INTERNATIONAL ENERGY AGENCY
Energy conservation in buildings and
community systems programme

Technical Note AIVC 56

A Review of International Literature Related to Ductwork for Ventilation Systems

Tor G. Malmstrom

A Review of International Literature Related to Ductwork for Ventilation Systems

Tor G. Malmstrom

2002

This report is part of the work of the IEA Energy Conservation in Buildings & Community Systems Programme - Annex V Air Infiltration and Ventilation Centre

Publication prepared by :

- T. Malmstrom, Royal Institute of Technology, KTH, Sweden;
- Z. Yue, Royal Institute of Technology, KTH, Sweden;
- O. Hassan, Royal Institute of Technology, KTH, Sweden;
- T. Eriksson, Royal Institute of Technology, KTH, Sweden;
- J. Andersson, Scandiaconsult, Sweden;
- F.R. Carrié, Ecole Nationale des Travaux Publics de l'Etat (ENTPE), France;
- P. Wouters, Belgian Building Research Institute, Belgium;
- Ch. Delmotte, Belgian Building Research Institute, Belgium.

Acknowledgement

This literature survey was made as part of the AIRWAYS project, funded in part by the European Commission (and for KTH also by the Swedish Energy Agency and Formas). All the partners in the project have contributed to the study, especially Johnny Andersson, Remi Carrié, Christophe Delmotte and Peter Wouters. At KTH especially Zou Yue has contributed to the work (also authored some parts) but also O. Hassan who has helped with the parts about noise. Jan Gustavsson, who is adjunct professor at KTH, has substantially contributed to the parts about air filters, and Professor Thomas H. Kuehn (University of Minnesota) to the description about cleaning methods during his visit as research professor at KTH.

Document AIC-TN56
ISBN 2 9600355 0 X

Annex V Participating Countries:

Belgium, France, Greece, Netherlands, Norway, and the United States of America.

Preface

International Energy Agency

The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Co-operation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster co-operation among the twenty-four IEA Participating Countries to increase energy security through energy conservation, development of alternative energy sources and energy research development and demonstration (RD&D).

Energy Conservation in Buildings and Community Systems

The IEA sponsors research and development in a number of areas related to energy. In one of these areas, energy conservation in buildings, the IEA is sponsoring various exercises to predict more accurately the energy use of buildings, including comparison of existing computer programs, building monitoring, comparison of calculation methods, as well as air quality and studies of occupancy.

The Executive Committee

Overall control of the programme is maintained by an Executive Committee, which not only monitors existing projects but identifies new areas where collaborative effort may be beneficial.

To date the following have been initiated by the Executive Committee (completed projects are identified by *):

I	Load Energy Determination of Buildings*
II	Ekistics and Advanced Community Energy Systems*
III	Energy Conservation in Residential Buildings*
IV	Glasgow Commercial Building Monitoring*
V	Air Infiltration and Ventilation Centre
VI	Energy Systems and Design of Communities*
VII	Local Government Energy Planning*
VIII	Inhabitant Behaviour with Regard to Ventilation*
IX	Minimum Ventilation Rates*
X	Building HVAC Systems Simulation*
XI	Energy Auditing*
XII	Windows and Fenestration*
XIII	Energy Management in Hospitals*
XIV	Condensation*
XV	Energy Efficiency in Schools*
XVI	BEMS - 1: Energy Management Procedures*
XVII	BEMS - 2: Evaluation and Emulation Techniques*
XVIII	Demand Controlled Ventilating Systems*
XIX	Low Slope Roof Systems*
XX	Air Flow Patterns within Buildings*
XXI	Thermal Modelling*
XXII	Energy Efficient Communities*
XXIII	Multizone Air Flow Modelling (COMIS)*
XXIV	Heat Air and Moisture Transfer in Envelopes*
XXV	Real Time HEVAC Simulation*
XXVI	Energy Efficient Ventilation of Large Enclosures*
XXVII	Evaluation and Demonstration of Domestic Ventilation Systems
XXVIII	Low Energy Cooling Systems*
XXIX	Daylight in Buildings*
XXX	Bringing Simulation to Application*

XXXI	Energy Related Environmental Impact of Buildings*
XXXII	Integral Building Envelope Performance Assessment*
XXXIII	Advanced Local Energy Planning*
XXXIV	Computer-Aided Evaluation of HVAC System Performance*
XXXV	Control Strategies for Hybrid Ventilation in New and Retrofitted Office Buildings (HYBVENT)
XXXVI	Retrofitting in Educational Buildings - Energy Concept Adviser for Technical Retrofit Measures
XXXVII	Low Exergy Systems for Heating and Cooling of Buildings
XXXVIII	Solar Sustainable Housing
XXXIX	High Performance Thermal Insulation Systems (HiPTI)
XXXX	Commissioning Building HVAC Systems for Improved Energy Performance

Annex V Air Infiltration and Ventilation Centre

The Air Infiltration and Ventilation Centre was established by the Executive Committee following unanimous agreement that more needed to be understood about the impact of air change on energy use and indoor air quality. The purpose of the Centre is to promote an understanding of the complex behaviour of air flow in buildings and to advance the effective application of associated energy saving measures in both the design of new buildings and the improvement of the existing building stock.

The Participants in this task are Belgium, France, Greece, Netherlands, Norway and the United States of America.

Disclaimer

Contents

Scope

This study is part of a SAVE-project: “AIRWAYS – Source book for efficient air duct systems in Europe”. The partners are BBRI, Belgium, ENTPE, France, KTH, Sweden and SCANDIACONSULT, Sweden. Industrial partners are ABB, Sweden, Bergschenhoek B.V., the Netherlands, and Lindab Ventilation AB, Sweden. The source book contains relevant information for choosing, designing, manufacture, installation, and maintenance of ductwork. Basis for the source book is the field experience of the partners and knowledge gathered by them, e.g. through interviews, inquires, and this literature study.
Focus of the study is on ducts but also ventilation principles, components for air treatment and similar topics, which are necessary for understanding the role of ductwork, are briefly described. The goal is energy efficient and well functioning systems.
Although one can find information about residential ventilation, this document is mainly focused on non-residential ventilation.

1 Introduction

Ductwork is used for transport of air used for ventilation or air conditioning in buildings. This study is limited to office and commercial buildings, and residences[1]. *Ventilation* is defined as exchange of air in order to improve the air quality in rooms and spaces. *Supply air* is used to control the air quality in the neighbourhood of persons while the role of *extract air* is to transport as much as possible of the pollutants out of the room. “*Air conditioning*” is used to control temperature and humidity in the rooms.
Ductwork is used for supply air as well as extract and exhaust air. The supply air is typically conditioned (filtered, warmed or cooled, sometimes humidified or dehumidified). It is important that the air is distributed properly in the building. Thus the duct system must be well balanced regarding airflow rates, or have provisions controlling the air distribution. The ducts should have a low leak rate. A primary function is also thermal insulation to protect the heat content of the air.
Another primary function of ductwork can be to transport smoke out of the building in case of fire, or assist in pressurisation of escape routes.

There are many boundary conditions regarding air ducts:

- **Energy use**. It is important to choose a technical solution that uses a small amount of energy (and exergy) for the air transportation in the duct. Low energy use also means that tight and (where appropriate) well insulated ducts should be used.
- **Minimisation of cost**. Between different layouts of a duct system, which all are able to fulfil the primary functions causing the system to be built, the one using least resources, based on the lifetime performance, should be chosen. Provided that the price of different resources as energy, material, building space and flexibility is adequate, this choice can be based on cost minimisation. In practice it is difficult to state that two different layouts of a duct system have the same quality regarding function. Whenever in doubt, proper function and low energy use should be given priority to cost minimisation.
- **Fire**. The duct system should not spread fire or smoke in the building. This gives restraints regarding e.g. duct system lay-out, duct material, and fire insulation of the ducts.
- **Acoustics**. Noise, or private conversations in rooms, should not be transmitted through the ductwork. Nor should noise be generated in the ducts so it is transmitted to the rooms. Noise generation is often governing the choice of air velocity in the ducts, resulting in velocities lower than economically optimal.

AIVC has published several reports connected to energy saving in duct systems. Some of these are the bibliography “Heat Pumps for Ventilation Exhaust Air Heat Recovery” (Limb 1996), the Technical

[1] Industrial applications are covered in the Industrial Ventilation Design Guidebook, Academic Press 2001.

Reports "Air to Air Heat Recovery in Ventilation" (Irving 1994), "Energy Impacts of Ventilation – Estimates for the Service and Residential Building Sectors" (Orme 1998), "Energy Requirements for Conditioning of Ventilation Air" (Colliver 1995) and "Acoustics and ventilation" (Ling 2001).

2 Air transport through ductwork

Ducts are used to separate room air and ventilation air. The reasons are:

- To make controlled ventilation air flow rates possible within the building;
- To enclose polluted exhaust or extract air;
- To keep supply air clean and conditioned (which requires thermal insulation of the ducts);
- To make heat recovery from exhaust air to supply air possible.

In some cases such separation, i.e. protection of supply or indoor air is not needed. Then ductwork may not be necessary. Separation can also be obtained in other ways, like plenums.

Some consequences of using ducts are:

- If fans are used, air displacement energy has to be supplied. This typically is electric energy (which is pure exergy). In order to save the valuable electric energy the pressure drop in the ductwork should be kept as low as possible without jeopardising proper control of the flow rates in the system. This increases the demand for careful aerodynamic design.
  Fan energy is transferred to the handled air as heat. For air heating some of the fan energy is thus recovered, but for cooling applications the fan heat results in an increased cooling load.
  It is important to remember that the pressure loss in an Air Handling Unit typically is of the same magnitude as the pressure loss in the ductwork. In order to save fan energy flow conditions in this unit should also be optimised.
- The ducts should be tight in order to save both fan energy and thermal energy (if the air is heated, cooled, or subject to heat recovery).
- The ducts should be insulated in order to save thermal energy, resist fire, and (for cold air) to avoid condensation on the ducts outer surface.
- The ducts should not pollute the supply air. This can be avoided by keeping the ducts clean and dry. (Please note the discussion in chapter 12).
- The ducts have to be kept clean in order not to change the flow resistances and, as a consequence, the flow balance.
- The ductwork has to be designed so spread of fire and fire smoke is minimised.
- Transfer and generation of noise has to be controlled.

3 Ventilation principles

Ventilation systems are of different types, often named after the driving force:
- "Natural" ventilation systems where the driving forces are thermal buoyancy ("chimney effect") and wind pressure;
- "Mechanical" ventilation systems where the driving force mainly is caused by fan(s).

The mechanical systems may be subdivided into:
- Exhaust systems where only the exhaust air flow is controlled by fans;
- Supply systems where only the supply air flow is controlled by fans;
- "Balanced" systems with both exhaust and supply fans;
- Balanced systems with heat recovery;
- Balanced systems with return air.

"Natural" ventilation systems sometimes have small flow rates, for instance when the temperature difference between indoor and outdoor air is small. To increase the airflow rate fans then can be used. Such fan-aided natural systems are sometimes called "Hybrid ventilation systems".

An important boundary condition for the design of ventilation systems is the risk for condensation damages in the building envelope caused by pressure differences. In cold climates such damages are typically due to convection of humid indoor air into cold parts of walls and roof. Thus a slightly lower air pressure indoors than outdoors is often recommended. This can be achieved in mechanical systems by using a higher fan-controlled exhaust airflow rate than the supply airflow rate.
Sometimes additional air flow (caused by wind or thermal forces) through leaks causes "overflow". Then, all the air coming into the building cannot be extracted by the ventilation system, which can cause condensation damages due to out-leaking air. This is often the case when the indoor air is warmer than the outdoor air. Such condensation typically occurs in the upper parts of the building. In the cooling case, when indoor air is colder than outdoor air, the thermal forces are reversed.

An important difference between "natural" and mechanical systems is that in the "natural" systems the airflow is caused by the pressure differences between indoor and outdoor air while in fan driven systems the resulting flow rates create pressure differences between the interior of the building and the outdoors. To minimise risk for condensation damages for "natural" ventilation systems, flow rate controlling devices should be located at the supply side.

Ventilation and air-conditioning systems can also be classified on basis of airflow control:
- Constant airflow rate systems
- Variable airflow rate systems (VAV)

VAV-systems have the potential to save electrical fan energy, which is a big advantage.
"VAV" is often associated with air-conditioning systems, controlled by room air temperature, and adjusting to different loads by changing the airflow rate. Variable airflow rate systems controlled by air quality sensors or the number of persons present in a room are sometimes called "demand controlled ventilation systems".
A method to vary the flow rate of outdoor air in constant airflow rate systems is the use of *return air*, i.e. exhaust air which is recycled into the supply air.

Advantages of using return air are:
- That it makes it possible to use the simple constant flow air distribution system (in relation to a variable airflow rate system);
- That it has the potential to decrease the concentration of particles in the rooms as air passes the filter(s) more than once.

Disadvantages of return air are

- That pollutant from one room (e g. perfume odour) can be supplied to the whole ventilation zone. This is especially important when the zone consists of several rooms and not an open space, e.g. an office landscape, as humans want privacy and are said to be protective of their territories.
- That exhaust air normally is polluted with particles. Care has to be taken that dirt is not deposited in the ducts being passed by the return air. This can otherwise cause later pollution of the supply air.
- That the use of return air to vary the flow rate of outdoor air uses more fan energy than a VAV-system without return air.

Ventilation principles are further discussed in Handbooks, see e.g. "A guide to energy efficient ventilation"(Liddament 1996), published by the Air Infiltration and Ventilation Centre (AIVC), UK.

4 Thermal forces and wind forces

The air duct system is of course part of the building total airflow system. The building is exposed to *wind forces*, creating over pressures on the wind side and under pressures on the leeward side. This influences the total air exchange in the building as it imposes airflow through the building. It also interacts with indoor air pressures, which can change the airflow rates in the duct system. The duct system can be directly exposed to such forces, if they act on the air intake or exhaust openings, see Figure 1.

The *thermal forces* result from differences in air density caused by temperature differences. In a cold climate, where outdoor temperatures are lower than indoor temperatures, these forces add to an upward flow in the building but decrease a downward airflow. Thus the lay out of the duct system will cause different interactions with the thermal forces (Rydberg 1959 and 1968, Feustel and Esdorn 1982). A system with the air intake placed low in the building and the exhaust openings at roof level (as natural ventilation systems are designed) will have higher airflow rates in wintertime than in summertime. (Such layouts often are found in older buildings designed before the age of car traffic, when the main pollution sources were chimneys) A system with both air intake and exhaust openings at roof level will tend to move the "neutral plane" (which is at the height in the building where indoor and outdoor air pressure is equal) down. The opposite is true for a system with both air intake and exhaust located at ground level.

The pressure difference Δp_h (Pa)due to thermal forces is

$$\Delta p_h = g\, h\, (\rho_o - \rho_i) \qquad (1)$$

where	g	acceleration of gravity	(m/s^2)
	h	height of building	(m)
	ρ_o	density of outdoor air	(m/kg)
	ρ_i	density of indoor air	(m/kg)

If this air pressure difference is the only one driving the airflow, it is used to overcome the flow resistances in the building, see Figure 2. It is evident from the figure, that to avoid big influence of stack effect in the building (that is big differences between indoor and outdoor air pressures) it is important that the *indoor* openings are small, e.g. around stair and lift shafts.

Wind forces Δp_w *(Pa)* are estimated with the formula

$$\Delta p_w = C_d\, \rho_o\, v_w^2/2 \qquad (2)$$

where	C_d	form coefficient	(-)
	v_w	wind velocity	(m/s)

How much wind and thermal forces influences the airflow rates depend on how big they are in relation to other forces in the system (as fan forces), that is how big the pressures they induce are compared to e.g. pressure drops in the duct system. Often pressure drops in air handling units are big and govern the total ventilation airflow rate in the ductwork. This can also be the case in domestic air systems if they are equipped with heat recovery.

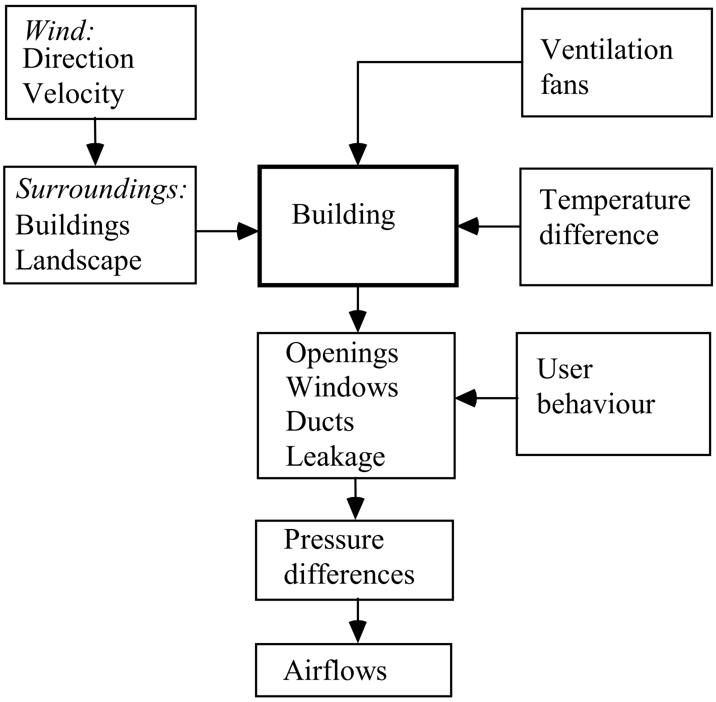

Figure 1 : The air flow system of a building

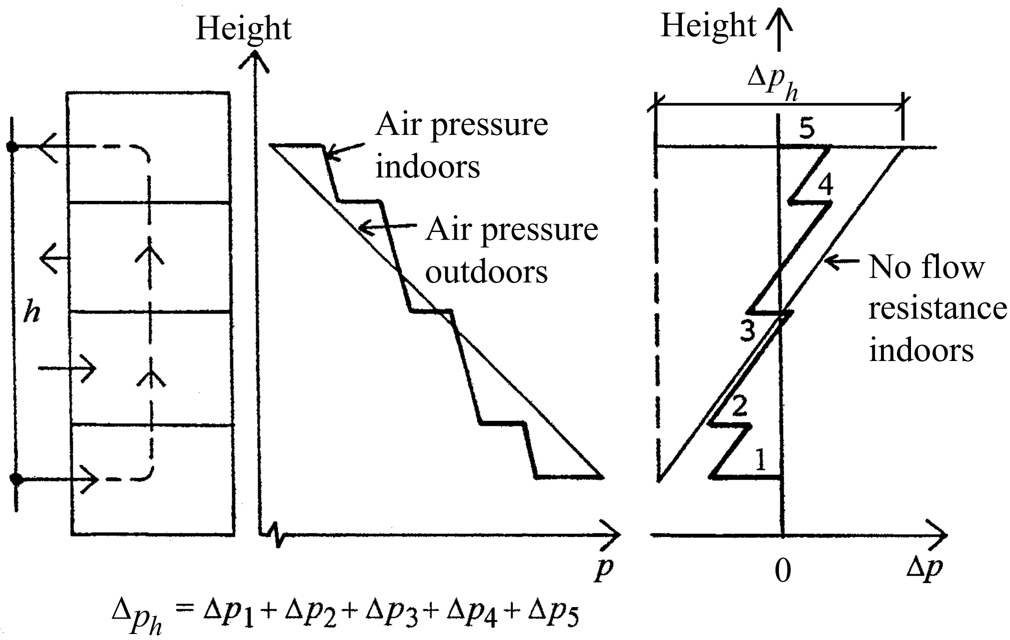

Figure 2 : Air pressure profile in a four storey building with flow resistances both in the outer wall and internally. The pressure difference Δp_h is caused by a density difference between outdoor and indoor air (heated indoor air).

5 Ducts and ductwork components

Straight rigid ducts (rectangular, circular, and oval) are made of:

- Metal (galvanised sheet metal, stainless steel, hot-rolled (and painted) steel, aluminium, or sheet metal with aluminium-zinc coating);
- Synthetic material (PVC, polyamide, etc.).

Flexible ducts are made of:

- Synthetic material (as PVC or polyamide) wrapped around a metal spiral coil;
- Metal (stainless steel or aluminium).

The cross-section of flexible ducts is circular. Their interior surface is in general either rough or bumpy (e.g., if the material is wrapped around a metal coil). Although widely used mainly because they seem easier to install, these ducts generate much higher pressure drops than rigid ducts.

Beside the ducts, duct systems also are equipped with components as:

- Area changing parts, reducers;
- Bends, sometimes equipped with vanes in rectangular ducts;
- T-junctions, dividing in supply systems and combining in exhaust systems;
- Saddles, to attach a take off duct to an existing main duct;
- Couplings;
- End caps;
- Cleaning covers;
- Silencers, to reduce fan and flow generated noise;
- Dampers, to stop air flow, to protect against spread of smoke, and to balance the air flow rates;
- Constant flow devices;
- Flow meters;
- Mixing boxes, to mix warm and cold air;
- Plenums and distribution chambers;
- Duct hangers.

6 Duct form

Rigid air ducts are circular, rectangular, or oval (less common today). Circular ducts typically are made from a 137 mm wide steel strip, spirally wound and seamed and formed into a round cross section. Flat oval ducts are made from circular ducts that have been formed into an elliptic shape in special machines.
Circular ducts are manufactured in standard sizes according to the table below (HVCA 1998 and Lindab 2000).

Diameter mm	100	125	160	200	250	315	400	500	630	800	1000
Plate Thickness mm	0.5	0.5	0.5	0.6	0.6	0.6	0.8	0.8	0.8	0.8	1.0

Table 1 : Standard diameters for round ducts.

Rectangular ducts were earlier most frequently used and still are in many countries. The use of circular ducts seems however to be increasing, and Evans and Tsal (1996) give the recommendation that circular ducts normally are most cost-effective. Also ASHRAE Fundamentals (1997) recommend that circular ducts should be used whenever feasible. This was already stated in the Carrier Handbook (1965).

It is in practice very rare to find a ventilation system with only circular ducts. Rectangular ducts are often used to overcome local space problems and when the air ducts are very big, as close to the intake of outdoor air (Ekelund 2001).

Circular ducts are easier to manufacture, make tight, handle and install (Ekelund 2001) than rectangular ducts (and can often also be made by thinner metal), and thus normally are less expensive. Also, for transporting the same airflow at the same pressure loss (that is, with the same equivalent diameter), the sheet metal area of a quadratic duct is 20% larger than that of a circular duct. The corresponding number for a rectangular duct with side ratio 5/1 is 90%. This also means that the heat losses to the surroundings are bigger for a rectangular duct and/or that it must be insulated better.

Bouwman (1982) compared minimum costs for one circular, one quadratic, and one rectangular (aspect ratio 4:1) duct. The cost included fan energy, duct and space cost, and cost for heat loss. The ducts were externally insulated. He found that the quadratic duct had 10% higher cost than the round duct, and the rectangular duct 35% higher.
As a rule of thumb, Bouwman writes that the optimum velocity varies as

$$v_{opt} \propto C^{-2/29} \qquad (3)$$

where C is a factor depending on the form of the duct:

$C = \sqrt{\pi}/2$ for a round duct

1 for a square duct

$\frac{1+a}{2\sqrt{a}}$ for a rectangular duct (a= the aspect ratio)

The influence is small. In practice the optimum velocity is independent of the duct form.
In Figure 3, pressure losses through a round duct (diameter *D=0.5 m, velocity v=5m/s, surface roughness ε=0.15mm*) are compared with rectangular ducts with the same cross section area and flow rate. Obviously, the pressure drop for a round duct is significantly less than for a rectangular one, at the same volume flow rate and velocity. As duct sizing typically is made by choosing duct velocities, this reflects the actual situation for many applications.

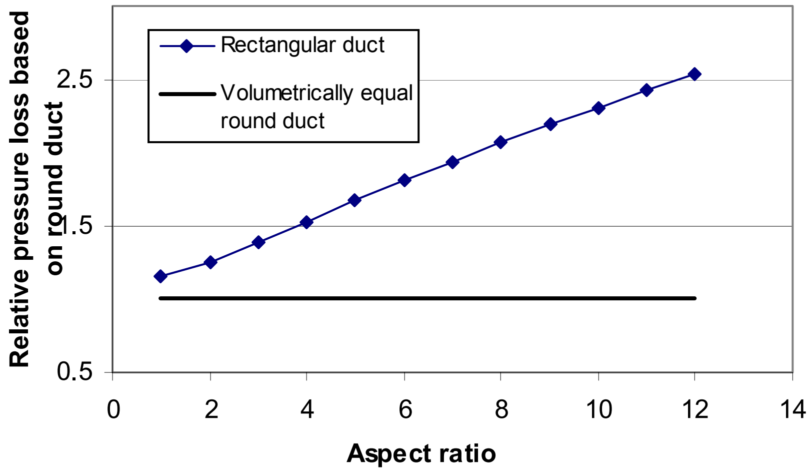

Figure 3 : Relative pressure losses through rectangular ducts (volumetric air flow rate $q_v = 1$ m^3/s and velocity v=5 m/s (Hassan and Yue 2001). Note, that (at least in Sweden) design engineering practice is to avoid (if possible) aspect ratios bigger than 3.

One reason for using rectangular ducts is that they can use the available space in a more efficient way than circular ducts, especially if the side ratio of the space is big. For such cases an alternative is to use several circular ducts in a row. Thus the relative space demand and cost for several small round ducts instead of one big has been investigated by Jagemar (1991). The calculations have been made for the same pressure loss per m for all cases.

A decrease of the space height to 60% of that needed for one circular duct can be obtained when 6 parallel circular ducts are used. However, the width of the space has to be increased 2.5 times. If a rectangular duct is used instead, the width has only to be increased 2 times.
Also the amount of sheet metal needed has been calculated. It is very approximately for:

- 2 circular ducts ca 1.25 times that for one rectangular duct;
- 3 circular ducts ca 1.5 times that for one rectangular duct;
- 6 circular ducts ca 1.7 times that for one rectangular duct.

The relative first cost has been investigated. Approximately it is for:

- 2 circular ducts ca 1.5 times that for one circular duct;
- 3 circular ducts ca 2 times that for one circular duct;
- 6 circular ducts ca 2.75 times that for one circular duct.

The span of the numbers is big. For instance, the last mentioned value varies between 2.4 and 3.1.

For rectangular ducts, Jagemar (1991) found that the first cost was about the same as for several circular ducts in a row.

It is possible to make rectangular ducts as tight as circular ducts, but it costs.
Thus the total energy use for rectangular ducts is larger than for circular, as both friction and leakage normally increases. As the investment cost is about the same for one rectangular duct and several circular, it is thus normally a good alternative to use several circular ducts instead of one rectangular (Jagemar 1991).

There are also differences between round and rectangular ducts regarding strength and acoustics (see chapters 10 and 15).
Typically rectangular ducts are weaker than round ducts.

Round ducts contain low frequency noise better than rectangular ducts, that is the transmission to the surrounding space is smaller, but round ducts instead depend more than rectangular on silencers to absorb low frequency noise which is transferred within the duct, (Hassan 2001).

7 System lay out and Sizing methods

It is always an advantage to have a symmetrical layout. For a constant flow rate system, a perfectly symmetrical duct system is self-balancing, at least when there are no thermal or wind pressure forces.
In practice duct layout often is governed by factors as the location of air handling units (fan rooms), shafts, fire partitions, and other building characteristics. This is of course wrong; the whole building system should be optimised as a total. However, there are no such methods or tools available. Building design still is an art of successive sub-optimisations. Ductwork "optimisation" is one of them.

Important factors to take into consideration are:

- Space demand;
- Size of airflow rates to different parts of the building (which is connected to the need of shafts and horizontal ducts);
- Balancing;
- Sound generation;
- Duct leakage;
- Heat losses;
- Fire and smoke;
- First cost;
- Cost for fan energy;
- Maintenance cost.

A recent study is Wigenstad (2000). He divides the system in five elements, which for ventilation applications are (relative cost is also given):

- Air intake (2%);
- Air handling equipment (40%);
- Vertical distribution (5%);
- Horizontal distribution (25%);
- Room related installations (28%).

The cost figures are related to Norwegian circumstances. According to Wigenstad (2000) the total cost for ventilation (without refrigeration) was ca 50 euro/m^2 served floor area.
Wigenstad (2000) points out that ventilation installations often are integrated with the building (false ceilings, walls, etc.). It is then important to harmonize the lifetime of the different parts. Such integration sometimes decreases the flexibility, which always should be considered. In many offices today the room partitions are changed up to several times a year.
For offices and hospitals the space demand for air handling equipment rooms typically is 8-12% of the building volume, and for shafts and ductwork 2-3% (Jagemar 1991).
In order to fulfil the primary function criteria, proper airflow rates in the duct system, the length of the duct, from air handling unit or extract fan to the last air terminal device, must not be too long. 60 to 80 m is often cited numbers. This includes then both a horizontal part and a vertical part of the duct, the vertical part should be shorter in order not to have too big influence of thermal forces, due to density differences between duct air and outdoor air. This height depends on the maximum density difference and the size of pressure losses in the air terminal devices, the ductwork and in air handling units.
The length of the horizontal distribution ducts is also related to building height as larger areas to be covered also means bigger ducts, which results in demand for increased storey height. This problem was studied by Bouwman (1982). He found that if the cost for increased building height was taken into account, an economical floor area per storey, to be served from one shaft, was 250-400 m^2. If the duct layout *not* influences the building height the corresponding value was 1000-2700 m^2.

One important factor is crossing of supply and exhaust ducts. This typically happens with ductwork located at office corridor ceilings with office rooms on both sides of the corridor. This can be avoided by location of one of this ductwork, normally the supply ducts, in the façade. Proper thermal insulation is then essential.

The cost for the air ductwork is material-, manufacturing- and transportation costs for the duct and insulation (including recycling costs), cost for the building space, cost for the installation of the ducts, and cost for cleaning and maintenance. All these costs tend to increase with duct diameter or size, even if there are important exceptions as decreased installation costs if ducts are of the same size (because of simpler handling) or slower fouling of large ducts compared to small for the same air flow rate (because of smaller air velocities).

7.1 Duct sizes for CAV-systems (Constant Air Volume)

Typical values used in Sweden are the following recommendations for circular ducts from a company manual (Jovent 2001). The current tendency is towards lower velocities.

Dimension (mm)	Velocity (m/s)	Friction (Pa/m)
1250-630	8	1-0.5
500-400	7	1-1.3
315-250	6-5	1.3-
200-160	4	1.1-1.4
125-100	3	1.1
(080-063)	2	0.8-1.5

Table 2 : Examples of velocities and friction rates common in Sweden (Jovent 2001). The somewhat high velocities for big ducts reflects the fact that such ducts typically are found in fan rooms and shafts where duct noise is less of a problem.

Important factors when choosing duct sizes are among others cost minimization, energy saving, noise level, and flexibility.

7.1.1 Cost minimization

The total cost for the ductwork is difficult to estimate. It should take into account the cost for the duct itself, thermal insulation, noise attenuation, fire precautions, the space needed in the building and building costs in other ways related to the ductwork, installation cost, maintenance cost, exchange and destruction cost, cost for fan energy and heat losses (also due to leakage), and anticipated cost for future changes of the ductwork (flexibility). In practice only some of these factors are taken into account when making cost minimizing studies.
It is important to note that, for a constant air flow rate in a duct, fan energy due to friction decreases rapidly, with increasing duct diameter and corresponding decrease of air velocity (typically $\propto v^{2.4}$), while most other costs increase with increased duct diameter but much slower (typically $\propto v^{--0.5}$). Thus the cost for fan energy has a strong influence on the “economical air velocity”, although the actual cost is a minor part of the total cost at the minimum point (according to Bouwman (1982) typically 17% and down to 7% if continuously increasing energy price is anticipated).

7.1.2 Single, unbranched duct

Bouwman (1982) used the air velocity in the duct as the main variable in seeking cost minimum. The corresponding “optimal velocity” v_{opt} m/s for a single transportation duct with no branches depended approximately on different factors as follows:

- $v_{opt} \propto$ (air dynamic viscosity)$^{-2/29}$
- $v_{opt} \propto$ (air density)$^{-8/29}$
- $v_{opt} \propto$ (air flow rate)$^{1/29}$
- $v_{opt} \propto$ (total cost for air duct)$^{10/29}$
- $v_{opt} \propto$ (fan efficiency)$^{10/29}$

- $v_{opt} \propto$ (yearly time of use)$^{-10/29}$
- $v_{opt} \propto$ (electricity cost)$^{-10/29}$
- $v_{opt} \propto$ (friction factor)$^{-10/29}$
- $v_{opt} \propto$ (duct form) as
  $v_{opt} \propto C^{-2/29}$
  where C is a factor depending on the form of the duct:
  C= $\sqrt{\pi}/2$ for a round duct
  1 for a square duct
  $\frac{1+a}{2\sqrt{a}}$ for a rectangular duct (a= the aspect ratio)

Note, however, that the material cost for the rectangular duct can be higher than that for a circular one (due to thicker sheet metal etc), which gives additional variation.

It is interesting to note that according to this study the optimal velocity is almost independent of the airflow rate q_v (m^3/s), and has a very small dependence on the duct form. In practice these dependencies can be neglected. The total duct cost influences the optimal velocity as the inverse of the third root of its value, and the fan energy cost influences proportional to the third root of its value. Thus the change of the optimal velocity for variable changes is rather moderate.

The dependencies for v_{opt} of electricity cost and annual time of use according to Bouwman (see above) give changes close to those predicted by a Swedish study with different approach (Jacobsson and Udd, 1982).

The optimal velocity above is the velocity when total cost is minimized. As the total cost varies slowly close to its minimum, substantial fan energy savings can be made at a small price, e.g. reduction to 50% fan energy for a total cost increase of 5%. This is reflected also in the recommendations for lower velocities when the electricity price is anticipated to increase.

Jagemar (1991) studied economical velocities in another way, that is the pay-off time for the investment to increase the duct size. The studied duct was externally insulated and had 0.32 (components/ m duct), each with total pressure loss coefficient ζ=0.6, which the author found to reflect normal practice. The economical velocity found in this way was about 5 m/s (the velocity in the bigger duct).

Exergy use for the transport of air (with a temperature difference to the surroundings) was studied by Gerdov (1996).

7.1.3 Branched ducts

For branched ducts the situation is of course more complex than for the unbranched. (Branched duct here means a main distribution duct from which air is branched off to local ducts.) Bouwman (1982) also studied this case. Optimal values depend of course on the cost relations. Bouwman's values relate to prices in Holland about 1980. For the most common sizing method, the equal friction method, he found the following values of v_{opt} m/s:

Number of branches, n	v_{opt} (m/s) [branch 1-branch n]
80	8.5-3
40	8.5-3.5
20	8.5-4
10	8-4.5
5	8-5.5

Table 3 : "Economical velocities" for a branched duct (Bouwman 1982)

7.1.4 Energy saving

For *heat* it is important to have well insulated ducts to minimize heat losses, and tight ducts to keep the warm or cold air in the duct. Note that especially loss of cooling capacity can lead to increased use of electricity, as the refrigeration equipment has to work more. Normally bigger ducts increase transmission losses because of the increased surface. Leakage also may increase due to increased surface, but may at the same time decrease due to decreased pressure level in the ducts. The best solution is of course tight ductwork.

Fan energy normally is electricity. As mentioned before important savings can be made at low cost by keeping air velocities on the low side of the total cost minimum.

Jagemar (1996) discussed target values for air distribution systems. A measure of the pressure losses in the system is the Specific Fan Power (SFP), or if only a single fan is studied, the Specific Fan Power for the Individual fan (SPFI). The specific fan power is measured in kW/(m^3/s), that is the fan power necessary to distribute an airflow of 1 m^3/s through the system. SFP is a measure of the total pressure losses Δp_{tot} in the system:

$$\mathrm{SFP} = \sum \Delta p_{tot} / \eta \qquad (4)$$

η is the fan efficiency. Normal values for the total efficiency are (VVS-AMA 1998):

Centrifugal fan , backward curved blades	65%
Centrifugal fan , forward curved blades	50%
Centrifugal fan , straight curved blades	50%
Vane-axial fan with guide vanes (ducted)	60%
Unducted axial fan	55%

Table 4 : Total efficiencies for different types of fans

Measured values of the total fan efficiency for HVAC applications are in the range 35-80 %, depending on fan type and size (Jagemar 1996). Total efficiencies higher than 70% are rare.
In order to get a low SFP high fan efficiency and low pressure drop are needed. To reach a SPF(I) target value a more efficient fan should always be considered.
According to Jagemar (1996), measured values of SPFI typically was at that time in the range 1-2 kW/(m^3/s). Target values of SPFI should be below 1 kW/(m^3/s). (Jagemar 1996) as European and American standards demands a SPF of less than 2 kW/(m^3/s).
With normal fan efficiencies this means that the total pressure drop in the system connected to the fan must not be bigger than 500-650 Pa. As an air handling unit typically has a pressure drop of 200-300 Pa, and the air terminal device may demand a pressure loss of 40-50Pa, this means that the pressure loss in the duct system should not be higher than 200-300 Pa. To achieve a low pressure demand care has to be taken when designing the system, as well as when it is constructed.

7.1.5 Noise

In practise, noise criteria sets an upper limit to duct air velocities, which often are lower than the economical one. According to Jagemar (1991) the sound level changes with the fifth power of the air velocity:

$$\Delta L_w = 10 \log (v_2/v_1)^5 \qquad (5)$$

This means that a velocity increase of 25% (v_2/v_1=1.25) results in an increase of sound level of 5 dB, which corresponds to an additional 0.5 m of silencer.

According to Andersson (1998) the sound level L_w in a duct can be estimated as:

$L_w = 10+50\log v+10\log A$ (6)

Where v air velocity (m/s)
A square area of the duct (m^2)

The mechanisms for sound generation in components differs from that for straight ducts, see Andersson (1998).

Op't Veld (1994, 1996) has studied noise in ventilation systems for dwellings. He recommends 4 m/s as a maximum in main and branched ducts, and 2 m/s as a maximum in ducts with supply air terminal devices.

Jagemar (1991) has compiled literature data regarding recommended air velocity in ducts with regard to noise:

Location	**Recommended velocity (m/s)**		
	Stampe (1982), dwellings	Stampe (1982), schools and offices	Höglund (1986)
Ducts in fan rooms and shafts	-	-	-
Main ducts	3.5-4.5	5-6.5	5-8
Branch ducts	3	3-4.5	3
Ducts with air terminal devices	2.5	3-3.5	1.5-2

Table 5 : Some recommendations for air velocities in ducts with regard to noise. For ducts in fan rooms and shafts somewhat higher velocities can be used.

The velocities in the table are recommended values which have a marginal to maximum values.

7.1.6 Flexibility

To design for lower velocities than those recommended from cost and noise point of view decrease the demand of fan energy, but also increases flexibility as future additions as well as changes of flow rate or flow distribution is easier.
In this connection it should also be mentioned that many architects prefer constant duct size when the ducts are visible.

7.1.7 Sizing methods

Bouwman (1982) studied different design methods and found that they resulted in different systems but that the total cost not varied much. Among the methods he studied were (in order of decreasing ratio of biggest and smallest duct diameter in the system):

- Constant velocity;
- Equal friction;
- "most economic";
- Static regain;
- Constant diameter.

Some of these methods will be discussed below.

7.1.7.1 Equal friction

The philosophy is to distribute the friction losses evenly in the system. It works well with symmetrical duct system layouts. For non symmetrical systems, dampers has to be used to balance the system, or higher friction rates have to be chosen for the shorter duct paths This method gives higher velocities for the larger ducts close to the fan.

Bouwman (1982) found that an “economical value”, based on first year costs, was close to 2 Pa/m. This is also recommended in USA, see e.g. Kuehn et al (1998).
However, if an increase in electricity price is anticipated, a lower value should be chosen. Bouwman recommends about 1 Pa/m, which roughly corresponds to a decrease of the velocities to about 70% of those calculated as economical based on the first year costs. This is close to the values in Table 5. The velocities have to be checked regarding noise.

7.1.7.2 Choice of velocity

A much used method is Choice of velocity, different in different parts of the system (values normally in the range 6- 2 m/s, the higher values closer to the fan. When space is expensive as in high rise buildings and ships, the velocities may be higher.) This is because of its simplicity the most common method. The lower velocities used in ducts far from the fan help to decrease noise, increase flexibility, and sometimes to make visible ductwork more attractive by avoiding dimension changes.

This method can be regarded as a variation of the equal friction method, where considerations has been taken to noise generation aspects. Andegiorgis and Peterson (1986) give the following recommendations based on international literature studies and Swedish experiences:

	Air velocity (m/s)	
	Dwellings	Offices, schools etc
Main ducts	4	6
Branch ducts	3	4.5
Duct with air terminal device	1.5	2

Table 6 : Duct velocity recommendations according to Andegiorgis and Peterson (1986). In offices, higher velocities can be used for ducts in fan rooms and shafts.

A comparison with recommendations given in 7.1.5, shows that these recommendations are similar to those of table 5.

A variation of this method is the “Constant velocity” method. Then the “economical” velocity is used in all the ductwork. This velocity often is around 5 m/s. The drawback is of course the noise problem as the velocities are high in the ducts close to the served building areas. The method is therefore not much in use.

7.1.7.3 Static regain.

This method aims at constant static pressure along the supply main duct. This is achieved by using “regained” kinetic flow energy (dynamic pressure) to cover the pressure losses. The velocity is lowered at the branches, where the airflow rate is decreased. Part of the resulting change in kinetic energy is lost, and the rest is regained. The “regain factor” is often not known with accuracy enough to make detailed calculations possible (ASHRAE Handbook 1997). This method results in high velocities close to the fan, as the corresponding kinetic energy shall cover the flow losses. An advantage is that the static pressure in the ducts will be close to the atmospheric pressure, which decreases leakage.
Compared with the equal friction method the resulting ductwork will have less variation in duct size, with smaller ducts and higher velocities close to the fan and perhaps also bigger ducts and lower velocities close to the served rooms.

According to Kuehn et al (1998) “static regain” is a typical “high velocity system” method used where space is very expensive. It is thus beside the scope of the present study.

7.1.7.4 30 % rule

This is a simpler variation of the Static regain method. When the flow rate has decreased to 30% of the rate at the previous area change of the duct, the duct area is decreased to the next standard size.

7.1.7.5 T-method

(Tsal and others, five publications 1987-1998).
This method represents here a class of ductwork optimisation methods taking into account costs which apply to the specific building being designed, which class has been called above "most economic" methods. These methods use if possible smaller ducts instead of dampers to balance the system, thus decreasing the cost. They are more feasible to use in USA than in Europe as ducts are available in many more sizes in USA. It can also be used for simulation of the flow rates in the duct system.
It is a method intended for use with computers. As cost for different types of ducts and for electrical energy can vary rather much, such methods have a potential to save energy, especially in uncommon applications, where standard methods, developed with experiences from normal applications, is not applicable. For more normal applications, as represented by an example in the ASHRAE 1985 Handbook, Tsal et al (1988) claims cost savings of 15-20 % when using the T-method. However, this cost reduction was achieved by decreasing the velocities in two duct sections from 10.5 m/s to 4.7 m/s respectively from 6.1 m/s to 4.1 m/s, that is to "normal" velocities as compared with the velocity rule above. Design time is also increased when using the T-method or similar. One very big advantage is, however, that the method also allows simulation of the system. Most computer based methods also makes redesign of the system easier.

7.2 Duct sizes for VAV-systems

The time of full utilization of the duct system is shorter for a VAV-system than for a CAV- system. Consequently, the "economical velocity" is higher, as it will be used a shorter time. This is also reflected in recommended values of the Specific Fan Power (SFP) values. For instance according to ASHRAE proposals 2.7 kW/(m^3/s) can be accepted instead of 1.9 for systems <10 m^3/s.
If higher velocities than recommended for CAV-systems are used, precautions must be made about noise.
As a VAV-system is intended to work with varying air flow rates in different parts of the system, special components as VAV terminal devices, pressure controlling dampers etc are used. The control system must also be specially designed.

8 Duct insulation

Depending on the intended function, there are four categories of duct insulation:

- Thermal insulation to decrease heat losses from the duct;
- Condensation insulation to avoid condensation on the duct inside or outside;
- Fire insulation to stop or delay spread of fire or smoke;
- Sound insulation.

The insulation can be located on the inside or the outside of the duct:

- Inside insulation has been used for sound attenuation and to use the duct wall as a vapour barrier;
- Outside insulation has been used for all other applications.

Insulation inside the duct is controversial. Fibres have come loose and contaminated the air or clogged filters (Woods 1997, Lysne et al 1996, Batterman et al 1995, Godish 1995, Nyman et al 1991). It has also been found that when water is present or the relative humidity is high, micro organisms grow in the insulation material (Nyman et al 1991, Morey et al 1991). Inside insulation also increases air friction and makes it difficult to clean the ducts. It is thus recommended that when possible, inside insulation is avoided.
If inside insulation has to be used, it has to be covered with plastic foil or similar on the inside, to protect the air from contamination and make cleaning possible. The cover will decrease sound attenuation.

Duct insulation is typically made of mineral wool, that is glass fibre or rock wool, which is fire resistant. Stone wool is made by diabase, and glass fibre insulation from glass raw material and used glass. Rock wool melts at higher temperature than glass fibre wool. Mineral wool is manufactured with a spinning process. Normal material densities are lower for glass fibre wool than for rock wool. Maximum insulation ability occurs at densities of 50-80 kg/m^2 for glass fibre and 70-120 kg/m^2 for stone wool. Recommended minimum values in Sweden for different qualities are 65 or 100 kg/m^2 (RA VVS 98).

The mineral wool is often delivered as matting, with or without metal net and with or without Aluminium foil, which decreases radiation losses. For rectangular ducts it is common to use panels that are made in different sizes. The mineral wool is fastened to the duct with bands, metal nets that can be sewn together, or similar. For each meter duct length there typically are 3 bands.

Thermal insulation is used because it is needed for the function of the installations (e.g. to keep the air cold enough for cooling of interior spaces), for energy conservation, and for fire protection. The need to conserve energy normally requires thicker insulation than fire protection.

Fire insulation should be on the outside of the duct. When ducts are allowed to pass through walls or other partition, insulation is especially important to protect also against fire breaking through the wall.

9 Fire safety

The ductwork is important for fire safety from the following points of view:

- Fire spread;
- Smoke spread;
- Smoke exhaust;
- Pressurization of escape routes.

The building is normally divided in several “fire cells”, designed not to allow a fire to spread to other cells. A good solution then is to have separate duct systems, one system for each cell. When this not is possible the passage through cell dividing firewalls has to be protected so fire cannot spread. This is achieved by using fireproof materials in the ducts and by tightening with extra fire resistant insulation round the ducts at and close to the passage through the wall, to prevent leakage of hot gases and heat conduction along the duct.
Ducts shall not burn or be so hot that building material, equipment or furniture outside their fire cell ignites. When there is a risk, a safety distance to such materials should be kept and/or sufficient insulation should cover the duct. Note that radiation tends to dominate the heat transfer. A hot gas inside the duct is the most dangerous case. To stop such flow, dampers controlled by fire sensors are installed in the duct system.

Smoke spread is obstructed with sealing as described above and by blocking the ducts with fire dampers. Dampers in supply ducts will close when the air is hot or polluted by smoke. Dampers in exhaust ducts will bring smoke more directly out of the building by by-passing e.g. heat recovery units. Special duct systems for extracting smoke can also be found. The fire damper system is normally controlled with the help of smoke detectors in the ducts and in the building. A literature survey was made by Falk and Malmström (1983).

Evacuation of people out of the building has of course highest priority, especially in high rise buildings. Escape routs without smoke can then be achieved by venting smoke out of the top of e.g. stair shafts. A more advanced method is to pressurize the escape route so air only can leak out and no smoke-polluted air can leak in. This can be achieved with special fan and duct systems or with redirecting air flows in the normal duct system. In both cases it is a problem that the equipment normally is not in use and thus may not be reliable when needed.

9.1 Fire cells and fire resistance

According to the study of Backvik et al (1996), which is based on Swedish rules and traditions, a fire cell should consist of one room or a group of rooms with similar use and with no direct connection to the use of other parts of the building. Fire safe walls should separate fire cells. A typical example of a fire cell is an apartment. Escape routes as stair shafts or hotel corridors are other examples. If it is not feasible to have a separate ventilation system for each fire cell, air ducts have to penetrate the fire safe partitioning walls, and then the same fire resistance demands apply for the ducts; they are not allowed to weaken the resulting fire resisting ability.
The fire resistance of ducts penetrating a fire safe wall depends on three factors:

- The temperature rise on the surface of the insulated duct at or close to the wall (20 mm) to the space with fire. Typical demand is a mean temperature rise of 140 °C around the duct and a maximal temperature increase of 180 °C in any single point. The test method is NT FIRE 005, ISO 834. To protect the duct from dangerous temperature rise, insulation is used. The tightness around the duct is important so fire cannot leak into the adjacent cell. Typically this means that:
    - no flames with lifetimes longer than 10 seconds are allowed, and only small and few flames with shorter lifetime;

- no leakage of gases hot enough to ignite a cotton flock. If the temperature on the protected side is 300 °C or more, the permeability around the duct is tested according to BS 476:part 20:1987. There are many types of sealing available on the market. The sealing should be applied direct to the duct wall and not around thermal insulation (Backvik et al 1996).

- The strength of the duct, so it does not change its form. Due to the heat from the fire the sides of the duct can expand causing deformation of the duct and leakage along the duct sides. If the duct is rectangular and not very small (longest side >0.25 m according to Swedish practice), the duct shall be strengthened at the wall passage by profiles of sheet metal applied to the duct side at both side of the wall. Detailed rules depend on the size of the ducts and type of wall (massive or frame type). The risk for vibration transfer between duct and wall must be observed.
- Duct hangers are also important to prevent ducts to fall down and thus spread fire or smoke, or to prevent ducts from breaking.

These fire resistance measures are regulated in national codes.

9.2 Escape routes

Escape routes have to be protected from smoke. This can be achieved by pressurization, keeping the fire room at a lower and escape routes at a higher pressure than the surrounding building, see BS 5588:Part 4:1978, Code of practice for Fire precautions in the design of building – Smoke control in protected escape routes using pressurization.

Pressurization can be achieved by:

- Using the normal ventilation system with changed flows and flow directions;
- Using special fire pressurization systems.

Backvik et al (1996) recommends the second alternative because the control system of the first is complicated. However, costs and space demands of the first alternative are lower. Pressurisation of a stairwell shaft should have air supplies at every 3-4 floor. Pressurisation has been described in American publications, see for instance Milke (2000).

Shall the ventilation system (a normal one, not an especially fire adopted system) run or be closed during a fire? Jensen (1998) discusses four cases for a mechanical exhaust system:

Case	Extract system	Fire room intact or open to the outdoors	Smoke spread
1	Working	Intact	Small risk
2	Working	Open	No risk
3	Not working	Intact	Big risk
4	Not working	Open	Medium risk

Table 7: Risk for smoke spread for four ventilation cases, air extract systems only (Jensen 1998)

When windows or doors to the outside is open in the fire room no big over pressure can be created. It probably is a good idea to keep the extract system working as long as possible. Hägglund et al (1998) found in full scale tests that with an exhaust system running there was no spread of smoke to adjacent compartments. With a supply-exhaust system running smoke was easily spread to adjacent compartments via the supply ventilation ducts. This was probably related to the increase of pressure in the room with the fire. This increase typically was in the range 100-300 Pa in a closed room (floor area 22 m^2) with only one opening, a circular hole with diameter 0.2 m, and the fire intensity ca 1 MW (Hägglund et al 1996). If the fire is diminished due to lack of oxygen, the pressure in the room drops to a under pressure of about 100-300 Pa. In a tighter room the pressure increase can be bigger, several kPa (Backvik et al 1996).

For a supply and exhaust system, the risk for smoke spread thus is bigger than for an exhaust system. There are lots of technical solutions to adopt ventilation systems for fire with help of dampers etc. Most frequent are to close the ducts around the fire room, but perhaps better is to close supply ducts only, or convert them to exhaust function. Note that in such cases equipment that does not work properly when the flow direction is reversed (like bag filters) cannot be used. The equipment should resist increased air temperatures.

A limiting factor for the height of vertical escape routes like stairwell shafts is the pressure gradient imposed by the temperature difference between indoors and outdoors. For 23 °C higher temperature indoors than outdoors, there is an increase in over pressure indoors with one Pa/m. Especially when the shaft is pressurized, this can give high over pressures in the upper parts, which can create difficulties to open doors. Normally, this creates a demand for sectioning of the shaft, 8 floors for each section is American praxis according to Klote and Milke (1992).

9.3 Fire dampers

There are several different types of fire dampers:

- For protection against fire;
- For protection against spread of smoke;
- For evacuation of smoke.

According to HVCA (1998) fire and smoke dampers are of the following types:

- Folding curtain;
- Single blade;
- Multi-blade.

The dampers shall be able to close against static air conditions when mounted in either the vertical or horizontal planes. Fire dampers shall close at a temperature of around 72 deg C (HVCA 1998).
The dampers should tighten also at high temperatures. A test code is NT FIRE 010. See also BS 476:Part 20:1987.
Common demands are that the damper shall be tight and/or available to open at a pressure difference of 1-5 kPa (VVS-AMA 98).

10 Acoustics

In a ventilation system the three main sources for noise are

- The fan;
- The air flow in the ductwork;
- The air terminal device.

The noise emanating from the fan will be reduced as it passes through the duct system. This dampening of the noise is achieved in many ways. The sound energy transmitted into the duct at the fan will probably be split up into several branch ducts in the same way the air is split up. Sound will also break out through the duct walls. Sound is also dampened at bends .

Noise can be created as well as dampened in the duct system. It is important to keep the air velocities low near ventilated rooms. As the fan noise has been dampened passing through the ductwork secondary noise sources like duct bends or dampers might disturb more. The third main noise source in the ventilation system is the air terminal device. Check data from the manufacturers and chose the best alternative. Several registers in the same room add together logarithmically.

According to Andersson (1998) the sound level L_w in a duct can be estimated as:

$$L_w = 10+50\log v+10\log A \qquad (6)$$

where	v	air velocity	(m/s)
	A	square area of the duct	(m^2)

The mechanisms for sound generation in components differs from that for straight ducts, see Andersson (1998).

According to ASRAE Handbook 1999, duct-related noise problems can be avoided by:

- low air velocities;
- avoiding abrupt changes in duct cross section area;
- providing smooth transitions at duct branches, takeoffs, and bends and locating them no closer to each other than 5 duct diameters, if possible;
- designing the duct system for low pressure drop;
- selecting a fan which can operate near its rated peak efficiency;
- designing the fan inlet and outlet for uniform and straight air flow;
- locating noise creating devices as fan powered mixing boxes, and volume controlling dampers away from noise sensitive areas;
- vibration isolation of fans or other mechanical equipment, and of ducts up to 15 m from the vibration isolated equipment;
- using noise attenuators in supply and return ducts, if necessary.

The paths for the sound transmission via ducts are:

- Airborne through the ducts;
- Breakout through the duct wall.

Round ducts are more rigid than rectangular, which is an advantage for minimizing sound breakout. Often rectangular ducts are made more rigid by using frames, profiled metal sheets etc.

The breakout transmission loss of a duct was defined by Cummings (1985) as (Hassan and Zou, 2001):

$$TL_{out} = 10\log[(W_i / A_i)(A_0 / W_r)] \qquad (7)$$

where	W_i	sound power entering the duct	(W)
	W_r	sound power radiating from duct	(W)
	A_i	cross section area of duct	(m^2)
	A_0	total radiation surface area of duct	(m^2)

The duct-radiated sound pressure level is obtained as (Allen 1960):

$$L_w(out) = L_{w(in)} - TL_{out} + 10\log(A_0 / A_i) \quad (8)$$

Values of TL_{out} (which is a measure of the ducts ability to reduce the sound transmission through the duct wall (breakout) to an adjacent space) are given in ASHRAE Handbook (1999), see Figure 4.
The values for round ducts are higher than those for rectangular ducts at lower frequencies but, at least for bigger ducts, not at high frequencies. For round ducts the values are higher for spiral wound ducts than for long seam ducts, due to the bigger rigidity. Small round ducts have higher breakout transmission loss than big, but such a difference is not evident for rectangular ducts.

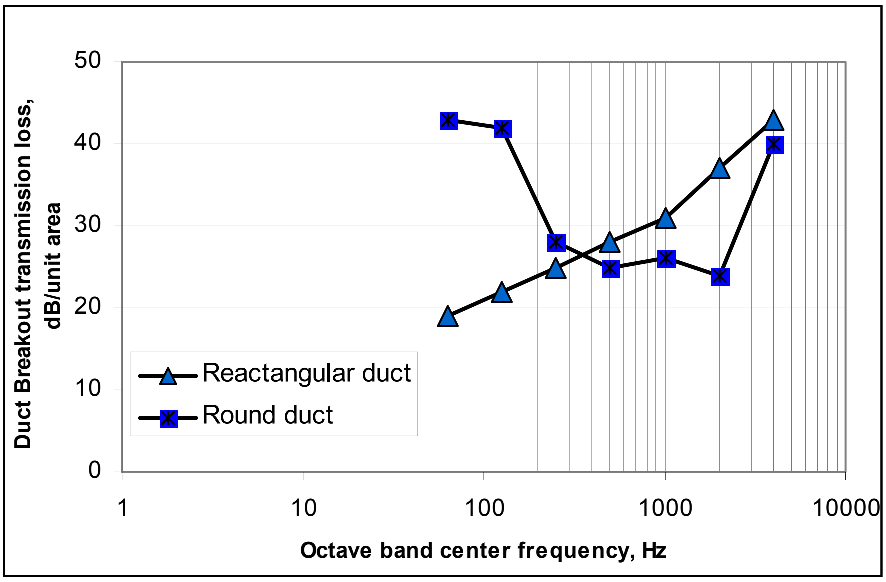

Figure 4 : Duct breakout transmission loss from ASHRAE Handbook (1999), tables 20 and 21, Chapter 46. The rectangular duct is 305 mm by 1220 mm metal sheet thickness 0.8 mm, duct length 6.1 m. The round duct is 0.8 mm spiral wound duct, length 3m, 815 mm in diameter. (Hassan and Zou, 2001)

The data demonstrates that a round duct is much more effective in reducing the low-frequency sound transmission through the duct wall (breakout) to an adjacent space.

Results of Harold (1988) have showed that the rectangular duct does not have good noise reduction properties at low frequencies in contrast to the round duct which has the higher breakout noise reduction that is needed for control of low-frequency rumble noise. The same author conducted also laboratory test where the sound power levels radiated from the rectangular and round ducts are measured, see Figure 5. The round duct has a performance at low frequency that is about 15 dB better than the rectangular one.

It is, however, advantageous to use the rectangular ducts if there is an equipment room enclosing the unit; some of the low-frequency energy is released from the duct into the equipment room before it gets to an occupied room.

Several authors (Scott, 1985; Dean, 1985; Hull 1985; Harold 1988) have indicated certain precautions regarding the transition sections going from rectangular to round in critical areas, which may radiate noise just like a rectangular duct. To overcome this problem, the transitions could be made, if possible, of heavy steel to have good transmission loss properties.

It is worth noting that flexible and rigid fibreglass ducts often comes 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, fibreglass or flexible round duct should not be used (ASHRAE 1999).

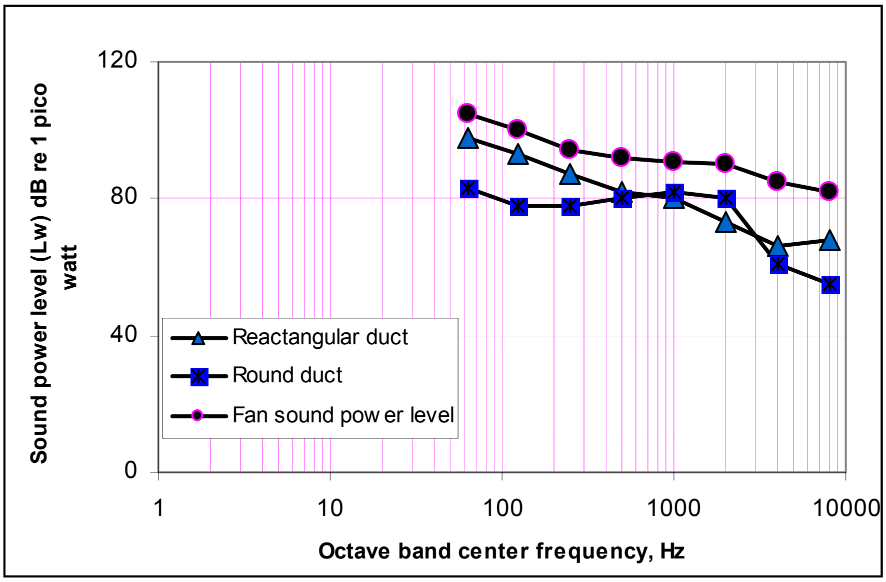

Figure 5 : Sound power levels radiated from rectangular and round ducts (Hassan and Yue, 2001). The fan sound power level is at 23,000 CFM and 3 in of water after Harold 1986. The rectangular duct 29 in by 67 in ,22 ga,19 ft long. The round duct is 37.5 dia.18 ga long seam 20 ft long. After Harold,1986.

One method to reduce the low-frequency noise of rectangular ducts is to use lagging treatment. Although lagging treatment can be more effective in high frequencies, it is less effective in low-frequency noise. "Mass loaded flexible material wrapped over an absorptive layer of material on a round duct (which is less prone to low frequency rumble due to the greater inherent stiffness compared to rectangular ducts) can effectively control noise and may be the only one possible solution due to space limitations." (ASHRAE handbook 1999).

According to ASHRAE (1999) there are three types of silencers:

- Dissipative silencers;
- Reactive silencers;
- Active duct silencers.

The dissipative silencers are most frequently used. They use mineral wool to absorb the sound. The wool is covered by perforated sheet metal. The damping ability is affected by the geometric design of the silencer and the type of material chosen.
Thicker absorption material is needed at low frequencies (125 and 250 Hz). To damp high frequencies (> 500 Hz), thinner absorption material can be used.
Reducing the distance between the clad surfaces increases attenuation. Silencers with small diameters have higher attenuation than silencers with large diameter.

A bibliography "Ventilation and Acoustics" was published by AIVC Limb (1997).

11 Air filters

This chapter is an edited version of a manuscript prepared by Jan Gustavsson.

If the right air filters are installed and regularly replaced, they can help to:

- keep the system clean;
- maintain the efficiency of equipment;
- prevent micro-organisms from entering the system;
- remove outdoor contaminants.

11.1 Classification of air filter

Filters are classified according to their dust spot efficiency or arrestance according to table 8 and under the following test conditions:

- the air flow shall be 0,94 m^3/s (3400 m^3/h) if the manufacturer does not specify any rated air flow rate;
- 250 Pa maximum final pressure drop for Coarse (G) filters;
- 450 Pa maximum final pressure drop for Fine (F) filters.

Filter type	EN 779 Class	Average (A_m) Arrestance (synth. dust) (%)	Average (E_m) Dust Spot Efficiency (%)	Final Pa
Coarse filter	G1	$A_m < 65$	-	250
	G2	$65 \leq A_m < 80$	-	250
	G3	$80 \leq A_m < 90$	-	250
	G4	$90 \leq A_m$	-	250
Fine filter	F5	-	$40 \leq E_m < 60$	450
	F6	-	$60 \leq E_m < 80$	450
	F7	-	$80 \leq E_m < 90$	450
	F8	-	$90 \leq E_m < 95$	450
	F9	-	$95 \leq E_m$	450

Table 8 : Classification of air filters according to EN 779:1993

For fine filters the EN 779 classification is based on a final pressure drop of 450 Pa. However, no HVAC installation is designed for such a high final pressure drop. A life cycle cost analysis will probably show the most economical final pressure drop in the range of 100 Pa to 200 Pa, which is far from the pressure drops used for classification. From an environmental point of view, the best final pressure drop is also much lower than those used for classification. From a hygienic point of view, filters are changed more frequently.

Classification is based on laboratory tests with synthetic dust and does not provide a basis for calculation of the life of air filters or assessment of the filter's performance in actual application. Moreover, the dust-holding capacity and average efficiency for classification vary with final pressure loss and airflow. In filters with electrostatically charged material, the charge is neutralised by the collected dust, resulting in less separation. Influence of electrostatic effect is determined in Nordtest Method VVS 117.

11.2 Lifetime/ Filter Replacement

The lifetime of a filter depends on the concentration and type of dust, air flow rate, and the selected final pressure loss. The filter material and filter construction are often a compromise, or a combination of filter effects and installation space.
Air flow changes in the plant have been the main criteria for changing filters, i.e. when pressure loss increases to the extent that the fan cannot maintain a minimum air flow. Reduction of fan power levels, energy use or economic evaluation, i.e. when the energy cost and filter cost reach a minimum, are becoming increasingly significant.
Ventilation systems often account for a big share of a building's energy use and the pressure drop in air filters perhaps the largest? Thus from the viewpoint of increasing energy efficiency, there are strong reasons for optimising the choice of air filters. A method for LCC calculations is described in Eurovent recommendation REC. 10 "Calculating of Life Cycle Costs for Air Filters".

11.3 Hygiene

Hygiene considerations are being applied more and more to filter replacement. *Möritz (1996, 1998)* have shown that when the average RH is higher than 80 % for three days, there is a risk of microbial growth in the filter and ventilation system. When it is difficult to avoid a high relative humidity in the air intake, filtration should take place in two steps. The first filter can then often be exposed to high humidity or to rain and snow. The second filtration step, using a filter of at least F7 quality (according to *VDI 6022:1998. Hygienic aspects for the planning, design, operation and maintenance of air-conditioned systems*), is not exposed to high RH and stops micro-organisms and particles. This filter can remain in place for about two years, as long as the final pressure loss is not reached within this period.
Organic impurities also become caught in the filters and could be released later. Particles and endotoxins from micro-organisms can loosen in low quality filters.

11.4 Tightness

It is important that the air filters are properly installed and the tightness and condition of the filter must be checked regularly by visual inspections of the installation. No visual leakage or traces of leakage should be accepted.

Eurovent 4/10:1996 - In Situ Fractional Efficiency Determination of General Ventilation Filters describes a method of measuring the performance of general ventilation air-cleaning devices in an installation. This method makes it possible to compare laboratory tests and check the air filter properties in real life. Eurovent 4/10 is a recommendation or guideline when testing in an installation and will cover the measurement of air flow, pressure loss and fractional efficiency.

Acceptable filter bypass leakage is defined in the EN 1886:1998 according to table 9. The norm defines different leakage rates in percentage depending on the filter class.

Filter class	G1-G4	F5	F6	F7	F8	F9
Leakage %	-	6	4	2	1	0.5

Table 9 : Acceptable total leakage, 400 Pa test pressure. EN 1886:1998.

12 Duct hygiene and cleaning

Particles which deposit in the air ducts can change the air flow rates and balance in the system because of increased roughness and decreased effective duct diameter (Wallin 1994). This is normally a bigger problem in exhaust ducts than in supply ducts, where the air is filtered. The deposits can also influence odour levels, as it acts as a bad active carbon filter, adsorbing gases if there is a sudden passage of odorous fumes and then releasing the odour to the ventilation air for quite a long time. This is of special importance if return air is used.
However, perhaps most important is that the particles together with water can cause microbiological growth. This is the most important aspect. It should come first. This is especially important for supply air ducts. The weakest part of the ductwork from this point of view is the air intake, where water can penetrate into the duct and wet filters etc.

12.1 What is a clean duct?

VDI 6022 prescribes that the inner surfaces of ducts should be broom-clean. According to Björkroth et al (2000) (who cites an internal report from Technische Universität Berlin by Fitzner et al), 4 g/m^2 is a reasonable limit-value for broom-clean, when JADCA method with solvent is used. The same source also discusses target values for surface micro-organisms: <15000 for moulds and <30000 for bacteria (CFU/m^2). However, little information is available from the sources as to the rationale of the particular values regarding impact on air quality or occupant perception (Björkroth et al 2000).

Björkroth et al (2000) propose the following target values for cleanliness (dust surface concentration only, reports about micro-organisms are to difficult to interpret):

Method	Detection efficiency (%)	Dust surface concentration g/m^2		
		Low standard	Medium standard	High standard
Total dust	100	20	10	5
Vacuum with blade	90	18	9	4.5
Wiping (JADCA) with solvent	80	16	8	4
Wiping (JADCA)	50	10	5	2.5
Gravimetric tape	35	7	3.5	1.8
Vacuum with brush	15	3	1.5	0.8
Vacuum (Wintest)	10	2	1	0.5
Vacuum (NADCA/HVCA)	2	0.4	0.2	0.1

Table 10 Target values for duct cleanliness

A visual inspection can be used as a first approximation during the dust inspection. When the metal duct surface can no longer be seen through the layer of dust, a value of 20 g/m^2 is likely to have been exceeded.

12.2 Design for clean ducts

12.2.1 Supply air systems

Care has to be taken about the whole duct system, from air intake to supply air terminal device. This is discussed in VDI 6022 and in Nordenadler (2000).
The ducts must be clean when they are new and freshly installed. In order to obtain this, it is important that they are manufactured so they are delivered with no oil residues, Björkroth et al 2000, found that

spiral wound ducts manufactured with processing oil(ϕ 200) had an average density of oil residuals 196 mg/m^2 and high odour emission. Similar spiral-wound ducts manufactured without processing oil had 12 mg/m^2 oil residues and no effect on air quality. The oil also increases the amount of dust that is deposed on the duct walls.
The ducts should be protected from dirt during transport, storage at the construction site, during the construction phase, and during and after the installation. Locked duct ends is one way of achieving this (VVS AMA 1998, Luoma 2000). Lundgren (2000) has pointed out several drawbacks with this method (as nesting of insects and more) and recommends instead cleaning before installation. If locks are used, these should be kept on the duct as long as possible, also after installation when appropriate to protect the ducts from fouling due to air flow caused by wind pressure or thermal forces. Of course the ductwork should not be used until the building has been cleaned and the air system is complete and protected by filters.

12.2.2 Air intake

The *air intake* has to be located so the air is as clean as possible (and also so wind pressure influence is minimized). Thus it should not be close to air exhaust openings. According to VDI 6022 the distance shall be at least 10 m. Of course locations close to other sources of pollution (like chimneys and cooling towers) shall also be avoided. Locations close to roads with traffic, garages, parking lots, and similar can also cause pollution of the air. To avoid the highest concentration of particles from cars, the intake should be located more than 3 m above the road (Peterson 1986). Also Pasanen (1994) has found influence of the height over the ground. For intakes located low (2m above the ground), high concentrations of bacteria was found inside the duct.
A bibliography " Air Intake Positioning to Avoid Contamination of Ventilation Air" was published by AIVC (Limb 1995).
Important for duct hygiene is that water (rain or snow) not is brought in with the intake air. The intake must be big so air velocity is low to achieve this. The grille protecting the intake must thus have a big effective area. An important function of the grille is to protect the intake from birds and other animals. (According to Batterman et al 1995 dirt from nesting birds is a frequent source for bacteria and mould in the duct system.) To achieve this the grille is often supplemented by a net. The grid size should not be too small in order to avoid blocking by leaves etc. In Norway a grid size of 5-12 mm is recommended. It is most important that the grille and net are well maintained. If the intake is located so it is difficult to inspect and clean it, and if there is risk for blocking it with leaves, ice or similar, VVS AMA (1998) recommends that the grill is made easy to open. When the risk for ice on the grille is big, a heated grille may be considered.
Interior lining should be avoided in the intake duct because of the risk for water penetration through the intake. When there is risk for condensation, insulation and vapor barriers as appropriate should be installed on the outside of the duct. The duct between intake and air handling unit should be as short as possible (VDI 6022). There should be possibilities for draining and cleaning. The drain should not be directly connected to the sewage system (because of the danger of ejecting polluted air). There should be an inspection opening.

The intake could lead to an intake space or chamber, originally used for dust settling but also as nod for ducts to different parts of the building. In this case there are possibilities to clean the space in conventional ways. It should be made with even surfaces, easy to clean, and the floor inclined towards the drainage opening. If possible heat should be added when needed to the outdoor air in order to keep the space dry.

12.2.3 air handling unit

The *air handling unit* (AHU) could consist of the following parts:

- Outdoor air damper;
- Filter;
- Exhaust air heat exchanger or if return air is used, a mixing box;
- Heater;
- Cooler;

- Humidifier;
- Fan;
- Sound absorber.

According to VVS AMA 98, the AHU shall be made so it is possible to clean with water, and consequently should be equipped with drainage in low points. Joints, corners etc, should be made so dirt not accumulates.

12.2.4 outdoor air damper

The *outdoor air damper* can easily be fouled which deteriorates the function, for instance cause leakage because the damper cannot be completely closed. Fouling can be decreased if high velocities in the damper are avoided. It should be designed so it is easy to inspect and clean. It is important from energy point of view that it is tight. When the system not is in use the damper should also stop tendencies to backward flow in the supply duct, that could cause fouling of the duct as it may not be protected by filter. The outdoor air damper often is located close to the filter in the AHU. It is then important that the damper has the same area as the filter, otherwise the full filter area will not be used.

12.2.5 filter

The *filter* is most important for duct hygiene. Filters have high pressure drops which causes much fan energy use. This has always to be considered. Here, cited recommendations are based on hygiene considerations.
Filters should be of at least class F7 for good indoor air quality.

Outdoor Air Quality	Indoor Air Quality		
	IDA 1 Excellent	IDA 2 Average	IDA3 low
ODA 1 (pure air)	F8	F7	F6
ODA 2 (dust)	G4/F8	G3/F7	G3/F6
ODA 3 (gases)	G4/F8	F7	G3/F6
ODA 4 (dust + gases)	G4/F8	G3/F7	G3/F6
ODA 5 (very high conc.)	G4/GF*)/F9	G4/GF*)/F8	G3/F6

Table 11 : Recommendations according to prEN 13779, earlier version. In the version dated October 2001 there are 4 classes instead of 3, and the filter recommendations are omitted.

VDI 6022 recommends two filter steps for the supply air if there is risk for water penetration or condensation. The first step should be at least class F5 and the second at least class F7. The filter should have a large area and small air velocities. It should have a pressure-drop measuring device to check the degree of fouling of the filter. Big pressure drop indicates time for filter change. Fouled filters steal fan energy, emit dust during start-up of the system, and increase the risk for filter damage. They can also according to several research reports cause smell (Nordenadler 1999).
The space around the AHU should allow easy filter changes.
It is important that the filter not allows any unfiltered air to pass it. This could happen if the filter is damaged or if air leaks around the filter frame. Sealing is necessary, see table 9.

It is most important that water not penetrates to the filter. The filters should also be protected from high relative air humidity. According to VDI 6022 they should not be exposed to more than 90% RH and, as a mean for three days, not more than 80% in the first filter step.

12.2.6 Heat recovery

Heat exchangers for heat recovery are rotating (regenerative), or recuperative (direct plate exchangers between exhaust and supply air or, when supply and exhaust ducts cannot be located beside each other, conventional finned heat exchangers connected with piping.
According to Nordenadler (1999), beside fouling leaking of return air to the supply side is a frequent problem. This is particularly so with rotating exchangers, but happens also with plate heat exchangers, especially if they have been exposed to frost (Lysne 1996). To avoid such leakage, pressure should be somewhat higher on the supply side than on the exhaust side (Carlsson et al 1995). This is said to be especially important for rotating exchangers, but too big pressure difference can also cause the wheel to slant.
VDI 6022 and VVS AMA1998 states that the recovery system should be easy to inspect and clean and also possible to disinfect. Filters (at least class F5) should protect the equipment also on the exhaust side if the plate distance is less than 6 mm (VDI 6022). Distances less than 2 mm should not be used.
Nyman et al (1991) writes that the exhaust air can foul rotating heat exchangers with particles that then enters the supply air when the air flow direction changes, if not the wheel is protected by filters also on the exhaust air side.
It is important that condensed water can be taken care of, in the warmer of the airflows. Need for defrosting must also be analysed.

12.2.7 Air heaters and air coolers

As already have been discussed for heat recovery systems, heaters and coolers should be protected from dirt. Air coolers often operate below the dew point of the air. Drainage must be provided, with no risk for overflow of polluted air from the sewage system. Coolers should be provided with a drip-plate below the battery, and .a droplet separator downstream. Coolers should not have filters or silencers directly downstream. The equipment should be easy to clean.
To avoid humidity downstream of the cooler, this shall be shut off before the other parts of the AHU (VDI 6022).

12.2.8 Humidifiers

Scheduled cleaning of the humidifier is important to avoid microbial growth. Material like plastic or stainless steel, not promoting microbial growth should be preferred.
The humidifier should be designed so it is easy to inspect, clean, and disinfect. It should be provided with a drip-plate, drainage, and a droplet separator.
The humidifier shall be controlled so the RH in the system, especially the filter, does not exceed 90%.
To avoid humidity downstream of the humidifier, this shall be shut off before the other parts of the AHU (VDI 6022).

12.2.9 Fans

The fan should be possible to inspect and clean. Especially belt driven fans can emit particles. Big belts are better than small in this aspect. Belt driven fans should be located between the first and second filter. The fan should have a smooth start to avoid emission of particles from the rope to the supply air.
When a high quality filter is used as the second stage it should be located downstream of the fan to avoid risk for leakage into the duct of polluted air.

12.2.10 Silencers

Like all other parts silencers shall be available for inspection and cleaning. Porous sound absorbing materials like mineral wool should be covered and not in direct contact with the air. Silencers may contain parts which obstructs cleaning. In such cases it may be necessary to remove the silencer while cleaning.

12.2.11 Supply air ducts

The ductwork should be possible to clean, but too many inspection openings should be avoided to minimize cost and leakage. They should be inspected at regular intervals.
If there is a risk for condensation at the ducts outside or inside, they should be insulated. Inside insulation should be avoided, especially if there is any risk for water, through penetration or condensation.

12.2.12 Extract air ducts

If return air is used, it is important for air quality that the extract air ducts are clean. Equipment in AHUs should also be protected from fouling. It is also known that severe fouling, resulting in decreased air flows and system unbalancing, can occur in such systems (Wallin 1994). Such ductwork thus should be protected with filters, and be easy to clean.

12.3 Duct cleaning

This chapter is an edited version of a manuscript prepared by Thomas Kuehn.

In the past, duct cleaning was performed because the build up of dirt in the duct was so great as to significantly reduce the amount of air flowing through it. More recently, it is recognized that air handling systems within buildings can be sources of indoor airborne contaminants. Thus duct cleaning may be required at much lower levels of contamination. The most troublesome contaminants are gaseous emissions from bacterial growth and the microbial spores themselves. Collection and growth of microorganisms can occur on filters, cooling coils, in drain pans, and on the interior surfaces of the air handler and the supply and return air ducts and plenums. Routine maintenance such as filter replacement and coil and drain pan washing can control much of the deposited contaminants. However, cleaning the ductwork is nearly always performed only when a problem is perceived and is therefore non routine.

An annotated bibliography *"Ventilation air duct cleaning, Office buildings"* was recently published by the AIVC (Limb 2000).The majority of the literature reviewed focused on cleaning exhaust ducts. The authors felt that the scientific basis for many of the guidelines that exist needs more research. The results of this bibliography include the cleanliness of ventilation ductwork, typical contaminants, system design issues and methods of cleaning and control. Maintenance programs and standards are also reviewed.

A *review of the published literature on duct and air handling unit cleaning was recently performed at the University of Minnesota* under contract from ASHRAE. The study included a review of duct cleaning methods, associated health effects to workers and building occupants, and the effectiveness of the cleaning protocols. The study also included a survey of duct cleaning companies and interviews with professionals in the U.S. who have experience with duct cleaning as a means to alleviate indoor air quality problems. The results have been published as two Technical Papers in the ASHRAE Transactions (ASHRAE Transactions, Vol. 106, Pt. 1, 2000). The following discussion briefly summarizes the findings of this study.

12.3.1 Duct Cleaning Methods

Methods used for cleaning include dry cleaning, wet cleaning, disinfection, encapsulation and duct lining removal. Dry cleaning is performed when the contaminants can be removed by simple mechanical means or the use of water is not practical. Hand washing is performed when access is easy or when the duct is large enough to allow personnel to move inside the duct. Smaller ductwork requires the use of rotary brushes and spray wands. Decontamination is performed using a variety of chemicals that kill or control the growth rate of microorganisms. Encapsulation is used to contain loose fibrous insulation and the incorporated nutrient and organic materials. Removal of duct lining material is usually the preferred method of cleaning when it is possible to do so. The usual cleaning procedure is to isolate a section of ductwork and provide a negative pressure using a vacuum cleaner at one end. Cleaning proceeds from the other end of the section towards the end with the vacuum. Various optical devices are used to observe the progress of the cleaning inside the ductwork.

12.3.2 Health Effects

Personnel in the building are usually well protected during the cleaning if the section being cleaned is isolated from the general air handling system and a HEPA filtered vacuum cleaner is utilized. The use of decontaminants and encapsulating agents is more problematic. The chemicals used should be approved for such application. In the U.S. this approval is regulated by the Environmental Protection Agency. Workers should have personal respiratory protection and should wear clothing suitable for their work. Most workers wear disposable facemask filters, gloves and washable clothing.

12.3.3 Effectiveness

The long term effectiveness of duct cleaning is not well documented. Methods to evaluate duct cleanliness are not well developed and range from simple hand wiping of a small surface area to the use of contact microbial growth plates. These diagnostic evaluation methods only apply to a small section of duct, often near an access panel where the cleaning has been very effective. They do not show areas that have been poorly cleaned or have not been cleaned at all. Better diagnostic evaluation methods should be developed and more data on the soiling rate should be obtained. In some cases, ducts required cleaning within a year of the previous cleaning.

12.3.4 Survey of Duct Cleaning Companies

Many companies have been in the duct cleaning business for many years. The main area of business for most of them is commercial buildings although many also perform duct cleaning in residences. The vast majority of their work came from building owners and operators who were experiencing indoor air quality problems and suspected that the air handling system was contributing to the problem. However, when asked whether duct cleaning was the best remedy for indoor air quality problems, many companies indicated that it was not.

12.3.5 Survey of IAQ Professionals

Consensus among the professionals was that prevention was the best approach to the problem of cleaning ductwork and air handling units. When cleaning was necessary, they recommended a thorough physical cleaning. They generally did not favour the use of disinfectants as they were not convinced that they were effective over a long time span. The professionals recommended removal of duct lining materials when practical rather than encapsulating it because they had observed breakdown of the encapsulating materials over time.

12.3.6 Recommendations

It is recommended that the best approach is to reduce or eliminate the necessity to clean the air handling system through the use of locks to protect the ducts until actual use and then good filtration. The supply ducts are usually well protected through the use of good filters. However, the return ducts and return air plenums, particularly above suspended ceilings, are not protected by filters and are the most heavily contaminated. If some of the return air is recirculated, the contaminants in the return can be transported back into the building. The use of fibrous duct lining should be discouraged. The fibres can become dislodged which posed as health hazard and the lining provides a very good reservoir in which microbes can grow. If used, special materials with long fibres should be used. They are very difficult to clean and decontaminate. Encapsulation may be effective for some time but this surface is difficult to clean in the future.

Some duct units have parts which more or less block the duct system, and thus obstruct or prevent cleaning of it. Such units are silencers with baffles, most dampers and some flow measurement units.
To permit cleaning of such units one can:

- install cleaning covers on each side of the unit;
- use a sealing clamp or similar to make it easy to remove the unit from the system.

13 Duct tightness

See Ductsave Report (Carrié et al 1999) and Carrie et al (2000).

In Europe, national ductwork airtightness standards are in general similar to EUROVENT 2/2 guidelines that propose a one-point measurement of the leakage flow rate at a given pressure differential. The installations are classified according to the value of the leakage factor at a reference test pressure of the system based on the following equation:

$$\frac{Q}{A} = f_{ref} = K \Delta p_{ref}^{\ 0.65} \qquad (9)$$

Where	f_{ref}	leakage factor at Δp_{ref}	($m^3\ s^{-1}\ m^{-2}$)
	Δp_{ref}	reference pressure differential across the leaks	(Pa)
	K	leakage coefficient per m^2 of duct surface area	($m^3\ s^{-1}\ m^{-2}\ Pa^{-0.65}$)

The maximum leakage rates for three airtightness classes (A, B, C) are based on maximum values of *K*. Class D is sometimes used based on the same geometric progression, i.e. $K_D = 0.001 \cdot 10^{-3}\ m^3\ s^{-1}\ m^{-2}\ Pa^{-0.65}$. According to EUROVENT 2/2, Δp_{ref} is set to the mean operating pressure of the duct system. It is noteworthy that this classification relies on an arbitrary flow exponent of 0.65 which according to DW/143 (HVCA, 1983) is justified by Swedish tests performed on a variety of constructions. However, measurements performed in the SAVE-DUCT project show a broad range of values.

Class A	K_A =	$0.027 \cdot 10^{-3}$ $m^3\ s^{-1}\ m^{-2}\ Pa^{-0.65}$
Class B	K_B =	$0.009 \cdot 10^{-3}$ $m^3\ s^{-1}\ m^{-2}\ Pa^{-0.65}$
Class C	K_C =	$0.003 \cdot 10^{-3}$ $m^3\ s^{-1}\ m^{-2}\ Pa^{-0.65}$

Table 12 : Airtightness classes defined in the EUROVENT Guidelines 2/2. For laboratory duct testing, these values are divided by 2.

The influence of air leakage on design and "optimisation" of duct systems have been studied by Bouwman (1982) and Tsal et al (1998). Bouwman have shown that the change in distribution cost is small if the ductwork is "optimised" with regard also to the leakage.

Tsal et al point out that the distribution energy demand to compensate for leaking in an existing duct system does not vary as the cube of the flow rate ratio. One reason is that the tight and the leaking systems do not have the same system resistance curve, but the leaking system has a "lower" curve. With no change of fan rotation speed, that is the same "fan curve", this will normally result in a larger airflow rate through the fan, which increase partly compensates for the leakage. The actual increase in fan power is thus less than as calculated with the "cube rule".

In addition to fan energy losses, energy losses due to leakage of conditioned Warm/cold air are important and should be avoided by making the duct system tight. Carrié et al have shown that this in fact is the case in the Scandinavian countries. Field tests showed that the duct systems in many cases fulfilled class D.

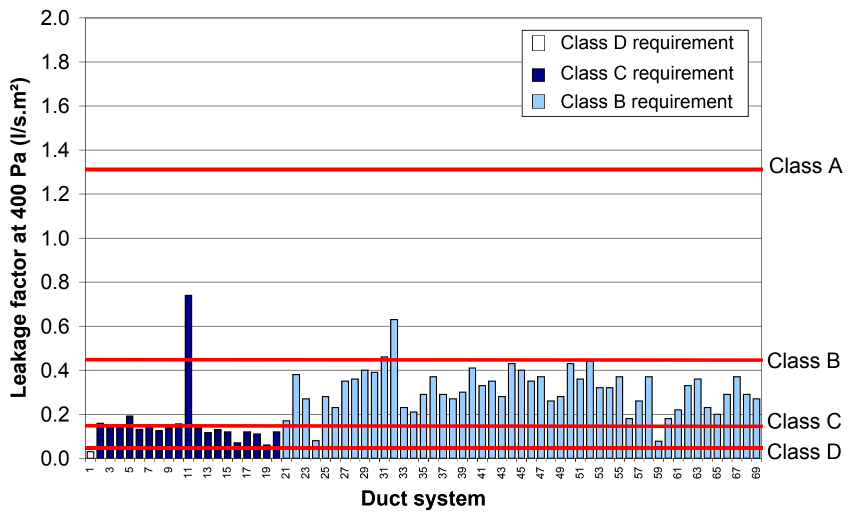

Figure 6 : Leakage factor at 400 Pa for 69 duct systems in Sweden. Leak-tests performed at commissioning (Carrié et al 1999).

In Carrié et al (1999), the following five site sealing methods are presented. Detailed information about these methods can be also found in SMACNA (1995):

- Gaskets;
- Tapes;
- Sealing compound;
- Internal duct lining;
- Aerosol-sealant.

All the sealant products have a relative short shelf life; often it is one year or less. SMACNA (1995) also point out that many sealants contain volatile solvents which could be hazardous to the building occupants. It must be recognized that no sealant system can be a substitute for mechanical attachment.

14 Duct construction materials

Duct construction is described in handbooks as ASHRAE Handbook Systems and Equipment (2000), branch publications as DW/144 Specification for sheet metal ductwork HVCA (1998), HVAC Duct Construction Standards (SMACNA 1995), and manufacturers catalogues.
They can be made of building material, which is not uncommon for large ducts close to the air intake, or even insulation material in cases where low first cost is essential. Ducts can also be made of fabric, especially when the leakage through the porous fabric is used for air supply. Most frequently used are sheet metal ducts, which are the object of this report.
They are typically made from hot dip galvanized sheet steel with a yield point in tension of 200 N/mm^2 (minimum 150 N/mm^2) and with a zinc coating of 275 g $zinc/m^2$ (Z 275) (VVS AMA 98).

Examples of materials are (LINDAB 2000):
- Ducts and hand made fittings EN 10142 – Fe PO2 G Z 275 MA–C;
- Pressed fittings EN 10142 – Fe PO2 G Z 275 MA–C and EN 10142 – Fe PO6 G Z 275 MB–C.

The application of zinc shall be made after the metal part has achieved its final form. The zinc could also be applied through other processes

The zinc layer will be corroded at a rate dependent on the surroundings. The following values (µm/year) are typical for Sweden (RA VVS 1998):

	Location	*u*m/year
Air	Indoor	<0.5
	Mountain	<1
	Rural area, far from the coast	0.5-1.5
	Small towns, far from the coast	0.75-1.5
	Big towns, far from the coast	1-3
	Big towns, at the coast	2-5
	Industrial area	2-10
Water	The North sea	12-46
	the Baltic	10
	Hard lake water	2-4
	Soft lake water	<20
	Drinking water	<15
	Distilled water (15C)*	50-200
Soil		4-500

*The same values can sometimes also be applicable for condensed water.

Table 13 : Corrosion rate for zinc coating

Other material used for metal ducts is for example:
- Stainless steel EN 1.4436 el AISI 316 or EN 1.4301 el AISI 304;
- Aluminum ISO/DIS 209-1;
- Plastic coated products;
- *Ducts*: i.e. hot dip galvanized sheet steel Z 275, with an internal and external coating,100 µm thick, of polyvinyl chloride (PVC);
- *Fittings* hot dip galvanized sheet steel Z 275 and then powder coated internally and externally with a mixed powder consisting of epoxy and polyester (PE) to a thickness of 80 µm. Powder coating can be optionally obtained in thick-nesses up to 200 µm.

Duct materials are discussed in Stratton (2000).

15 Strength

15.1 Circular ducts

Requirements and testing of strength for circular ducts are described in the Standard Proposal prEN 12237 (Oct. 2000).
Ducts of Ø250 mm and above are normally given stiffening corrugations to increase radial stiffness.
What happens when a circular, spirally-wound-duct is exposed to big pressure differences is described in Lindab, 2000:

- Positive pressure: In case of high positive pressure, the seal moulding lips will first start to whistle. At considerably higher pressure, the joints between the ducts will be forced apart. The high pressures needed for this to happen are not relevant to ventilation installations.
- Negative pressure: In installations with high negative pressure, there is a risk that the ducts could collapse. This phenomenon is referred to as buckling, and can suddenly happen at the weakest point in the system. Buckling wanders along the duct, which can be completely flattened. The weakest point is frequently a "transport dent" on a duct.

The critical under pressure decreases rapidly with increasing duct diameter: For D=100 mm it is about 30 kPa, for D=315 about 10 kPa and for D=1000 about 1 kPa. (Lindab and ABB 2000). It depends on the thickness of the sheet metal and the stiffening profilation on the duct. For this reason, use only undamaged ducts, for which under pressures less than ca 1 kPa normally cause no problem. For big duct sizes this is an important limiting factor.

Straight-seamed round ducts have more need of special stiffening arrangements than the spirally-wounded (HVCA 1998).

15.2 Rectangular ducts

Rectangular ducts are more sensitive than circular ducts for both over and under pressures. A rule of thumb is that the maximum sheet metal area without an outside stiffening profile is 1 m^2. Stiffeners are often frames of profiled steel, around the duct or attached to the longer side.
HVCA (1998) gives rules for stiffening arrangements for three different classes of strength: low pressure (limited to 500 Pa positive and 500 Pa negative), medium pressure (limited to 1 kPa positive and 750 Pa negative) and high pressure (limited to 2 kPa positive and 750 Pa negative).
For ducts with widths over 1.2 m, tie rods should be applied in order to limit the reinforcement size (SMACNA 1995).
Rectangular ducts are manufactured in standard sizes, see catalogues, but are often "tailor made". The complexities of the construction make the workmanship extremely important, especially for ducts with large dimension and aspect ratio.

15.3 Flat oval ducts

Flat-oval ducts are formed by spirally-wound circular ducts, using a special former. They are used when available space not permit the use of circular ducts. Larger ducts (area/meter duct >1.2 m^2) are stiffened by swages (HVCA 1998).

15.4 Strength evaluation

The strength analysis of a *rectangular* duct represents a difficult problem. Theoretically, the solution should be based on a three-dimensional structural analysis. Unfortunately, no closed form solution can be found to permit this kind of analysis. In practice, following different methods are employed:

- Linear elastic analysis: Information and calculation methods can be found in SMACNA(1980).

- Large deflection analysis: This method is used to calculate the panel collapse load. Information and calculation methods can be found in ASME AG-1.
- Finite element analysis: In Wu et al (1998), these three methods have been applied on the calculation of the structural integrity of a 2.9 m ×1 m rectangular HVAC duct with angle stiffeners spaced 1 m apart. The calculation results show that, compared with the finite element analysis, closed form solutions based on elastic analysis and the large deflection method are often too conservative to enable verification of structural adequacy.

16 Hangers and supports

Hangers and supports are described in HVCA (1998).

There are many hanger alternatives for rigid metal ducts. It is best to follow the manufacturer's instructions to install them, but in their absence, typical hanger constructions for metal ducts and flexible ducts can be found in SMACNA standard (1995), for fibrous glass duct, NAIMA standard (1998).

Normally, a duct hanging system consists of the following three parts:

1. Upper attachment, which is the connection between the hanger and the building. A safety factor between 4 and 5 is generally necessary for the installation. According to SMACNA (1995), 4 upper attachment methods are applied in ductworks:
    - Concrete inserts, which are mainly used in the simple duct layout and installed before the concrete is poured;
    - Concrete fasteners, which are installed after the concrete is poured so duct location allows great flexibility;
    - Structural steel fasteners, which are installed with beam clamps or powder-actuated fasteners;
    - Integral hanging system in cellular metal deck offered by manufacturers.
2. hangers, which normally are made of round steel rods or strips of galvanized steel
3. Lower attachment to the duct section.

The support shall be strong enough to carry the weight of one person standing on the duct (VVS AMA 98). The loads are specified in prEN 1507 and prEN 12237.

Hangers and support shall be made so they do not transmit vibrations and sound, and according to fire regulations.

17 Fundamentals of duct flow aerodynamics

For ventilation and air conditioning applications, laminar airflow in ducts almost never occurs. The flow is turbulent, but seldom fully developed turbulent "pipe flow". For this, long straight ducts with no flow disturbances are needed.
In turbulent duct flow, there is an exchange of fluid between the fast moving fluid in the central part of the duct and the slow moving air close to the duct walls. The air entering the slow region looses most of its kinetic energy due to the internal friction, while the slow moving fluid entering the fast region has to be accelerated. This process causes a pressure drop along the duct.
For flow in an air duct between two sections 1 and 2 (air density is taken as constant, which is standard procedure for ventilation duct flow), Bernoulli's law gives:

$$p_1 + \rho \frac{v_{m1}^2}{2} = p_2 + \rho \frac{v_{m2}^2}{2} + \Delta p_{12} \qquad (10)$$

Where	p	static pressure	(N/m^2)
	ρ	fluid density	(kg/m^3)
	Δp_{12}	pressure loss between section 1 and 2	(N/m^2)

v_m are mean velocities, defined as the ratio of volumetric fluid flow rate q_V and duct flow area. For a circular duct:

$$v_m = \frac{q_V}{\pi R^2} \qquad (11)$$

Where R duct radius (m)

For a part of a straight duct of constant diameter D and length L, the pressure loss is due to friction (λ_f: is the friction factor):

$$\Delta p_{12} = \lambda_f \cdot L/D \cdot \rho/2 \cdot v_m^2 \qquad (12)$$

The friction factor λ_f ·is a function of Reynolds number (the product of mean duct air velocity, duct inner diameter, and the inverse value of the kinetic viscosity of the air) and the duct roughness. If these values are known, the friction factor can be found in a Moody Chart, which is found in most handbooks. A corresponding relation, which allows the friction factor to be obtained directly and not iteratively as the original Colebrook-White equation is given by Miller (1978):

$$\lambda_f = \frac{0.25}{\left[\log\left(\frac{k}{3.7D} + \frac{5.74}{\mathrm{Re}^{0.9}}\right)\right]^2} \qquad (13)$$

where	k/D	the roughness factor
	Re	Reynold's number

(NOTE: In reality, air in ducts of course is compressible. If there is no heat exchange with the surroundings, the flow is isenthalpic and associated with no temperature change. The ratio p/ρ is constant. Air at these normal pressures behaves as an ideal gas, so $p/\rho = RT$, where R is the gas constant and T the absolute temperature of the gas. The pressure decrease caused by the flow is compensated by the expansion of the gas. The temperature drop of the air normally associated with an expansion process is exactly compensated by the temperature increase due to internal friction in the gas caused by the flow.
The pressure losses occurring in the duct airflow are compensated by the fan. In this compression process, the temperature of the air is increased.
As earlier mentioned, for engineering purposes, the flow is looked upon as incompressible, in order to have analogy with water flow. In incompressible flow, the "losses" are directly

transferred to the fluid as heat. Although the total energy of the flow is constant, it is thus meaningful to talk about "energy losses", meaning the flow energy transferred to heat in the fluid because of internal friction.)

If the duct is rectangular, the analysis of flow pattern becomes more difficult than that of the round duct. Due to the lack of analytical description, the predictions of turbulent pressure drops in noncircular ducts are mainly based on empiricism and momentum analysis. Nikuradse (1930) was one of the first who suggested to utilize the hydraulic diameter in predicting turbulent pressure drop along noncircular duct. He found that it could be used to correlate pressure loss data for ducts of different shape. Ducts with the same hydraulic diameter but with different cross-sectional forms thus will tend to have similar pressure drop for the same duct length and air mean velocity. The hydraulic diameter is defined by:

$$D_h = 4A/P \tag{14}$$

where A = duct square area (m^2)
P = duct perimeter (m)

For ventilation application, the circular equivalents' relationships for rectangular ducts based on flow capacities could be more convenient than the "hydraulic diameter" based on mean velocities of flow. 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:

$$D_e = \frac{1.30(ab)^{0.625}}{(a+b)^{0.25}} \tag{15}$$

Equation (15) is used in ASHRAE handbook (1997). However, the following equation is recommended by the *European Committee for Standardization*, see CEN 1505:1997:

$$D_e = 2b(\pi^{2-n}(1+a/b)^{1+n}/(a/b)^3)^{1/(n-5)} \tag{16}$$

where

$$n = 1/(1.05\log \mathrm{Re} - 0.45) \tag{17}$$

Equation (16) implies that the equivalent diameter depends not only on the rectangular duct dimensions but also on the Reynolds number of the airflow. In the following section, a comparison between these two models is presented (Hassan and Yue, 2002).

In term of aspect ratio *a/b*, equation (15) can be rewritten as:

$$D_e = \frac{1.3b(a/b)^{0.625}}{(1+a/b)^{0.25}} \tag{15a}$$

The ratio R_{de} of D_e values calculated from equation (16) to those according to equation (15a) is:

$$R_{de} = \frac{2(\pi^{2-n}(1+a/b)^{1+n}/(a/b)^3)^{1/(n-5)}}{1.3(a/b)^{0.625}/(1+a/b)^{0.25}} \tag{18}$$

In Figure 7, the ratio R_{de} is plotted in terms of rectangular duct aspect ratio and Reynolds number. These two methods can get the same D_e at Reynolds number about 200,000 for all aspect ratios. When the Reynolds number is less than 200,000, the D_e values calculated with equation (15) are higher than the values calculated with equation (16). When the Reynolds number is higher than

200,000, the D_e values calculated with equation (15) are lower than the values calculated with equation (16). The deviations of these two methods increase with the respect ratio. However, for normal ventilation duct aspect ratios (*a/b*<4), the deviations is less than one percent.

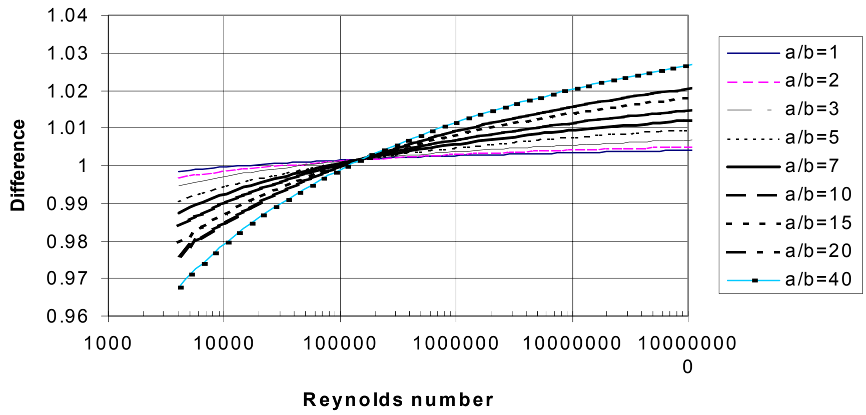

Figure 7 : Comparison of calculated D_e values evaluated from equation (6) and (5)

Beside friction, pressure losses also occur during flow in bends, T-junctions and other components This is illustrated by .

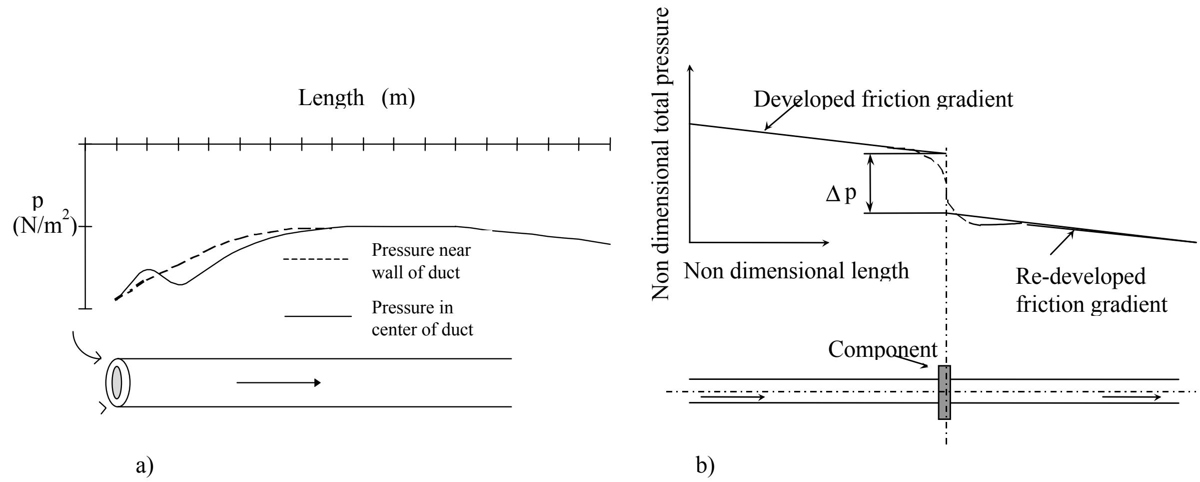

Figure 8 : Pressure gradients a) Static pressure measured with a pitot static probe in the centre of a duct, downstream of an exhaust terminal device. b) Definition of component pressure drop. After Miller (1978).

As is shown in Figure 8 b, the loss coefficient is defined based on the extrapolated pressure difference in the plane of the component. This is to allow for the customary calculation of duct friction losses, based on the total duct length.

$$\Delta p_{12} = \zeta_{12} \cdot \rho \frac{v_{m1}^2}{2} \quad (19)$$

In Figure 8b the pressure difference Δp could be interpreted as a difference in static pressure. However, please note that the kinetic energy before and after the component are equal (the friction gradient is the same as are duct diameter and flow rate. The loss coefficient is based on the difference in total pressure, that is the sum of static and dynamic pressure.

$$\Delta p_{12} = (p_1 + \rho \cdot v_{m1}^2/2) - (p_2 + \rho \cdot v_{m2}^2/2) \quad (20)$$

Of special interest is the case with a sudden area increase of the duct resulting in a velocity decrease and increase in static pressure. The net effect is a ”total pressure” loss (the loss of dynamic pressure is bigger than the increase of static pressure). A corresponding effect can be achieved in the main duct after a T-junction, where air has been extracted. If the diameter of the main duct is not reduced, the velocity will decrease and dynamic pressure will transform to static pressure. In this way static pressure can be kept more constant along the duct, which makes flow balancing much easier. The corresponding duct sizing method (“static regain”) has the disadvantage of high air velocities in part of the duct system.

Pressure loss coefficients for T-junctions need special attention as the flow rate changes. A short discussion is given below as an example (Eriksson and Malmstrom, 2000). See Figure 9.

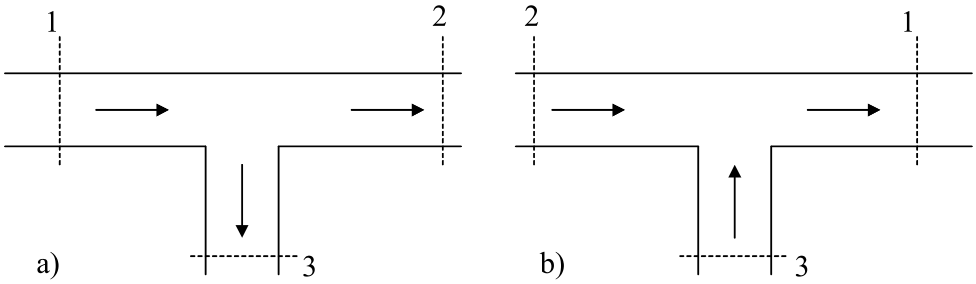

Figure 9 : Flow dividing and combining T-junctions.

The loss coefficient ζ_{12} for the flow from leg 1 to leg 2, figure 9a, is defined by:

$$p_1 + \rho \frac{v_{m1}^2}{2} = p_2 + \rho \frac{v_{m2}^2}{2} + \zeta_{12} \cdot \rho \frac{v_{m1}^2}{2} \tag{21}$$

$$\Delta p_{12} = \zeta_{12} \cdot \rho \frac{v_{m1}^2}{2} \tag{22}$$

The loss coefficient is always based on the mean velocity in the leg of the T-junction with the total flow, leg 1. For pressure loss coefficients, see references below. For further discussion of pressure losses in T junctions, see Holl et al (1951), Gilman (1955), Ashley et al (1956), Gardel (1956). For pressure losses in long ducts with evenly distributed branches, see Haerter (1963)

Very approximate values for the pressure losses, given only to indicate the size of order, are:

- T-junctions, dividing.. *All ducts have the same diameter*.

  Δp_{12} is about 0

  Δp_{12} Δp_{13} is about ($\rho \frac{v_1^2}{2}$ +0.5 $\rho \frac{v_3^2}{2}$)

- T-junctions, combining. *All ducts have the same diameter*.

  Δp_{21} is for $v_3/v_1 < 0.6$ about $\rho \frac{v_1 \; v_3}{2}$

  Δp_{21} is for $v_3/v_1 > 0.6$ about 0.6 $\rho \frac{v_1^2}{2}$

  Δp_{31} is for $v_3/v_1 < 0.3$ about $\frac{1}{0.3} \rho \frac{v_1 \; v_3}{2}$ - $\rho \frac{v_1^2}{2}$

Δp_{31} is for $v_3/v_1 > 0.3$ about 0.6 ($\rho \frac{v_1^{\ 2}}{2}$ -2.2 $\rho \frac{v_2^{\ 2}}{2}$ + $\rho \frac{v_3^{\ 2}}{2}$)

Other duct parts as bends, area changes, and other duct components also create flow energy ("pressure") losses. All the different losses, calculated with values obtained from laboratory experiments, are summarized when the energy demand for a duct system is calculated. The real demand often is different than this sum. One reason is the flow interaction of the different duct parts, sometimes called the "system effect". Normally this effect means bigger energy need, because the influences are disturbances. Sometimes designers add 5% to the fan pressure in order to compensate for this effect. It is thus important to make an aerodynamically good design, which beside the decreased energy cost also will give less noise. This can be made by avoiding abrupt area changes, use vanes in bends with big flow rates, and similar. A rule of thumb is to have a distance of 10D (duct diameters) or more between components, if possible, and no smaller than 6D. Vanes in bends are especially useful in rectangular bends. In Sweden VVS AMA (1998) requires a vane in bends in rectangular ducts 800 mm or higher, located at a distance of 1/3 of the height from the inner side of the bend, which has to be rounded with an inner radius of 100 mm. It is important that the vane is securely fix in the duct, as vibrations create a most disturbing noise.

Of special importance is the connection between the fan and the ductwork, as the outlet velocity of the fan often is the highest velocity in the duct system, and seldom even. The connection must be designed to create an even velocity profile with no swirl in order to minimize the losses. The fan connection should also be made as soft studs, in order not to transmit vibrations to the ductwork, which is a complication.

For further discussion of and numerical values for pressure loss coefficients, see handbooks like ASHRAE Handbook, Eurovent (1996), Miller (1978), Idelchik (1986), and Idelchik and Fried (1989).

18 Flow rate balancing and control.

When designing duct systems, the possibilities for proper flow balancing and control is a main goal. From this point of view, constant static pressure and low air velocities through out the ductwork is ideal and flexible solution. Such large ducts are seldom feasible. Then dampers is used.

Dampers for different applications are used in a ventilation installation. *Regulating dampers* are used to control the air flow rates in a system.
The damper shutter is normally designed so a minimum air flow rate can leak through, even if the damper is shut. This makes it less sensitive than a shut-off damper. to angle changes.
The manual dampers are adjusted when the installation is commissioned, and are cheaper than the automatic ones.

Regulating dampers are of two types, parallel-blade and opposed blade (ASHRAE Handbook 1999). Their control characteristics are dependent on their authority, that is their share of the total pressure drop in the part of the system they serve. According to ASHRAE Handbook (1999), parallel-blade dampers have a linear characteristic (that is, the air flow rate is directly proportional to their degree of opening)at an authority of 1, while the opposed blade type has a near linear relationship at much smaller authorities, ca 5 %. This is important from fan energy point of view.

Shut-off dampers are used to save energy by stopping unintentional airflow, to prevent the spread of dangerous gas and smoke etc. They are often of the single blade type, see also “Fire dampers”.
These dampers often have rubber seals on the damper blade. The damper can either be designed as a straight piece of ducting, or as a T-junction to switch the air flow from one duct to another. The shutter is normally either fully open or fully closed.

Two types of *tightness* are applicable to dampers:

- Tightness to the environment This specifies the magnitude of the air leakage through joints and leaks in the duct sides in relation to the duct surface. This leakage is classified into tightness classes A, B, C and D.
- Tightness against air flow past a closed damper shutter. This refers to the amount of air leaking past the closed shutter, in relation to shutter area. This relationship is classified into five sealing classes 0–4. There are no tightness requirements for the classes 0 and 1 since they are regulating dampers. The highest class, tightness class 4, refers to very tight shut-off dampers.

Damper commissioning and maintenance are discussed by Terranova (2000) and Sellers (2000). Seller warns for duct failures due to fans operating against closed dampers, or negative pressures due to sudden actuating of smoke or fire dampers. Possible solutions are: pressure relief doors where over- and under-pressurisation could occur, interlocked operation of the fire alarm so the fan is stopped before fire or smoke dampers are released, slow damper closure speeds, and interlocked operation of fan and dampers so the fan cannot start if the dampers are not open.

18.1 Systems for constant flow rates (CAV)

The traditional ventilation system is designed for constant airflow rates. The dampers (if they are necessary) are set during commissioning and their position is not intended to be changed until re-commissioning.
Balancing of constant flow systems can be made with the *proportional method*, which traditionally starts with all dampers open. The basis of the method are to first adjust the proportion between the flows, starting with the parts with highest relative air flow rate, and then change the total flow, provided that change is not more than 30%. The steps of the method are:

- Measure the flow rates and calculate the ratios between design and actual flow rate.. Normally dampers are open. Make adjustments to make the final change, less than 30%, possible.

- Start with the zone of the duct system that has the highest flow ratio, and the branch in that zone with highest flow ratio. Find the terminal device on that branch with the smallest ratio, and leave that damper (or similar) fully open. Adjust the other devices to the same flow ratio. Adjust then the flow in each separate branch in order after decreasing branch flow ratio.
- Adjust the flow ratio between the branches of the ventilation zone in the same manner as the terminal devices on one branch. By starting with the branch with the lowest flow ratio one branch damper will be fully open.
- Adjust the other zones in the same manner; in order after decreasing flow ratio.
- Adjust the flow proportion between the zones; starting with the zone with the smallest ratio, which damper should be fully open.
- Adjust the total flow, but changes of more than 30% are not allowed.

Note that this method depends on a fairly large number of dampers. After balancing some dampers should be fully open. Otherwise fan energy is wasted.

Constant flow in duct systems or parts of duct systems can also be maintained with *constant flow dampers*. They are automatic dampers, which maintain a constant, pre-set air flow. The force needed for regulation can be taken from the passing air stream.
By reducing fan speed in the system and at the same time check that all units are working, i.e. that even the worst located unit gets sufficient pressure, you can both reduce electricity use of the fan and the noise production.

18.2 Systems for variable flow rates (VAV)

In this system flow rates to the various zones are varied after demand. The zone flow rate is controlled with a regulating damper. The total airflow rate in the system is typically adjusted to keep a constant static pressure in one point in the system, often located about 2/3 of the duct system length from the fan. This is achieved by modulating fan outlet vanes, inlet vanes, blade pitch, or a variable speed motor.

Balancing of VAV systems is described in ASHRAE Handbook (1999):

- Determine maximum total air flow rates (usually less than the sum of maximum flow rates to each zone);
- Get fan curves and flow rate control information from the fan manufacturer;
- Get from manufacturers the maximum and minimum pressure requirements for terminal devices to be used;
- Construct a theoretical system curve, starting at the minimum pressure requirement of terminal devices to obtain flow, and ending at maximum flow;
- Position the terminal devices to their maximum share of the air flow;
- Set the fan to operation at approximate fan speed;
- Check some terminal devices for pressure and flow rate;
- Measure the total airflow rate;
- Increase the fan speed if the pressure or flow rate is to low;
- Reduce the speed if the air flow rate is correct but the pressure too high;
- Check for duct connection loss at the fan if the air flow rate is correct but the pressure low. If there is no such loss, adjust all terminal devices to the correct air flow rate;
- If applicable, re-run the last four steps with the return or exhaust fan at design flow rate and minimum outdoor airflow;
- Verify the maximum and minimum airflow settings at the terminal devices;
- Set the terminals to minimum and adjust the fan control till minimum static pressure and airflow rate are obtained;
- Take a final decision regarding the placement of the sensor for the pressure controller.

Kosonen and Gronwall (2000) points out that in traditional VAV systems, the static pressure of the system has to be adjusted to a relatively high and constant value. According to them, large fan energy savings (25-55% compared to traditional VAV systems) can be achieved with a model based airflow management system. This system is based on a simulation model of the airflow in the ductwork and intelligent dampers. The authors attribute partly the big savings to the use of dampers with much smaller pressure drop (the fixed minimum static pressure of the damper is 15 Pa) than dampers in conventional systems.

19 References

Author	Year	Title	Journal
Ahmed S. and Brundrett E.	1971	Characteristic length for non-circular ducts	International Journal Heat Mass Transfer, Vol 44 pp 157-159
Allen C. H.	1960	Noise Reduction	(L.L. Beranek, ed.),Ch.21. New York: Mc Graw-Hill
Andegiorgis T. and Peterson F.	1986	Kanalsystems Dimensionering-hastighetsmetoden	Dept. of heating and ventilation
Andersson J.	1998	Akustik och Buller	Svensk Byggtjänst
Ashley C. M., Gilman S. F. and Church R. A.	1956	Branch Fitting Performance at High Velocity	ASHAE Transactions. 62:279
ASHRAE	1999	ASHRAE Handbook: Applications	ASHRAE Handbook-1984, ch 46, Atlanta: American Society of Heating, Refrigerating, and Air conditioning Engineers, Inc.
ASHRAE	1984	ASHRAE Handbook: Systems	ASHRAE Handbook-1984, ch 34, Atlanta: American Society of Heating Refrigeratting and Air Conditioning Engineers, Inc
Backvik B., Bengtson S., Fagergren T., Granberg O. and Jensen L.	1996	En Handbok om Brandskyddsteknik för Ventilationssystem	Ventilationsbrandskydd AB
Batterman S. and Burge H	1995	HVAC Systems as Emission Sources Affecting Indoor Air Quality: A Critical Review	HVAC and Research, January 1995, vol 1, no 1, p 61-80
Björkroth M.	1997	Effect of Particulate Filters on Perceived Air Quality	Healthy Building ´97, Vol 1, p 557-562
Björkroth M., Müller B., Küchen V, P. M. and Bluyssen P. M.	2000	Pollution from Ducts: What Is the Reason, How to Measure It And How to Prevent It	Proceedings of Healthy buildings 2000, vol. 2
Bouwman H. B.	1982	Optimum Air Duct System Design	Report No. C 512 of IMG-TNO, Delft, Holland
Carlsson T., Kovacs P., Karlsson M., Ruud S. and Fransson J.	1995	"State of the art" - Undersökning av ett Större Antal Installationer med Roterande Värmeväxlare	SP-rapport 1995:24
Carrie F. R., Bossaer A., Andersson J. V., Wouters P. and Liddament M. W.	2000	Duct Leakage in European Buildings: Status and Perspectives	Energy and Buildings
Carrié, F.R., Andersson, J. and Wouters, P.	1999	Improving Ductwork - a Time for Tighter Air Distribution Systems	Air Infiltration and Ventilation Center, Coventry, UK
Carrier air conditioning company	1972	Handbook of Air Conditioning System Design	
Colebrook C. F.	1938-1939	Turbulent Flow in the Pipe, with Particular Reference to the Transition Region between the Smooth and Rough Pipe Laws.	Journal of he instituion of civir engineering, London, vol 11, pp 133-156
Colliver D.	1995	Energy Requirements for Conditioning of Ventilating Air	Air Infiltration and Ventilation Centre, 16th AIVC Conference Implementing the results of ventilation research , held Palm Springs, USA, 18 - 22 September, 1995, Proceedings Volume 1, pp 1-12.
Cummings A.	1985	Acoustic Noise Transmission through Duct Walls	ASHRAE Tansactions, Vol 91, part 2A pp 4861
Dean R. H., Dean F. J. and Ratazenberger J.	1985	Importance of Duct Design for VAV Systems	Heating/Piping/Air conditioning, August 1985, pp 91-104
Ebbing C. E. and Fragnito D. and Inglis S.	1978	Control of Low Frequency Duct-Generated Noise in Buildings Air Distribution Systems	ASHRAE Transaction Vol 84, part 2, p 191
Eckert E. R. and Irvine T. F.	1957	Incompressible Friction factor, Transition and Hydrodynamic Entrance Length Studies of Ducts with Triangular and Rectangular Cross Sections	Proceeding of the 5th Midwestern conference on fluid mechanics, University of Michigan press, pp 122-145
Ekelund C.	2001	Runda och Rektangulära ventilationskanaler	Div. Of Build, Serv, Eng, KTH, Stockholm.
Eriksson J.B. and Malmstrom T.G.	2000	Pressure drops in T-junctions - a comparison	ASHRAE Transactions
European committee for Standardisation	1998	Ventilation for Buildings-Air Handling Units-Mechanical Performance	CEN EN 1886:1998

Author	Year	Title	Journal
European committee for Standardisation	1993	Particulate Air Filters for General Ventilation-Determination of the filtration performance	CEN EN 779:93, European committee for Standardisation, Brussels
European committee for Standardization	1997	Ventilation for Buildings-Sheet Metal Air Ducts and Fittings with Rectangular Cross Section-Dimensions.	European committee for Standardization CEN1505
Eurovent	1996	Determination of Fractional Efficiency of General Ventilation Filters	Eurovent 4/10 Eurovent/Cecomat, Paris
Eurovent/Cecomaf	1999	Recommentdation REC. 10 Concerning Calculating of Life Cycle Cost for Air filters	Eurovent/Cecomaf
Evans R. A. and Tsal R. J.	1996	Basic Tips for Duct Design.	Ashrae Journal, July 1996, pp 37-42
Falk Å. And Malmström T.	1983	Rökspridning via ventilationsinstallationer En förstudie	KTH, Uppvärmnings- och ventilationstekniks A4-serie nr 87
Feustel H.and Esdorn H.	1982	Mass Flow Distribution in Highrise Buildings with Forced Ventilation Systems	Conf. proceedings "System Simulation in Buildings". Liège, 1982
Fray A. (Editor)	1988	Noise control in Building Services	Ch 7, Pergamon Press, UK
Gardel A.	1957	Les Pertres de Charge dans les Écoulements au Travers de Branchements en Té	Bull. Tech. De la Suisse Romande. 83:123-130, 144-148.
Gilman S. F.	1955	Pressure Losses of Divided-Flow Fittings	ASHAE Transactions. 61:281
Gunn D J, Darling C W W	1963	Fluid Flows and Energy Losses in Non-Circular Conduits	Trans. OMSTM Chem Engrs, vol 41, pp 163-173
Haerter, A. A.	1963	Flow Distribution and Pressure Change along Slotted or Branched Ducts	ASHRAE Transactions No. 1863
Hägglund Bengt, Nireus Kjell, Werling Per	1996	Pressure Rise Due to Fire Growth in a Closed Room. Description of Three Full-Scale Tests	FOA Scientific Report, Stockholm, Sweden
Hägglund Bengt, Nireus Kjell, Werling Per	1998	An Experimental Study of the Smoke Spread via Ventilation Ducts	FOA Scientific Report Stockholm, Sweden.
Harold R. G.	1986	Round Duct can Stop Rumble Noise in Air-Handling Installations	ASHRAE Transactions 92(2) oo 189-202
Hartner J. P., Koh J. C. Y. and McComas S. F.	1962	A Comparision of Predicted and Measured Friction Factors for Turbulent Flow through Rectangular Ducts	Journal of heat trans. Trans. ASME 84, pp 82-88
Hassan O. and Yue Z.	2001	Litterature Survey: Pressure Drop and Noise Transmission in Rectangular and Round Ducts (Manuscript)	Authers address: Building Science, KTH, 100 44 Stockholm, Sweden
Heck M. and Muller H. A. (Editors)	1975	Pocket Book of Technical Acoustics	Springer, Berlin, pp 313-318
Holl T. W., Gilman S. F., Martin R. T. and Konzo S	1951	Pressure Losses of Take-offs for extended-Plenum Duct Systems	ASHVE Transactions. 57: 419
Howard G. (editor)	2001	Industrial Ventilation -- Design Guidebook	Academic Press
Huebscher R. G.	1948	Friction Equivalent for Round, Square and Rectangular Ducts	ASHVE Transaction 54, pp 101-118
Hull D. O.	1985	VAV Duct Design with Static Regain and a Microcomputer	Specifying Engineer pp 68-72
HVCA	1998	DW/144 Specification for Sheet Metal Ductwork-Low, Medium and High Pressure/Velocity Air Systems	HVCA (Heating and Ventilating Contractors Association), England.
Idelchik I.E.	1986	Handbook of Hydraulic Resistance Chapter 7: Merging of Flow stream and Division into Flow Streams- Resistance Coefficients of Wyes, Tees and Manifolds	New York, USA: Hemisphere Publishing Corporation
Idelchik I.E. and Fried E.	1989	Flow Resistance -A Design Guide for Engineers	New York, USA: Hemisphere Publishing Corporation.
Irving S.	1994	Air - to - Air Heat Recovery in Ventilation	AIVC Technical Note TN 45
Jacobsson S. and Udd B.	1982/6	Ekonomisk dimensionering av ventilationskannel	Tekniskt meddelande, Wahlings installationsutveckling, Stockholm, Sweden
Jagemar L.	1996	Design of Energy Efficient Buildings_-Applied on HVAC Systems in Commercial Buildings	Department of Building Services Engineering, Chalmers University of Technology, Gothenburgh
Jagemar L.	1991	Energiekonomi. Val av Fläktar och Kanalutformning för Luftbehandlingssystem i Servicelokaler och Industrier	Chalmers Unversity, Building services department, Lic. Thesis, Gothenburg

Author	Year	Title	Journal
Jensen L.	1998	Ventilationssystems Uthållighet vid Brand	Lunds university, Institutionen för byggnadskonstruktionslära. Report 98/7046, Lund, Sweden
Jensen L.	1998	Brandgasspridning via Ventilationssystem	Rapport TABK-98/7050 LTH, Lund
Jones C. D.	1979	Friction Factor and Roughness of United Sheet Metal Company Spiral Duct	United sheet metal, Division of United McGill Corp, Westerville, OH
Jones O. C.	1976	An Improvment in the Calculation of Turbulent Friction in Rectangular Ducts	Journal of fluids engineering pp 173-180
Jovent	2000	Råd och anvisningar för installationsarbete	JOVent AB, Stockholm, Sweden
Klote J. H. and Milke J. A.	1992	Design of Smoke Management Systems	ASHRAE 1992, 223pp
Kosonen R. and Grönvall I.	2000	The Model-Based Air Flow Management System	Proceedings of Healthy buildings 2000, Vol.2
Kuehn T. H., Ramsey J. and Threlkeld J. L.	1998	Thermal Environmental Engineering (3rd ed.)	Prentice Hall, NJ, USA
Liddament, M W	1996	A Guide to Energy Efficient Ventilation	AIVC Guide GU03
Lilly J G	1983	Sound Power Radiated by Round Spiral Ductwork	Proceedings of Noise conference 83 pp 297-304, New York:Noise control Fondation
Limb M.	1995	Air Intake Positioning to Avoid Contamination of Ventilation Air	AIVC Annotated Bibliographies, BIB 4
Limb M.	1996	Heat Pumps for Ventilation Exhaust Air Heat Recovery	AIVC Annotated Bibliographies BIB 5
Limb M	1997	Ventilation and acoustics	AIVC Annotated Bibliographies BIB 7
Limb M.	2000	Ventilation Air Duct Cleaning, Office buildings	AIVC Annotated Bibliographies BIB 10
LINDAB	2000	LINDAB Product Catalogue Ventilation 2000	LINDAB AB, Sweden
Ling M K	2001	Acoustics and Ventilation	AIVC Technical Note TN52
Lysne H	1996	Forprosjekter; "Kriterier for Skifte av Filter i Ventilasjonsanlegg" "Komponenter i Ventilasjonsanlegg som Risikofaktor for Helse"	Rapportnr STF84 A96202 SINTEF, Trondheim, Norway
Mauback K.	1970	Reibungsgeserze Turbulenter Strömungen	Chemie-Ing-Techn, 16, pp 995-1004
Melling, A. and Whitelaw, J.H.	1976	Tubulence Flow in Rectangular Duct	Journal of Fluid Mechanics, 1976, v 78, n pt 2, Nov 23
Milke J. A.	2000	Using Models to Support Smoke Management System Design	Fire protection engineering
Miller D.S.	1978	Internal Flow Systems	Cranfield, Bedford, England: BHRA Fluid Engineering
Morey, P., and Williams C. M.	1991	Is Porous Insulation inside an HVAC System Compatible with a Healthy Building?	Indoor Air Q´91, p 128-135
Möritz M.	1996	Verhalten von Mikroorganismen auf Luftfiltern	Universität Berlin
Möritz M.	1998	Hygienische Untersuchungen zur Begrenzung der Standzeit von Luftfiltern in RLT-Anlagen	Universität Berlin
NAIMA	1998	Fibrous Glass Duct Construction Standards	NAIMA (the North American Insulation Manufacturers'Association)
Nikuradse J.	1933	Srömungsgesetze in Rauhen Rohren	VDI-Forschungsheft, 4, pp 1-22
Nikuradse J.	1930	Investigation of Turbulent Flow in Tubes of Non-Circular Cross Section	Ingen.Arch Vol 1, pp 306-332
Nordenadler J.	2000	Clean supply air system	Proceedings of Healthy Buildings 2000, Helsinki
Nordtest	1998	Test Method for Electric Filters. Determination of the electrostatic enhancement factor of filter media	Nordtest NT VVS 117:1998, Nordtest, Espo, Finland
Nyman , E	1991	Mikroorganismer i Ventilationssystem	Byggforskningsrådet R45, Stockholm, Sweden
Op't Veld P. and Passlack-Zwaans C.	1996	IEA Annex 27: Evaluation and Demonstration of Domestic Ventilation Systems. Assessments on Noise.	Proceedings of 17th AIVC Conference, "Optimum Ventilation and Air Flow Control in Buildings", Volume 1, pp 217-225.

Author	Year	Title	Journal
Op´t Veld	1994	Noise Aspects of Ventilation Systems in Dwellings.	Netherlands, Cauberg-Huygen, February 1994, IEA Annex 27 Evaluation and Demonstration of Domestic Ventilation Systems, Report No. 910767-1, 24pp + app
Orme M.	1998	Energy Impact of Ventilation: Estimates for the Service and Residential Sectors	AIVC Air Infiltration Review, Volume 19, No 1,
Pasanen, P	1994	Impurities in ventilation ducts	IAQ´94, p 149-153
Rehme K	1973	Simple method of prediction friction factors of turbulent flow in non-circular channels	International Journal heat mass transfer Vol 16 pp 933-950
Rydberg J.	1968	Om Ventilationsystems Funktion	VVS 39. No. 2 Pages 55-65. Stockholm, Sweden
Rydberg J.	1959	Störningar vid ventilationssystem (Disturbances in ventilation systems)	VVS No.9 pp251, Stockholm,Sweden
Schiller L.	1923	Über den Strömungswiderstand von Rohren Verschiedenen Querschnitts und Rauhigkeitsgrades	Zeitschrift für angewandte mathematik und mechanik, 3, pp 2-13
Scott K. B.	1985	Round vs. Rectangular Duct	Air conditioning, Heating and Refrigerating News pp 41-43
Sellers D.	2000	Dampers - Good Commissioning Begins at Design	HPAC Engineering, September 2000
SMACNA	1995	HVAC Duct Construction Standards Metal and Flexible	SMACNA, USA
SMACNA	1997	Rectangular Industrial Duct Construction Standards	SMACNA
SMACNA	1990	HVAC Systems - Duct Design	SMACNA (The Sheet Metal and Air Conditioning Contractors' National Association, Inc. (SMACNA)
SPIRO	1992	The Best All´Round Solution-A Designer´s Guide to the Benefits of Selecting a Round Ductwork System	SPIRO International S.A
Stampe O.B.	1982	Glent Klimateteknik	Glent&Co, Hvidovre, DK
Stanton T. E. and Pannell J. R.	1914	Similarity of Motion in Relation to the Surface Friction of Fluids	Transaction of Royal society, 214A, London, pp 199
Stratton J. H.	2000	Metal Duct Selection and Application	ASHRAE journal June 2000, pp24-32
Terranova J.	2000	Equipment Notebook - Dampers - Often mistaken for maintenance -free devices	HPAC, Heating Piping Air Conditioning Engineering
Tsai R. J., Behls H.F. and Mangel R.	1990	T-Method Duct Design, Part III: Simulation	ASHRAE Transactions 96 (2)
Tsai R. J., Behls H. F. and Varvak L. P.	1998	T-Method Duct design: Part IV - Duct Leakage Theory	ASHRAE Transaction
Tsai R., Behls J. P., Varvak P. E. and Leo P.	1998	T-Method Duct Design: Part V- Duct Leakage Calculation Technique and Economics	ASHRAE Transaction
Tsai R.J., Behls H.F. and Mangel R.	1988	T-Method Duct Design Part II: Calculation Procedure and Economic Analysis	ASHRAE Transaction
Tsai R.J., Behls P.E. and Mangel R.	1988	T-Method Duct Design, Part 1:Optimization Theory	ASHRAE Transaction
Tsal R. J. and Adler M. S.	1987	Evaluation of Numerical Methods for Ductwork and Pipeline Optimization	TA Vol 93/1
Tsal R. J., Behls H. F. and Varvak L. P.	1998	T-Method duct design: Part IV - duct leakage theory.	ASHRAE Transactions CD, proceedings of the 1998 ASHRAE Annual Meeting, held Toronto, Canada, June 1998, 13 pp
Tsal R. J., Behls H. F. and Varvak L. P.	1998	T-Method duct design, Part V: duct leakage calculation technique and economics	ASHRAE Transaction
VDI	1998	Hygienic Aspects for the planning, Design, Operation, and Maintenance of Air-Conditioned Systems	VDI 6022:1998
Ver I L		Predicting of sound in straight lined and unlined ductwork of rectangular ductwork	ASHRAE TRP--319 BBN report 5116
VVS	1998	VVS AMA 98	Svensk Byggtjänst
VVS	1998	RA VVS 98	Svensk Byggtjänst, Stockholm

Author	Year	Title	Journal
Wallin, O.	1994	Computer Simulation of Particle Deposition in Ventilating Duct Systems	Sweden, Royal Institute of Technology, Div. For Building Serv. Eng. Bulletin No 31, April 1994, 162 pp refs
Wigenstad T.	2000	Optimalisering av Föringsveier.	Report NTNU, Inst. för klima- og kuldeteknikk, NTNU, Trondheim, Norway
Woods J. and Goodwin A.K.	1997	Glass Fiber Emission from HVAC Ductwork: A Review of the Literature	Healthy Buildings/IAQ´97, Vol 1, p 587- 591
Woods Fan Division	1973	Design of Sound	The Electirc Company
Wright J. A.	1994	Optimum Sizing of HVAC Systems by Algorithm	4th Int. Conf. on System Simulation in Buildings, Liège, Belgium. Page 503-526
Wright S. K.	1945	A New Friction Chart for Round Ducts	ASHVE Transaction 51:303-316
Wu T., Driesen G.E. and Amin D.A.	1998	Structural Integrity Justification of a Rectangular Duct under Large External Pressure	Proceedings of the 1998 ASME/JSME Joint Pressure Vessels and Piping Conference

The Air Infiltration and Ventilation Centre was inaugurated through the International Energy Agency and is funded by the following six countries:

Belgium, France, Greece, the Netherlands, Norway and United States of America.

The Air Infiltration and Ventilation Centre provides technical support in air infiltration and ventilation research and application. The aim is to promote the understanding of the complex behaviour of the air flow in buildings and to advance the effective application of associated energy saving measures in the design of new buildings and the improvement of the existing building stock.

Air Infiltration and Ventilation Centre
Operating Agent and Management
INIVE EEIG
Boulevard Poincaré 79
B-1060 Brussels
Belgium

www.aivc.org
aivc@bbri.be

Tel: +32 2 529 81 00
Fax: +32 2 529 81 10
inive@bbri.be
www.inive.org