THE IMPLEMENTATION AND EFFECTIVENESS OF AIR INFILTRATION STANDARDS IN BUILDINGS

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PAPER 9

AIR LEAKAGE OR CONTROLLED VENTILATION?

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SYNOPSIS

This paper compares the conventional exhaust system with a supply-exhaust system concerning which degree of control of the air exchange in the individual rooms is possible.

Ventilation efficiency and air exchange efficiency are defined. Some examples show the local concentration, mean ventilation efficiency and mean air exchange efficiency for some simple ventilation schemes. Exhaust systems require a very tight building with small make up air openings.

The different systems' ability to avoid leakage out from the building of indoor air is compared too. The calculations indicate that the exhaust system can give small duration of indoor air. The supply-exhaust system gives a greater duration, but the flow can be considerably reduced if the supply air flow rate is reduced. An extremely low velocity exhaust system will also reduce the unwanted outflow.

INTRODUCTION

Ventilation comes from the latin word "ventilare" which means "expose to the wind". In Rome, where the climate is nice, people then probably thought of a gentle, cooling breeze. In Sweden a more natural association is draught. The art of ventilation, as it is practiced in Sweden of today consequently aims at not being noticed. It is typical that the increased interest in ventilation the last few years originated from damage in buildings, especially in small detached houses used as family dwellings. The damages are due to moisture. They have to a great extent been attributed to lack of air exchange due to the tightness of many new buildings.

For a ventilation engineer it is natural to supply the required flow rate of ventilation air in a controlled way, matching the degree of control of the building itself. The answer for him is a mechanical supply and exhaust system with heat recovery from the exhaust air. The recovered heat can be used for preheating the supply air. However, in order to avoid overpressure inside the building, which could cause condensation of moisture from humid indoor air in the walls, it is customary to supply only 70 - 90% of the exhaust flow rate. This somewhat decreases the efficiency of the heat recovery, but more important is that in summertime no preheating of the supply air is needed. In order to make use of heat from the exhaust air the whole year the use of heat pumps is gaining popularity. The heat is then used for the preparation of hot water or for heating.

Ventilation air flow rates are low. The Swedish Building Code requires a flow rate corresponding to about 0,5 air changes. But the Code also recommends that when there is a heating demand the flow shall not exceed this minimum flow rate. In offices, for instance, this small flow rate is not sufficient for climatisation summertime. Consequently recirculation air is used during the heating season, which gives a better mixing. Return air is also used in other types of building sometimes, in order to increase total flow and avoid stagnation zones. One example is children's service homes.

For the building engineer, on the other hand, mechanical ventilation is not a natural choice. He will probably accept mechanical exhaust ventilation though, even if it is not required by the Building Code. But the supply system is often considered as "double" unneccessary ducting. (Also the official regulations of loans for building have this attitude). Often reference is given to the risk for overpressure and condensation in the walls, although there are very few indications that there really should exist such a risk when the ventilation air flow rate is the stipulated one, controlling air humidity. Another argument is of course the possibilities to use a heat pump for heat recovery.

In his Keynote Address to the 3rd AIC Conference, Mr. Billington pointed out that ventilation historically has been associated with heating : it aimed at removing smoke produced by fires. The fact that the smoke could be seen focused interest on its effective removal. The supply air, on the other hand, can not be seen. Also today removal of moisture and odours from kitchen and bathroom often are considered as the real purpose of the ventilation of dwellings neglecting other air quality and air distribution aspects. The consequence of this is that the exhaust air is taken only in the kitchen and the bathroom (in order to get as big removal flow rates as possible there) but also that the total exhaust air flow rate is controlled, not really the air exchange in the dwelling. Also this tradition is an argument for using only exhaust systems.

If a tight house is provided with a mechanical exhaust ventilation system, special make up air openings are made in the walls. Of course these artificial air leakage openings mean better control than "natural" leakage openings but they also give problems, especially for comfort. A concentrated cold air flow entering a room is more likely to cause draught problems than the diffuse flow from many small openings. The make up air can be heated, by for instance placing the opening behind a radiator, but then the risk for freezing has to be dealt with.

Another aspect is the quality of the air. It is easier to filter supply air than make up air, especially as effective filtering of make up air means that the indoor air pressure has to be decreased because of the necessary pressure loss. A supply system also means possibilities for cooling the supplied air, for instance.

In practice make up air normally is not treated in any way, neither heated nor cleaned, which then is a significant difference in quality compared with supply air. Draught problems often cause the make up air opening to be closed, which of course considerally changes the planned ventilation, especially as they are reopened only after long time.

But which degree of control of the air exchange in the individual rooms in a dwelling is possible with a conventional exhaust system? This will be discussed in this paper and illustrated by means of simple examples. First, however, the means of expressing differences in the efficiency of systems to distribute air and remove contaminants will be defined.

VENTILATION EFFICIENCY AND AIR EXCHANGE EFFICIENCY

As mentioned in the introduction ventilation has traditionally been associated with removal of contaminants. The customary definition of *ventilation efficiency* reflects this fact as it is a measure of the systems ability to remove contaminants emitted in the room.

Figure 1 illustrates what happens when a contaminant is emitted. The mean ventilation efficiency $\langle\varepsilon\rangle$ for the room is defined as

$$\langle\varepsilon\rangle = \frac{C_e(\infty) - C_s}{\langle C(\infty)\rangle - C_s}$$

where $C_e(\infty)$ = Steady state concentration of contaminant in the exhaust air (kg/m^3)

$\langle C(\infty)\rangle$ = Steady state mean concentration of contaminant in the room (kg/m^3)

C_s = Concentration of contaminant in the supply or make up air (kg/m^3).

$\langle\varepsilon\rangle = 1$ can be used as a reference value. Then the exhaust air has the same concentration of contaminants as the mean concentration in the room. This can happen for instance if the contaminants and the room air are well mixed. $\langle\varepsilon\rangle < 1$ indicates changes in $\langle C(\infty)\rangle$ as $C_e(\infty)$ has a fix value, due to mass continuity. $\langle\varepsilon\rangle$ bigger than 1 indicates good function, clean room air.

It can be noticed that the size of the room has no direct influence on the ventilation efficiency, but on the length of the transient period before steady state is reached. It is natural to relate the air flow rate q to room value V.

$$n = \frac{q}{V}$$

where n = the specific air flow rate $\left[\frac{m^3/s}{m^3}\right]$

q = ventilation air flow rate (m^3/s)

V = room volume (m^3)

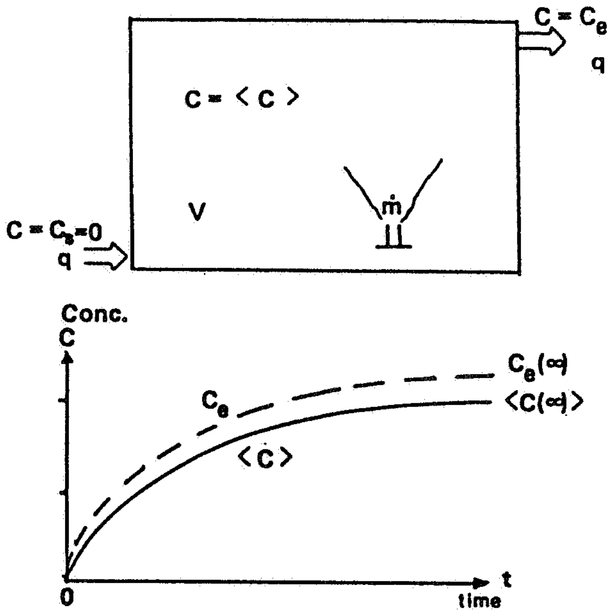


Figure 1. Emission of a contaminant in a room. The emission starts at time t=0. The concentrations are then zero because the concentration in the supplied air is supposed to be zero. V is the volume of the room (m^3) in the emission rate of contaminant (kg/s) and q the ventilation air flow rate (m^3/s). Density of air is supposed to be constant.

The specific flow is often expressed in $m^3/h/m^3$ or h^{-1} and is then called "air changes per hour". The value n^{-1} obviously indicates the time for one "air change", which time is called τ_n.

$$\tau_n = \frac{1}{n} = \frac{V}{q}$$

τ_n is a measure of the average residence time in the room of the air leaving the room that is the time between passing the supply device and passing the exhaust device. The average residence time in the room of the contaminants leaving the room τ_t^c, can also be defined, see Sandberg (1983). As τ_t^c is the time in which an amount of contaminants corresponding to the total content in the room (corrected for C_s) is exhausted from the room, mass balances give (compare fig. 1):

$$\dot{m}\,\tau_t^c = V\cdot(\langle C(\infty)\rangle - C_s)$$

$$\dot{m} = q\cdot(C_e(\infty) - C_s)$$

This gives

$$\langle\varepsilon\rangle = \frac{\tau_n}{\tau_t^c}$$

The mean ventilation efficiency $\langle\varepsilon\rangle$ for the room can also be expressed as the ratio between the mean residence times in the room of the exhaust air and of the contaminant in the exhausted air. Big ventilation efficiency indicates rapid removal of contaminants.

Obviously the ventilation efficiency depends not only on the ventilation of the room but also on the location of the contaminant source.

τ_n is the time it takes to remove an amount of air corresponding to the volume V of the room. If in the time τ_n all the air in the room shall be exchanged too the air must flow as a piston through the room, see fig. 2. This of course is impossible.

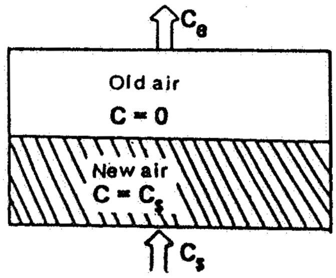


Fig. 2. Piston flow of air through a room, with no mixing of "old air" and "new air". The "new air" is thought to be marked by means of tracer gas, starting at time t=0. At $t=\tau_n$ all the "old air" is removed.

Piston flow is the upper limit, the most effective way of exchanging the air in the room. The concentration of "old air" is the maximal, 100%, until there is no more. Another popular reference case is "complete mixing". Then the concentration of "old air" is the same in all the room. This means that the location of the exhaust terminal device in this special case is unimportant. It means also that the exchange of air is less effective than for "piston flow" because all the time also some new air is exhausted. For "complete mixing" the exchange of air is described by the well-known equation:

$$\frac{\langle C(t)\rangle}{\langle C(\infty)\rangle} = 1 - e^{-\frac{t}{\tau_n}}$$

where $\langle C(t)\rangle$ is the concentration of "new air" in the room (and in the exhaust air) at the time t.

In fig. 3 a graph of this equation is plotted and also a straight line showing how air is exchanged in the piston flow case. The curves also show the distributions of residence time (in the room), that is the time from passing the supply device, for the room air in the two cases. For instance, after a time τ_n 100% of the room air has a residence time shorter than τ_n for the piston flow case but only 63% for the "perfect mixing" case. The area above the curves is a measure of the mean residence time in the room $\langle\bar{\tau}\rangle$. For the case piston flow this area is A in Fig. 3 which gives

$$\langle\bar{\tau}\rangle = \frac{\tau_n}{2} \qquad \text{("Piston flow")}$$

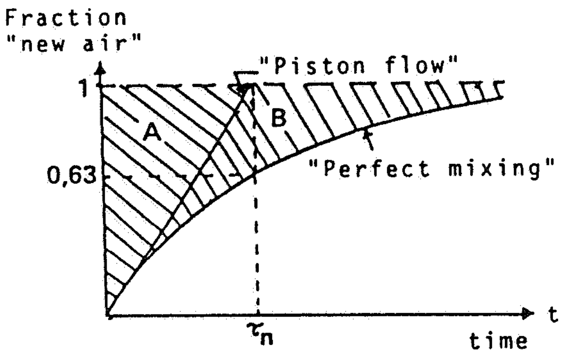


Fig. 3. Graphs illustrating the exchange of air in a room for the two cases "piston flow" and "perfect mixing".

For "perfect mixing" the area is A + B in fig. 3. This area can be calculated by integrating the basic equation given above:

$$\langle\bar{\tau}\rangle = \tau_n \qquad \text{("perfect mixing")}$$

The smallest possible value for the mean residence time for the room air is $\tau_n/2$, which is valid for piston flow. The *air exchange efficiency*, ε_a, is defined as the ratio between this minimum value and the value $\langle\bar{\tau}\rangle$ for the case in consideration:

$$\varepsilon_a = \frac{\tau_n}{2\,\langle\bar{\tau}\rangle}$$

"Perfect mixing" obviously gives $\varepsilon_a = 0{,}5$. $\varepsilon_a < 0{,}5$ means short-circulating ventilation air from the supply device to the exhaust device and consequent stagnation zones. $\varepsilon_a > 0{,}5$ means that there is a tendency towards piston flow which is good.

If τ_n is known, that is the ventilation flow rate q and the volume V, both $\langle\varepsilon\rangle$ and ε_a can be calculated from concentration measurements in the total exhaust air only.

DIFFERENT VENTILATION SCHEMES

In order to compare some different types of ventilation schemes, calculations of local concentrations, ventilation efficiency and air exchange efficiency have been made for the simple combinations of two rooms shown in fig. 4. "Perfect mixing" of air and contaminant has been assumed within each room and there is no circulation of air between the rooms. The efficiency values are valid for the total system. The results are shown in fig. 4, too.

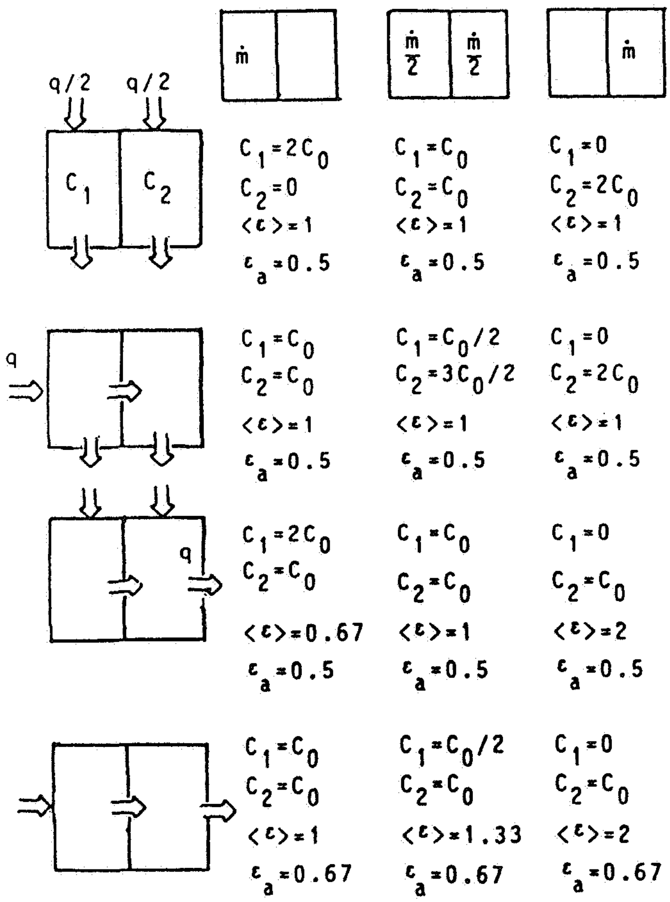


Fig.4. Local concentrations, mean ventilation efficiency $\langle\varepsilon\rangle$ and mean air exchange efficiency ε_a for four different, simple ventilation schemes and three different contamination sources. "Perfect mixing" in each room. $\frac{\dot{m}}{q} = C_o$, $C_s=0$.

The two extreme cases are that the rooms are ventilated separately or in series. With the assumption made the first case will have efficiencies as a single room with perfect mixing. The other extreme case, the connection in series, has a tendency towards piston flow and gives better efficiency values. Also the maximum local concentration that occurs is lower for this case.

The two intermediate cases both have the same air exchange efficiency as the first case, separate ventilation, and also the same maximum local concentration.

The case with only one exhaust point has different ventilation efficiencies, though. when the contamination emission is in one room only, illustrating the improved removal function when the exhaust air flow is big and close to the contamination source but also the deterioration when the contamination source is located so that the contamination is diluted by only part of the flow and the ventilation distributes the contamination to other rooms.

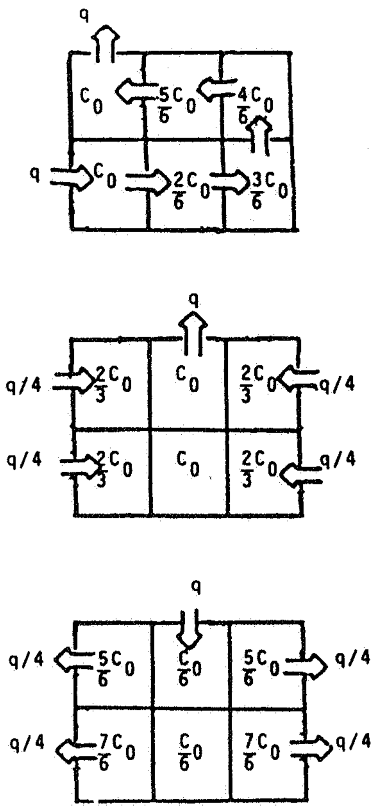
The example indicates the wellknown and obvious rules for ventilation lay-out:

- o as much air as possible shall flow through each room,
- o all rooms shall have a supply or an exhaust air terminal device or be connected in series between rooms with such devices,
- o exhaust air terminal devices shall be located as close to the main contamination sources as possible.

The limiting factors are

- can contamination from one room be allowed to flow through another,
- leakage for the outside, between rooms and circulation of air between rooms,
- draught problems and noise.

Other important functions are of course economy and practical construction problems.



$\langle\varepsilon\rangle = \frac{36}{21} = 1.71$

$\varepsilon_a = \frac{36}{42} = 0.86$

$\langle\varepsilon\rangle = \frac{9}{7} = 1.29$

$\varepsilon_a = \frac{27}{42} = 0.64$

$\langle\varepsilon\rangle = \frac{9}{7} = 1.24$

$\varepsilon_a = \frac{27}{42} = 0.64$

Fig.5. Ventilation efficiency $\langle\varepsilon\rangle$, air exchange efficiency ε_a and local concentration for three ventilation schemes. Each room has a volume V/6 (V=total volume). Contamination emission $\dot{m}/6$ in each room. $\dot{m}/q = C_o$.

Another example is shown in fig. 5. It is a case also appearing in the next section of this paper.

As the total volume in this example is divided into six rooms instead of two, the "connection in series" case, used only for reference, results in a bigger value of the ventilation efficiency $\langle\varepsilon\rangle$ and the air exchange efficiency ε_a. The reason is that this case is closer to "piston flow" as "perfect mixing" is assumed in each room. It can also be noticed that for all three cases in fig. 5 $\langle\varepsilon\rangle$ is 2 ε_a. This is a consequence of the fact that the contamination sources are distributed evenly in the building, in this example.

The case with four supply air devices and one exhaust air device is a normal scheme, for instance for mechanical exhaust system. It has a good efficiency in the example, $\varepsilon_a = 0{,}64$. The reserved, third, case has the same efficiencies but a less favourable distribution of local concentrations.

The examples indicate that schemes like the second case in fig. 5, which are normal for mechanical exhaust systems among others are good. In the next section their possibilities to work as planned will be discussed.

INFLUENCE OF STACK EFFECT AND WIND

Consider a building with a mechanical exhaust system, fig. 6.

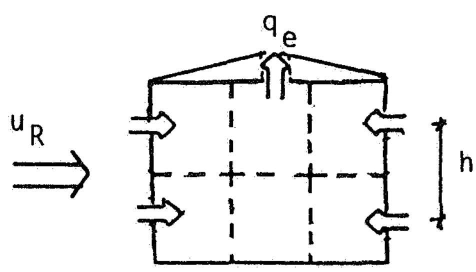


Fig. 6. Ventilation scheme for a building. The exhaust air flow is fixed, q_e. The wind velocity is u_R (m/s) and the height difference between the openings h (m).

The openings in the wall indicated in the figure are assumed to represent both make up air openings and the leakages. Flow resistances within the building are neglected. The exhaust air flow is fixed. The air flow when the system is influenced by wind pressure and stack effect has been calculated by Etheridge and Sandberg (1984). According to them the flow depends on the number

$$A_r = \frac{\Delta\rho\ g\ h}{\rho_o\ u_R^2}$$

where ρ_o = density of outside air (kg/m^3)

$\Delta\rho$ = difference of density between outdoor and indoor air (kg/m^3)

g = acceleration due to gravity (m/s^2)

h = height difference between openings (m)

u_R = wind speed at reference point (m/s)

A_r is a measure of the ratio between buoyant forces and wind forces. But the pressure difference acting across the building also depends on the form of the building and of the surroundings. This can be expressed by means of the difference between the pressure coefficients on the wind side and on the leeward side, ΔC_p. Etheridge and Sandberg consequently have expressed the flow as a function of

$$\frac{\Delta C_p}{A_r}$$

ΔC_p normally has values in the interval

$$0{,}2 < \Delta C_p < 1{,}0$$

In table 1 values are given for $\Delta C_p/A_r$ for different wind speeds u_R, for an indoor temperature of 21°C and an outdoor temperature of 6°C which is the yearly mean temperaturefor Stockholm. The height h is 3 m.

u_R (m/s)	1	2	3	5	7
ΔC_p = 0,2	0,13	0,50	1,13	3,15	6,17
ΔC_p = 1,0	0,63	2,52	5,67	15,7	30,9

Table 1: Values of $\Delta C_p/A_r$ for an outdoor temperature of 6°C, h = 3 m.

The flow through each opening is first supposed to vary as

$$q = 0{,}009 \cdot \sqrt{\Delta_p} \quad (m^3/s)$$

where Δ_p = pressure difference across the opening (Pa).

This gives a total flow through the four openings of 900 m^3/h when Δ_p = 50 Pa, which is the standard for leakage tests. If the volume of the building is 300 m^3 this flow corresponds to 3 air changes per hour, which is the maximum value stipulated for small buildings in the Swedish Building Code for leakage. As this value in our example includes also the flow through the make up air openings, the building is tighter than required.

In this example it is assumed that the openings discharge coefficients are 1.

Fig. 7 shows different flow configurations and the values of $\Delta C_p/A_r$ when the change occurs for a fixed exhaust air flow rate corresponding to 0,5 air changes per hour. When the change occurs the resulting air flow in the room, where the flow changes direction, obviously is zero. A comparison with table 1 shows that the first change is at a low value of $\Delta C_p/A_r$ valid for small wind velocities, which are normal and frequent. Compare fig. 8. If ΔC_p is big also the second change occurs at low and frequent wind velocities. Note that table 1 is valid for the yearly mean outdoor temperature, +6°C, and not for an extreme case.

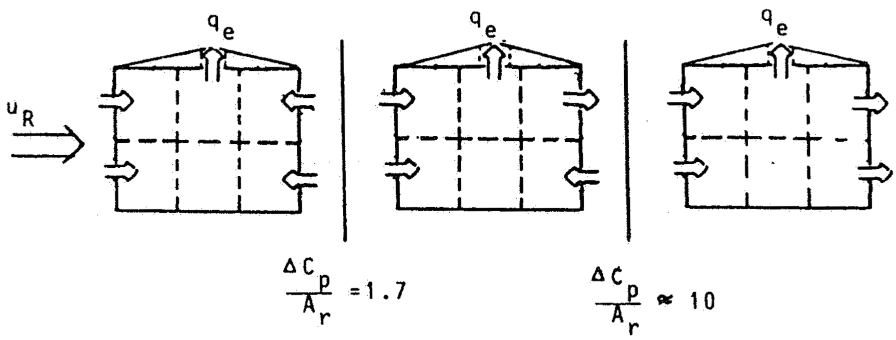


Fig. 7. Different flow configurations for a mechanical exhaust system. The total volume of the building is 300 m^3 and the fixed exhaust air flow q_e = 150 m^3/h.

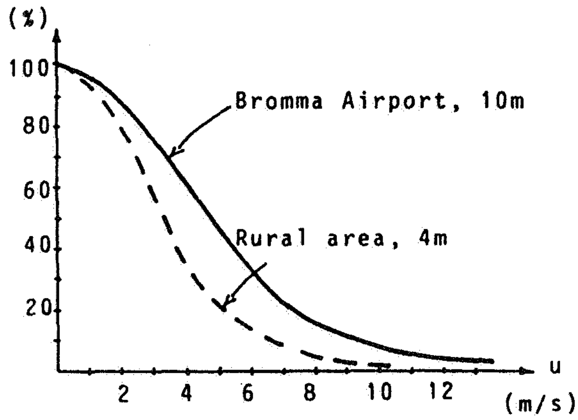


Fig. 8. Wind velocity in Stockholm at a height of 10 m in a free field and reduced to 4 m and rural area.

If the building is tighter , say corresponding to a total flow at 50 Pa of 2 air changes instead of three (including the flow through the make up air openings), the first change will occur at $\Delta C_p/A_r \approx 6$. Also this value corresponds to frequent wind velocities, if ΔC_p is not small. A total tightness value of 1 air change per hour at $\Delta_p = 50$ Pa gives the first change at $\Delta C_p/A_r \approx 20$. It is evident from table 1 and fig. 8 that also this value is within the normal intervals although it will not occur frequently.

Consider again the first discussed case, corresponding to a total air change of 3 air changes per hour at 50 Pa. At an outdoor temperature of about $-20^{\circ}C$ and no wind all the exhaust air will enter through the two lower openings. This is because of the buoyant force. This does not only give bad ventilation but also draught problems.

The example indicates that in order to control the exhaust ventilation very tight buildings with small make up air openings are required. In reality for instance the turbulent nature of the pressure fluctuations and thermal connection improves the exchange of air. The tendency is correct, however. Experiments made at the National Swedish Institute for Building Research indicate that mechanical exhaust systems start to function properly only when the leakage at a pressure difference of 50 is smaller than 1 air change per hour (with the make up air openings closed), that is the building must be very tight.

LEAKAGE OF INDOOR AIR

As mentioned in the introduction, there is in Sweden much consideration about the risk for condensation in the walls. Mechanical exhaust systems are considered safer, from this point of view, than supply and exhaust systems. In this section a comparison between different systems will be made concerning their ability to avoid leakage out from the building of indoor air.

As an example consider the simple scheme according to fig. 9.

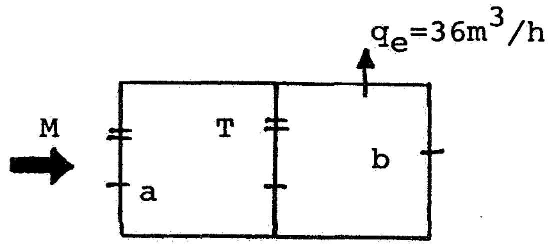


Fig. 9. The ventilation scheme for mechanical exhaust.

The volume of each room is 36 m^3. The exhaust air flow rate is fixed and 36 m^3/h, that is 0,01 m^3/s. In order to get the room connected in series the make up air opening M, the transfer air opening T (including leakage) and the leakages a and b, are concentrated only to the walls according to fig. 9. The total leakage a + b corresponds to 1 airchange per hour at Δ_p = 50 Pa. The flow rates are calculated from the appropriate pressure difference (in Pa) as shown below. At a pressure difference of 50 Pa 2 q_M is equal to 1 air change per hour and $q_a + q_b$ is also equal to 1 air change.

$$q_M + q_a = 0{,}0019\ \Delta_p^{0{,}6} \quad (m^3/s)$$

$$q_T = 0{,}011 \cdot \Delta_p^{0{,}55} \quad (m^3/s)$$

$$q_a = q_b = 0{,}00064\ \Delta_p^{0{,}7} \quad (m^3/s)$$

The wind pressure coefficient for the building is ΔC_p = 0,7 which is in the middle of the interval mentioned in the preceding section.

Fig. 10 shows the leakage flow. Note that it is defined positive inward and that it is given in m^3/h. The flows have been calculated for different wind velocities and the durations have been calculated from fig. 8, the lower curve. As can be seen from fig. 10, there will be a flow out of indoor air 12 % of the time. This is of course a little misleading as the wind is not blowing perpendicular to the building all the time. However for a detached house there will always be one windside and one leeward side. What the graph indicates is that there is a risk for outflow somewhere in the building 12 % (or a little bit less) of the time.

Fig. 11 shows results when there also is a fixed supply air flow, 90 % or 70 % of the exhaust air flow. For this case there are not make up air openings in the walls, only leaks. As can be seen there is a considerable reduction of outflow when the supply air rate is reduced. For the reduced air flow rate outflow has a duration of 30 % which is about three times the time compared with the example for the exhaust system, see fig. 10.

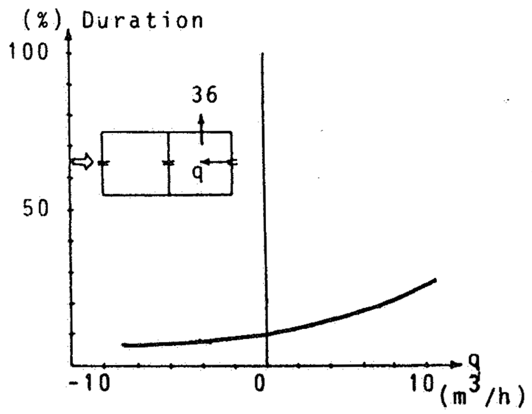


Fig. 10. Leakage air flow on the leeward side for a mechanical exhaust system.

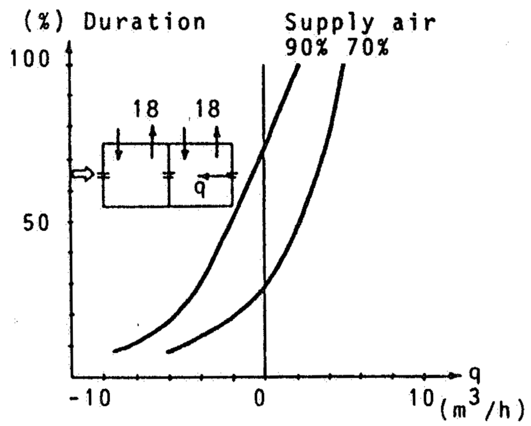


Fig. 11. Leakage air flow on the leeward side for a mechanical supply and exhaust system. The supplied air flow is fixed, as is the exhaust air flow. Calculations are made for two cases, supplied air flow rate 90 % and 70 % of the exhaust air flow rate.

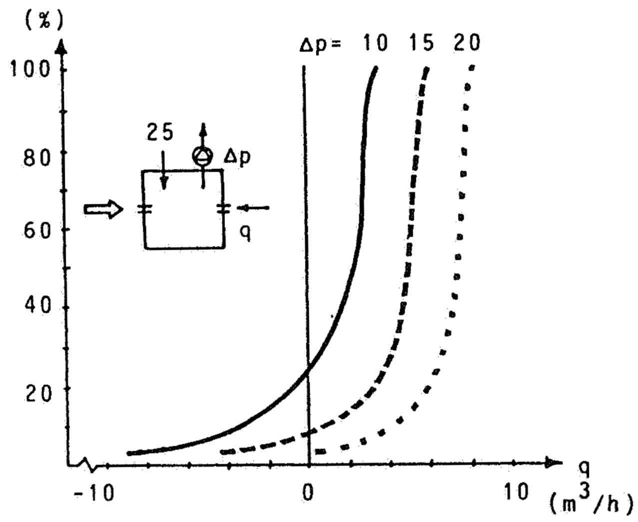


Fig. 12. Leakage air flows on the leeward side for a mechanical supply and exhaust air system. The supply air flow rate is fixed but the exhaust air system is of low pressure type.

Fig. 12 shown only for discussion. It is calculated for an extreme low velocity and low pressure exhaust system, which is supposed always to work against the leeward side pressure at its outside opening. The pressure loss in the exhaust ducting is

$$\Delta_p = 40\,000 \cdot q^{1,8} \quad \text{(Pa)}$$

The pressure loss at nominal flow is 10 Pa. The parameter Δ_p in fig. 12 refers to the fan pressure which is very low. When the fan pressure is increased from 10 Pa to 15 Pa a negative pressure is created inside the building which counteracts outflow (the fan curve is supposed to be straight, giving the same pressure independent of the flow). If for instance a heat recovery system is used the pressure loss in the duct system will be too big. A similar effect could then be achieved by using a variable fan and control the pressure difference.

Fig. 11 shows that a considerable reduction of outflow can be achieved by using a supply air flow rate lower than the exhaust air flow rate. In order to maintain this advantage it is necessary to clean the exhaust ductwork regularly. Swedish experience is that reductions of about 50 % (of the flow) may happen within one year.

The example indicates that the exhaust system in a tight house and with small make up air openings can give a small duration of outflow of indoor air. The supply and exhaust system have a bigger outflow although a considerable reduction is achieved by reducing the supply air flow rate which is customary to do in Sweden.

CONCLUSIONS

In order to have good ventilation each room in a dwelling must have a controlled air exchange. This is not possible to get with a mechanical exhaust system unless the building is very tight. The outflow of indoor air is bigger with a supply and exhaust system than with an exhaust system. A considerable reduction is achieved, however, if the supply air flow rate is reduced.

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