Basics of Duct Design

This month we begin the first of HVAC&R Nation's three-part series on ducts. Two duct design experts, JJW (Bill) Siganto and Murray Mason provide a wealth of information on the topic.

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Exposed office ductwork

What is good duct design?

Good duct design optimises:

- Duct size (manufacturing costs)
- Duct systems pressure losses (operation cost)
- Duct systems acoustics (environmental costs), and
- Air balance procedures (commissioning costs)

Of these, duct sizing is the simplest. In a few minutes, even the most complex systems can be sized with a duct sizing slide rule. The various texts suggest three methods:

- Constant friction gradient
- Constant velocity, and
- Static pressure regain

A pragmatic solution to optimise the conflicting elements in duct design, the late W R (Roy) Ahern, offered the following approach[1] for the sizing of low-pressure air conditioning ductwork.

- For air quantities greater than 4500 l/s size duct at 10m/s.
- For air quantities less than 180 l/s, size duct at 4.5 m/s.
- For air quantities greater than 180 l/s, but less than 4500 l/s

Then size duct on basis of friction loss of 1.2 Pa/m

For these rules, Ahern offered no references other than his own extensive experience backed by his unerring ability to reduce all technical programs to first principles. In the larger size where velocity is the criterion, the limiting factor is noise breakout. His choice if 1.2 Pa/m as the mid-range friction gradient is higher than that used by many designers.

For the smaller sizes, the proposed velocity limit often produces a high friction gradient. Consequent dynamic losses are quite low with the resultant velocity pressure around 12Pa.

Velocity pressure (or velocity head)

The terms velocity pressure and velocity head are interchangeable.

For air at comfort temperature the velocity pressure is 0.6 times the square of the duct velocity.

$P_v = 0.6 \times v^2$, in Pascals (Pa), if v is measured in per second.

Static pressure friction

Static pressure friction causes static pressure loss. The calculation of the friction gradient "f" (Pa/m) is complex, involving velocity pressure, duct diameter and duct surface roughness. Modern computers can make it routine. However, most designers still rely on duct sizing slide rules.

The static pressure loss in a section of duct is:

$P_T = f \times L$, where L is the duct length in meters.

Fitting lengths should be included in the estimation of "L". Where a fitting involves a change in velocity than fitting should be included in the duct section with the higher velocity.

Total pressure

Total Pressure is the sum of Static Pressure and Velocity Pressure.

$P_T = P_S + P_V$

Since one of the variables of friction gradient is velocity pressure, it follows that total pressure is dependent on the velocity pressure, which for a given duct area varies as the square of the flow. As air flows through a duct system from a fan to terminal the total pressure is always decreasing. Because of changes in velocity, the same cannot be said of velocity pressure and static pressure.

Fittings

When a fitting changes the direction of airflow or its velocity, then as well as the friction loss referred to above, a dynamic loss is involved. The quantum of dynamic loss is expressed in total pressure. Various fittings are assigned a "fitting loss factor" (K). The dynamics loss through the fittings is then calculated from,

$P_T\ (Pa) = P_V$

Where the velocity pressures is usually related to the maximum velocity occurring in the fitting. AIRAH[2], ASHARE[3] and SMACNA[4] are the most widely used references for fitting loss factors. Duct designers are often frustrated by the disparity

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between the references for K values for the same fitting, particularly for those involving divided flow. This so concerned Ahern 50 years ago that he designed his own fittings and calculated K values from unassailable first principles.

Long runout of flexible duct were not extensively used in Ahern's time, but these days it is reasonable practice to limit velocity to 3.5m/s and for friction loss calculation purposes to allow twice the measured length.

The almost universal use of "cushion head" involves an additional pressure loss of at least one and a half velocity heads[1].

Balancing designers should ensure that the duct system is proper. Unless the designer considers how the duct systems can be properly balanced, more than likely it won't be. The following is considered fundamental:

- Design for application of proportional balancing methods
- Provide adequate dampers for balance of sub systems
- Do not rely on splitter devices as balancing aids

References

1 Ahern WR, AIRAH Journal Vol.16, No.6 :"Duct sizing", June 1977

2 AIRAH Application Manual " Duct Design DA3", 1987

3 ASHRAE *Handbook of Fundamentals* 2001

4 SMACNA "HVAC Systems Duct Design" 3rd Edition 1990.

This information first appeared as the article *Basics of Duct Design* by JJW (Bill) Siganto and appeared in Ecolibrium magazine.

Further considerations

Duct sizing methods

There is no single duct sizing method that will inherently give the 'best' duct design. Whilst most people are aware of the constant pressure gradient, constant velocity and static regain methods, there is a further method known as the balanced pressure drop method and yet another more recent method known as the T-Method Optimisation.

The balanced pressure drop method is described in the AIRAH Application Manual DA3 and involves sizing the duct layout using the constant pressure gradient or static regain method, determining the index run (the path with the greatest pressure drop) and then reducing the duct sizes in all other paths (without exceeding velocity limits) such that the out of balance pressure drop in each path is minimised. The objective of this method is to achieve a more nearly balanced system thereby reducing noise and making the system more easily balanced when commissioned.

The T-Method Optimisation optimises the duct design on the basis of system capital cost and the present worth of energy. It is described in detail in the ASHRAE Fundamentals Handbook.

Fitting losses

While fitting losses can be allowed for by allowing an equivalent length, more reliable and comprehensive data is available in the form of loss coefficients (k). Care must be taken when using this data, however; because different texts base loss coefficients on different velocities in the fitting eg. the branch path pressure loss for a divided flow fitting can be expressed as a k factor based on the branch duct velocity or based on the main or upstream duct velocity. These different loss coefficients are related by:

$$\mathbf{k_u = k_B.(V_B / V_u)^2}$$

Where:

k_u = the loss coefficient based on the upstream velocity

k_B = the loss coefficient based on the branch velocity

V_u = the upstream duct velocity

V_B = the branch duct velocity

Hence the pressure loss through the branch path is:

$$\mathbf{1/2p \times k_u \times V_u^2 = 1/2p \times k_B \times V_B^2}$$

(where p = density of air = 1.2 kg/m^2)

Awareness should also be factored in for a number of fittings, notably bends. The published data must be corrected for the angle of turn of the bend. Note there is also a Reynolds Number correction. This can give a significant increase in pressure drop at higher velocities (Refer clause 6-30 of AIRAH Application manual DA03)

Fitting interaction

Another important consideration with fitting losses is that fittings in close proximity can have a higher (and in some cases lower) combined pressure loss. While it is reasonable to say that fittings should not be located close together, particularly in an S configuration, in practice this often cannot be avoided, eg. when ducts have to drop under beams. (Clauses 6-40 to 6-120 of DA3 discuss the effects of fitting interaction and also the effects of poorly configured fan layouts.)

Duct attenuation

Published data on lined duct attenuation is generally very sparse. Much of the data is for only a limited set of sizes making interpolation for intermediate sizes extremely difficult. Duct attenuation is not linear, ie. if you keep increasing the length of duct, the attenuation does not keep increasing in proportion. This is because of self-generated noise in the duct. Suppliers attempt to account for this by publishing attenuation for different lengths of duct. Thus we get the anomalous situation where two lengths of two metre duct either side of a transition gives (apparently), a higher attenuation to that of a four meter length of straight duct.

To determine the attenuation accurately, account must be taken of self-generated noise in the duct. The same applies to fittings. The noise level in a duct system does not progressively decrease away from the fan until it reaches zero. There is a lower limit caused by self-generated noise.

Duct being wrapped

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Self-generated noise

Self-generated noise is generally proportional to velocity to the sixth power (pressure is proportional to the square of velocity), the duct cross sectional area, a characteristic dimension in the case of fittings and the frequency. The formulas though not excessively complex, do mean that a complete duct design, including an acoustic analysis, is very tedious. It will also necessitate the use of a computer. If noise criteria of NR 25 or even NR 30 are to be achieved, a full acoustical analysis is essential.

Acoustical analysis of a ductwork system

The acoustical analysis of a ductwork system is normally carried out by calculating the sound power level along the ductwork system, starting with the fan. Taking account of the attenuation and self generated noise of ducts and fittings, the sound power at each air terminal is calculated. Some of this sound is reflected back up the duct leading to the terminal. The reflected noise is a function of the diameter of the duct at the connection to the terminal (the terminal neck diameter) and the frequency and design charts are available to determine this.

The noise coming from the duct (after deducting the end reflection loss) is then added (logarithmically) to the noise generated by the air terminal. This then gives the total noise entering the room at this point.

The sound pressure level at any "listener" position in the room is then calculated from the acoustical properties of the room (expressed as the room constant R), the directionality of the noise from the terminal (expressed as the directivity factor Q) and the distance (r) from the terminal to the listener position. Hence:

$$L_p = L_w + 10 \log \left[\frac{Q}{4 r^2} + \frac{4}{R} \right]$$

Where

L_p = the sound pressure level at the listener position for each terminal

L_w = the sound power level at the terminal

Good duct design

As stated, good duct design involves sizing the ducts, determining the pressure losses, calculating the noise levels, determining the out of balance pressures and optimising this against the total cost of the system. Whilst systems that are not noise critical can be simply sized using a duct sizing slide rule, commissioning can be difficult if no thought is given to the out of balance pressure along each duct path, particularly if the layout is very non-symmetrical. This can result in unexpected excessive noise levels. If the system is noise critical (eg. a TV studio or theatre complex) then there is no simple way to design the system and perform the optimisation manually. Detailed pressure drop and acoustical calculations (including self-generated noise) must be carried out. ▲

This information first appeared as the article *Basics of Duct Design*: *A further word* by Murray Mason and appeared in Ecolibrium magazine.