Pressure Drop in and Noise Radiation from Rectangular and Round Ducts –Literature Survey

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ABSTRACT

In this paper, a literature survey on rectangular and round ventilation ducts is presented. The comparison is based on two important aspects: pressure drop and noise radiation. The pressure losses in the ductwork should be kept as low as possible without jeopardizing proper control of the flow rates in the system. Pressure loss through a rectangular duct is significant higher than a volumetrically equal round one. The higher the aspect ratio, the higher-pressure loss in the rectangular system. Measurement data in the literature clearly show that the prediction error of friction factor always increases with increasing aspect ratio. Also, it is important to design a duct system that has low noise radiation since the sound propagating in a duct can break out with high enough amplitude in the residential areas to cause annoyance. It is concluded that when taking into consideration both the pressure drop in and the noise radiation from the ventilation system, round ducts generally perform better than rectangular ducts.

KEYWORDS

Pressure drop, aspect ratio, breakout transmission loss, frequency spectrum

INTRODUCTION

Ductwork is one of the most important parts in ventilation systems and ducts should be designed to transport a given volume of air as efficiently as possible. There are two basic cross-section shapes for an air duct: round and rectangular ducts. The selection of a duct system should consider both economic and technical aspects. Therefore, there is a continual interest within the HVAC industry in comparing ductworks of different cross-sections with respect to air tightness, installation and operation cost, space requirement, strength, pressure loss, noise transmission and radiation, and similar aspects, see Scandiaconsult AB's report (1992) and Ekelund (2001). In this paper, a literature survey on rectangular and round ducts is carried out to determine how the geometry of a duct cross-section can affect the pressure drop and the noise radiation into rooms and residential areas.

CIRCULAR DUCT

Pressure Loss

For a fully developed flow in a straight duct, pressure drop is caused by the friction loss, which can be calculated by the Darcy-Weisbach equation:

$$\Delta p = f \cdot \frac{L}{D} \cdot \frac{\rho \cdot v^2}{2} \qquad (1)$$

The coefficient f is usually known as the friction factor and typical operating conditions in air systems are usually with Re higher than 4000. In this region, the friction factor can be calculated based on the results of Colebrook (1938-1939):

$$\frac{1}{\sqrt{f}} = -2.0\, lg\left(\frac{\varepsilon}{3.7D} + \frac{2.51}{Re\sqrt{f}}\right) \qquad (2)$$

where ε is the absolute roughness. The duct design chart given in the ASHRAE handbook (1997) is based on absolute roughness 0.09 *mm*, which is evaluated from the work of Griggs et al (1987). Many experimental studies indicate that the Moody chart or the Colebrook equation is the best means to predict the value of f in a round duct.

Noise Radiation

The ability of a HVAC duct to retain the internal sound power is characterized by its breakout sound transmission loss. The breakout transmission loss performance of a duct is defined by Cummings (1983,1985) as

$$TL_{out} = 10\log\left[(W_i / A_i)(A_0 / W_r)\right]\text{dB} \qquad (3)$$

see Nomenclature at the end of the paper. The calculated values of *TL* in ASHRAE Handbook-HVAC Applications (1999), table 21 for round ducts are in accordance with the definition of *TL* expressed in Eqn.3. The round duct is characterized by its high transmission loss and it can prevent duct-radiated rumble noise problems. This is because the curvature of the duct walls provides a substantial increase in the rigidity, which makes it more difficult for the interior sound field to produce a deformation resulting in a net volume displacement. A round duct is far superior to a rectangular duct in containing low-frequency noise. These facts are well documented by several authors: Lilly (1983), Ver (1983), Cummings (1983), Cummings (1985), and Harold (1986). Figure 1 shows the data plotted for both the round and rectangular duct according to tables 20 and 21 in ASHRAE (1999). As can be seen, the breakout transmission loss under the frequency range of interest is better by 25 dB or more for the round duct than the rectangular one. However, at high frequencies there is no particular advantage from the round duct, see Ver (1983b) and Scott (1985).

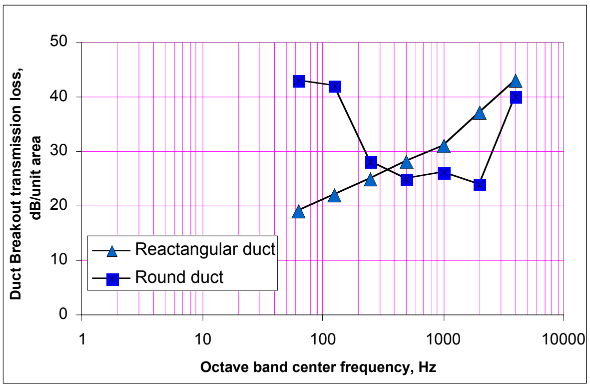

Figure 1: Duct breakout transmission loss for a rectangular duct: 305 mm by 1220 mm in 22 gage, length 6.1 m, and for a round duct: 815 mm in diameter, 22-gage spiral wound duct, and length 3 m. After ASHRAE (1999) tables: 20 and 21.

RECTANGULAR DUCT

Pressure Loss

If the duct is rectangular, the analysis of the flow pattern becomes more difficult than that of the round duct. Nikuradse (1930) is one of the first who suggested utilizing the hydraulic diameter in predicting turbulent pressure drop along noncircular duct. The pressure drop for ducts with the same hydraulic diameter, but with different cross-sectional forms, will tend to be the same for the same duct length and mean flow velocity. The hydraulic diameter is defined by:

$$D_h = 4A / P \tag{4}$$

In Figure 2, pressure losses through a round duct (*D=0.5 m, v=5m/s, _=0.15mm*) with rectangular ducts are compared with the same cross section area and flow rate. Obviously, the pressure drop through a round duct can be significantly less than the rectangular one.

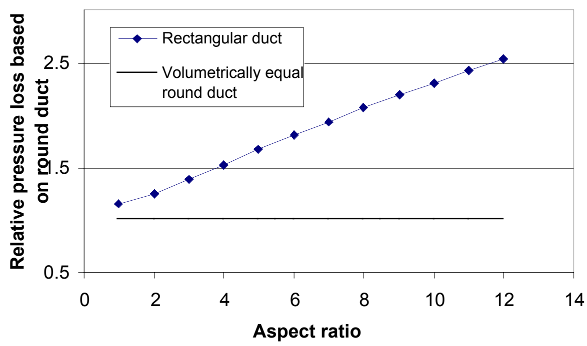

Figure 2: Relative pressure losses through rectangular ducts (airflow rate = 1 m^3/s, *v*=5m/s)

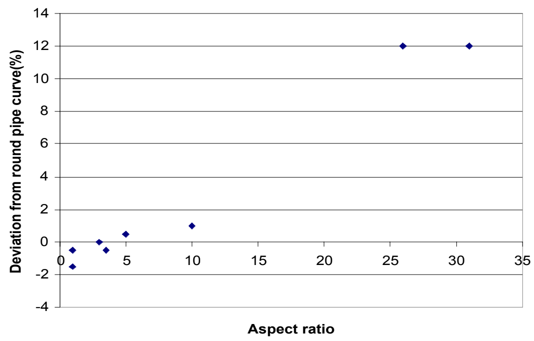

Figure 3: Summary of deviation of friction pressure loss from smooth round pipe curve for rectangular duct (evaluated from Jones 1976)

For ventilation application, "equivalent diameter" concept based on flow capacities could be more convenient for rectangular ducts. Based on the air friction chart developed by Wright (1945), Huebscher (1948) derived the circular “equivalent diameter” of a rectangular duct for equal friction and flow capacity:

$$D_e = \frac{1.30(ab)^{0.625}}{(a+b)^{0.25}} \tag{5}$$

Eqn.5. is most commonly used in USA. However, the following equation is recommended by the *European Committee for Standardization*, see CEN 1505 (1997):

$$D_e = 2b(\pi^{2-n}(1+a/b)^{1+n} /(a/b)^3)^{1/(n-5)} \tag{6}$$

where

$$n = 1/(1.05 \log Re - 0.45) \tag{7}$$

Eqn.6. implies that the equivalent diameter is dependent not only on the rectangular duct dimension but also on the Reynolds number of airflow. A study in Zou (2001) has showed that the deviation of Eqn.5. and Eqn.6. increases with the respect ratio. However, for the normal ventilation duct aspect ratio (a/b<4), the deviations are always less than 1% when the Re number is between 4000 and 10^7.

Experimental investigations as well as theoretical studies cast some doubts on the general applicability of the hydraulic diameter concept. Jones (1976) reviewed a considerable quantity of data obtained for friction pressure loss in *"smooth"* rectangular ducts. Based on the comparison between measurement data and exact solution of laminar flow, the following data set is culled to compare with smooth circular tube data, see Figure 3. In spite of lack of intermediate aspect ratio between 10 and 25, it still indicated the monotonic effect of aspect ratio on the turbulent friction factor in rectangular ducts. Similar observations can also be made in the measurements with "rough" rectangular duct, see Griggs et al (1992).

Noise Radiation

The new tabulated transmission loss data for typical rectangular ducts given in the ASHRAE Handbook (1999) can be combined with the duct dimensions to predict the *TL*.
In predicting the transmission loss of rectangular ducts, Eqn.3. can be used. For a duct of length *L* and cross section *a x b* (*a* being the largest dimension):

$$A_i = ab, \qquad A_0 = 2L(a+b)\,, \qquad (8), (9)$$

Duct fittings such as elbows will not materially affect the *TL* value but should be included in the effective radiating surface area in calculating the actual sound power from a certain length of ductwork. In this context, Ver (1983) presented two practical methods based on the work of Cummings (1983,1985) for predicting the *TL* in rectangular ducts. As indicated above, the rectangular ductwork has little capacity to contain the low-frequency noise, see Figure 1, and the portion of the duct near the fan of a large air handler will be noisy, see Scott (1985) and Lilly (1983).

In order to reduce the low-frequency noise of rectangular ducts, lagging treatment can be implemented. Although lagging treatment can be more effective at high frequencies, it is less effective for low-frequency noise as shown by Ver (1978), Ebbing et al. (1978) and Harold (1986). The sound spectrum becomes more unbalanced than before the treatment, since the rest of the sound spectrum is too quiet to provide any masking of the rumble noise, and the rumble sounds become worse even though the level is slightly reduced. The performance of lagging treatment, however, can be improved if the rectangular duct is thickly lagged, and if it is sealed airtight. Methods for estimating the insertion loss (*IL*) of lagging on rectangular ducts are available, see ASHRAE (1999). The insertion loss is defined as

$$IL = 10\log\left(\frac{W_r(without\ lagging)}{W_r(with\ lagging)}\right)\mathrm{dB} \qquad (10)$$

As indicated above, the insertion loss *IL* of rectangular ducts is high at high frequencies; however, it can create a more unbalanced sound spectrum that emphasizes the low-frequency rumble noise.

DISCUSSION

The concept of "equivalent diameter" involves the assumption that the mean shear stress should be constant around the noncircular duct boundary. In other words, the isovels must be parallel to the boundary and they must satisfy the log law up to the corner bisector.

However, in actual practice, the velocity gradient is highest at the mid-point of a side, and least in the corner, and the shear stress varies accordingly. This assumption can also be questioned since it completely neglects the existence of secondary flow. These secondary flows convect higher longitudinal momentum to the corners and this leads to the deviation from the log law much earlier than in circular ducts, see Ahmed et al (1970). A theoretical point of view, the concept of "equivalent diameter" should not be applied either to low mean velocities of flow (non-turbulent case) or to the duct whose cross-section is far from circular, e.g. rectangular ducts with high aspect ratios. Nevertheless, the specific choice of duct type and design with view of acoustical performance depends not only on the duct-radiated noise and attenuation but on other aspects such as unit location and vibration isolation of both the unit and the ductwork.

Cummings (1983, 1985) and Ver (1983) have shown that the theoretical prediction of the breakout transmission loss of round ducts is extremely complex. The prediction scheme that was found to be the best available is that proposed by Heckl and Müller (1975), which usually yields a somewhat conservative estimate for the TL. Tables 20, 21 and 22 in ASHRAE (1999) that contain the TL data have led to considerable simplifications in handling the duct acoustic performance calculations, especially for round ducts. It is worth noting that flexible and rigid fiberglass ducts often come in round configurations and may be referred to as round ducts. These types of ducts do not have high transmission loss properties because they do not have the mass or stiffness associated with round ducts. With regard to sound breakout, fiberglass or flexible round ducts should not be used.

CONCLUSION

Pressure loss through a rectangular duct is significantly higher than a volumetrically equal round one. The higher the aspect ratio, the higher the pressure loss in rectangular systems.
It is also difficult to obtain an accurate turbulent friction factor in rectangular ducts. The measurement data in the literature clearly show that the prediction error of the friction factor always increases with increasing aspect ratio.
In large air-handling units, the rectangular ducts do not have high breakout transmission loss and breakout transmission reduction at low frequencies, thus having no capacity to contain the low-frequency noise, especially in the initial run of supply ducts close to the unit. The acoustical treatment of rectangular ducts is not so effective in controlling the rumble noise. Round ducts on the other hand have superior breakout transmission loss at low frequencies, thereby containing rumble noise. At higher frequencies, however, there is no advantage from round ducts in reducing the noise radiation; in this case, acoustical treatment and lagging methods can certainly increase the capacity of the round ducts. Consequently, when taking into consideration both the pressure drop and the noise transmission, the round duct has, in general, better performance than the rectangular one. Finally, it is recommended that architects and HVAC engineers cooperate to create environmentally efficient duct systems in sustainable buildings when noise levels are to be taken into account. For instance, enough space above the ceiling tile provided for round ductwork can be very "cost-effective" compared with lagged rectangular ductwork.

NOMENCLATURE

A =cross-sectional area, A_i = cross section area of duct, A_0=total radiation surface area of duct, a =length of one side of rectangular duct, b =length of adjacent side of rectangular duct, C_f =geometry factor for laminar flow, D =diameter of duct, D_e =circular equivalent diameter of rectangular duct for equivalent length, fluid resistance and flow capacity D_h = hydraulic diameter for equivalent length, fluid resistance and mean velocity of flow f =friction coefficient ε = absolute roughness, L =length of duct, Δp =pressure drop due to friction loss, P =perimeter, R_{de} =ratio of calculated D_e value evaluated from Eqn.6. to Eqn. 5. , Re =Reynolds number based on hydraulic diameter, W_i =sound power entering the duct, W_r = sound power radiating from duct, v =mean velocity.9

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