioned by a linear diffuser. For a 1-way discharge, this is the distance to the next outlet. For 2-way discharge, this is half way to the next diffuser, but the same total distance apart. ADPI for linear slots uses the 100 FPM terminal velocity, instead of the 50 FPM distance used for everything else, but the analysis is similar.

What about grilles?

Grilles are seldom used in open plan offices, but when done so, ADPI is typically not a valid calculation, as ceilings are usually much higher than 9 feet, and often, there is no suspended ceiling. For designs that call for opposing grilles at throws near or in excess of 100 FPM, there is a high risk of occupant discomfort due to unwanted drafts.

Like linear slots, the spacing between outlets is used to determine the area served at a given design flow rate. Grille throws may be shortened by increasing the spread (adjusting the vertical blades). Excessive drop can be controlled by adjusting the horizontal blades.

What about perimeter zones?

Perimeter loads are far less today than when overhead heating was first attempted back in the 70's. Nonetheless, all the data indicates that effective overhead heating requires room to discharge delta-t to never exceed 15°F (8.3°C). Cold surfaces of windows will generate a downward convective airflow that is best treated by directing warm (not hot) supply air down a window. ASHRAE Standard 62.1 states that if the discharge delta-t exceeds 15°F (8.3°C), or if the 150 FPM throw doesn't make it down the surface to within 4.5 ft of the floor, ventilation air will short circuit out ceiling returns, requiring additional ventilation air. The ASHRAE Handbook indicates that excessive vertical stratification will also result, negating compliance to the standard.

Perimeter Diffusers

Air outlets in a perimeter zone need to be close enough where at a 150 FPM throw, at the heating CFM, that the air projects down the window. Slots should ideally be located a couple feet away from the window, configured for 2-way discharge. ADPI really doesn't work for heating, but there are proposals to come up with methods based on recent research. It is important that the responsibility for properly adjusting linear slot pattern controllers be clearly identified at the design stage and verified after construction. This is best done by the installing contractor.

First Cost vs Installed Cost

In selecting diffusers, one needs to consider the fact that it costs on the order of $100/diffuser to connect and balance once installed. This cost is in addition to the cost of the diffuser and may make the diffuser cost inconsequential.

PART IV – TERMINAL UNIT SELECTION

In designing an air distribution system, a number of codes affect our selections. Some ASHRAE Standards are referenced in codes, some are included. Who approves a design varies considerably, but all the standards can be considered the "acceptable standard of care", so it is best to understand what they require.

Meeting ASHRAE Standards 62.1, 55, and 90.1

ASHRAE Standards 62.1 and 90.1 are referenced directly in many (most) codes. ASHRAE Standard 55 (comfort) is usually referenced, but not specified. ASHRAE Standards 62.1 and 90.1 are LEED prerequisites. They must be met to be a LEED project; compliance with ASHRAE Standard 55 is worth up to three points. The ASHRAE Handbook suggests that if air discharge temperatures are high, excessive room stratification will be likely. However, ASHRAE Standard 90.1 limits the quantity of air that may be reheated, so in northern climates, it will be difficult to maintain comfort with single duct reheat systems. The solution is, of course, a fan powered terminal unit.

Single Duct VAV Reheat Terminal Unit

Single duct (and dual duct) VAV terminal units have been in common use for almost 50 years. In moderate climates, or buildings with supplemental perimeter heat, these units can meet most occupant comfort needs. The inlet flow probe on VAV terminal units has been improved so that it is able to provide a good airflow signal over a broad range of airflows and accurate temperature control, provided that the air is distributed properly, of course. The turn down of a VAV terminal unit was limited in the past by the ability of pneumatic velocity controllers. Modern DDC controllers have greatly extended the usable range of the “pressure independent” VAV inlet.

The flow probe in today’s VAV terminals is a magnified probe. There are many designs, but the “flow cross” is most often specified. Research conducted in California a couple years ago showed that all of the different probes tested had exhibited a linear response (to the square root of the signal) down to flows unlikely to be reproducible with any DDC controller. Krueger used to recommend a minimum signal of 0.03”, but now recommends 0.01”. While a warning is shown, the customer can still order a unit with a lower design flow.

There are limits to the reheat capabilities of hot water coils based on flow, turbulence, and circuitry. Due to improved thermal resistance at the perimeter, loads are constantly dropping. At the same time, so are reheat water temperatures. Lower entering hot water temperatures and restrictions on return water temperatures with some boiler designs offer selection challenges. In fact, we are seeing requests for enlarged water coils to reduce the operating air pressure drop.

In recent years, we began to see a move towards fan assisted VAV units for a number of reasons.

Fan Powered Terminal Units

The fan powered VAV terminal unit has been in common use for 50 years. There are two types, series and parallel.

- Parallel units induce plenum air on-demand for heating *in parallel* with a VAV damper. Airflow rates for the fan and VAV damper are independent from each other.
- Series units induce plenum air, which mixes with primary air from the VAV damper and then passes through the fan. The fan must always deliver more air than the VAV valve to prevent backflow into the ceiling plenum.

With a parallel unit, the central system fan must deliver air to the diffuser, which then passes through it. With a series unit, the central system only has to get air to the VAV inlet. This results in a lower system fan static pressure requirement for series unit systems than with parallel unit systems.

Note that the series fan must run at all times during occupancy. However, with constant airflow from a series terminal unit, diffuser turndown is no longer an issue.

So which is a better choice?

While climate has a big effect on system differences, the answer has been one of local practice rather than actual data. Many building codes require summation of motor hp or other measures. Unfortunately, available building load programs do not accurately reflect the savings that result from newer technologies. ASHRAE and AHRI have prepared a joint research program to look at the system energy use of series vs parallel units and PSC (Permanent Split Capacitor Start Motor) vs ECM (Electrically Commutated Motors).

Validation of Energy Use

AHRI and ASHRAE sponsored research was initiated to provide data to allow accurate and validated energy use calculations. ASHRAE Research Project 1292 was conducted at Texas A&M University. A second project is underway, AHRI 8012, to input the results of this research into Energy Plus, Trace, HAP, and several other energy calculation programs. This will allow engineers to accurately predict the energy use of these fan powered systems. Over 24 technical papers have been prepared on the results of this study.

Realizing that ASHRAE technical papers do not get widespread exposure, three ASHRAE Journal articles have been prepared on the A&M research:

Part 1: (Oct 2017) Described the purpose.
Part 2: (Nov 2017) Summarized the findings.
Part 3: (Dec 2017) Covered issues with energy models.

The system energy data shows that in most situations, series units with EC motors will use less energy than parallel units with either type of motor. In addition, if the fan airflow on the series unit is reduced at less than design loads, rather than operated at constant airflow, energy savings can be significant. When the energy calculation software is updated, engineers will be able to document the savings.

What is the minimum recommended airflow?

A series fan powered terminal unit with EC motor can be turned down close to the primary (VAV) airflow. It will use significantly less energy when turned down, compared to design airflow. The choice and spacing of diffusers is a major determinant in how far it can be turned down. Moreover, if the series unit is coupled with a sensible cooling coil on the induction port (like Krueger's model KLPS-D) further gains in efficiency will be realized with extended economizer operation. Krueger's QRPC provides informatino to make product evaluations quick and easy.

SUMMARY

To review, the data provided in Krueger's QRPC is based upon the latest load designs and ASHRAE research, using simple calculations. The guide shows both recommended diffuser separation as well as the minimum recommended airflow at the defined spacings. This will allow the design engineer to optimize an HVAC selection for the most commonly selected types and sizes of air outlets.

Quick reference information is also provided for a complete range of other Krueger products. This, along with the prior discussions, will help the user of the new QRPC better understand the technical basis for the recommendations made.

AUTHOR INFORMATION:

Dan Int-Hout, Chief Engineer, Krueger
dint-hout@krueger-hvac.com
www.krueger-hvac.com