Impact of ductwork leakage on the fan energy use and sound production of central mechanical ventilation units in houses

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ABSTRACT

Various studies demonstrate a significant impact of ductwork leakage on the fan power consumption of ventilation systems. They have shown that the total energy used by fans can be reduced by 30-50% by improving the airtightness of the ductwork system. However, most of those studies focused on non-residential and multi-family buildings. This study focuses on single-family dwellings; specifically houses.
This paper first explains why fan energy use increases with ductwork leakage and then presents a model, which is based upon (Leprince & Carrié, 2018), that is used to estimate the impact of ductwork leakage on the fan energy use of central mechanical ventilation units with heat recovery in three houses and with a DCV system in one house. The calculations have shown that fans connected to leaky ductwork (3*Class A) in the four houses use 57-169% more energy than fans connected to very airtight ductwork (Class D), if they ventilate to provide the hygienic flowrate at Air Terminal Devices.
Obviously, the harder a mechanical ventilation unit has to work to displace more air to achieve the hygienic flowrate at the Air Terminal Devices, the more sound it will produce. It is estimated that a mechanical ventilation unit with heat recovery will produce at least 2.5 dB(A) more sound pressure level in the habitable rooms whith leaky ductwork.

KEYWORDS

Ductwork airtightness, fan energy use, single houses, sound pressure

1 INTRODUCTION

There are a number of studies that demonstrate a significant impact of ductwork leakage on fan energy use (Soenens & Pattijn, 2011) (Stroo, 2011) (Berthault, Boithias, & Leprince, 2014) (D.F., 2011) (Bailly, Duboscq, & Jobert, 2014) (Levinson, Delp, Dickerhoff, & Modera, 2000) (Carrié, Bossaer, Andersson, Wouters, & Liddament, 2000) (Krishnamoorthy & Modera, 2016) (Leprince & Carrié, 2018).
(Soenens & Pattijn, 2011) concluded that more than 30% of the energy used by the fans in the ventilation systems in a hospital wing, care home and office building could be saved using airtight ventilation systems (Soenens & Pattijn, 2011). Those results are consistent with (Stroo, 2011) and with the experimental study of Berthault in a multi-family building (Berthault, Boithias, & Leprince, 2014), which concluded up to 50% energy savings with class C airtight ductwork compared to 1.5*class A.
However, recent measurements performed in France in the context of the Effinergie + label (Moujalled, Leprince, & Mélois, 2018) have shown that almost 50% of the ductwork systems in the tested houses have ductwork airtightness 2.5*class A or worse. This stresses the need to

change construction habits because the ductwork in most of the tested buildings was designed to achieve at least class A (required by the Effinergie + label), but missed the target. Unfortunately, the negative impact of ductwork leakage on fan energy use and sound production is still neglected in most countries (Leprince, Carrié, & Kapsalaki, 2017), particularly in residential buildings.
This paper aims to:

- explain the impact of ductwork leakage on flowrate and pressure drop;
- calculate the impact of ductwork leakage on the fan energy use of central mechanical ventilation units with heat recovery in 4 houses with different ductwork systems, hygienic flow rates and pressures drops;
- estimate the impact of ductwork leakage on the sound pressure in bedroom and living-rooms

2 IMPACT OF DUCTWORK LEAKAGES ON FAN ENERGY USE AND SOUND PRESSURE

2.1 Fan energy use

The fan power consumption depends upon the flowrate produced by the fan and the pressure difference on either side of the fan.
The nominal efficiency of the fan is defined by the following equation:

$$\eta = \frac{\Delta P * Q}{P_{el} * 3600} \qquad (1)$$

η	-	Efficiency of the fan
ΔP	Pa	Pressure difference at fan
Q	m^3/h	Flowrate at fan
P_{el}	W	Electrical power of the fan

This efficiency may not be constant according to the pressure difference and flow rate.
The higher the pressure drop (resistance) in the ductwork, the higher the pressure difference the fan needs to produce to overcome this resistance and achieve the hygienic flow rate. Generally, axial fans are able to produce high flowrates, but cannot generate enough large pressure difference to overcome any resistance without running at higher speeds and producing more sound. On the other hand, centrifugal fans are able to generate large pressure differences, but their flowrates are limited.

2.2 Pressure losses

Pressure drop in ductwork systems is due to the irreversible transformation of mechanical energy into heat (ASHRAE, 2013). There are two types of losses:

- friction losses (occurring along the ductwork)
- and dynamic losses (occurring at bends and junctions)

Friction losses

Friction losses occur along the entire length of duct. They are due to fluid viscosity. Friction loss can be calculated using the Darcy equation (ASHRAE, 2013)

$$\Delta p_f = \frac{1000 f L}{D_h} * \frac{\rho V^2}{2} \qquad (2)$$

Δp_f	Pa	Friction losses in terms of total pressure
f	-	Friction factor
L	m	duct length
D_h	m	hydraulic diameter
V	m/s	velocity
ρ	kg/m^3	air density

Friction losses are proportional to the flow velocity to the power of 2 so also to the square of the flowrate.

Dynamic losses

Dynamic losses result from flow disturbance caused by duct accessories, which change the direction of the flow (bends) and of the hydraulic diameter (adaptors) and at converging/diverging junctions.
Dynamic loss can be calculated using the following equation (ASHRAE, 2013):

$$\Delta p_t = \frac{C\rho V^2}{2} \qquad (3)$$

C	-	Total loss coefficient
Δp_t	Pa	Total pressure loss
V	m/s	velocity
ρ	kg/m^3	air density

Total pressure loss in the ductwork

Total pressure loss in a duct section is calculated by combining friction and dynamic losses.

$$\Delta p = \left(\frac{1000f}{D_h} + \sum C\right)\left(\frac{\rho V^2}{2}\right) \qquad (4)$$

Therefore, the pressure loss in the ductwork system is proportional to the square of the flowrate and the higher the flowrate to overcome ductwork leakage, the higher resistance in the ductwork.

Fan and pressure losses

The fan needs to compensate for the additional flowrate due to ductwork leakage and also the additional pressure drop to maintain the hygienic flowrate. Therefore, both the flowrate and the pressure at the fan needs to be increased.

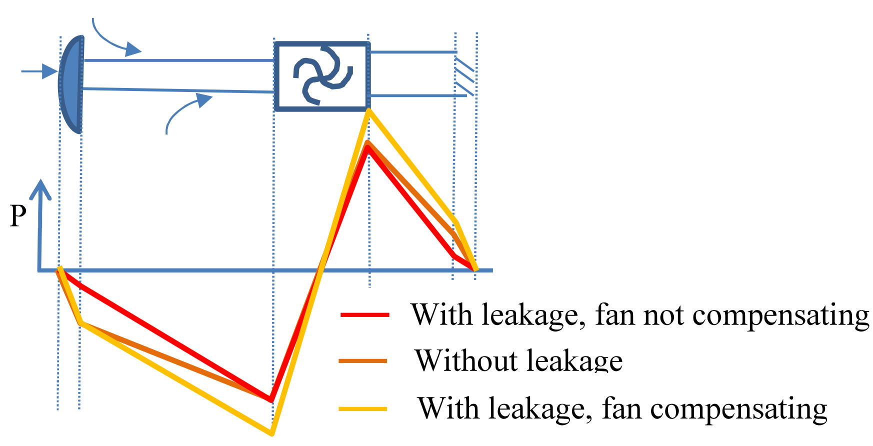

Figure 1: Pressure profile within the system with and without leakages according to the fan pressure drop

The flowrate (Q) at the Air Terminal Devices (ATD) depends upon the pressure at the ATD according to a power law.

$$Q = C\,\Delta P^n \qquad (5)$$

C and n depend upon the air terminal device (n is close to 0.5).
Therefore, the lower the pressure drop at ATD's, the lower the flowrate.
Therefore, as shown in Figure 1, if the fan is not compensating for the additional pressure drop due to ductwork leakage, the pressure drop and flowrate at ATD's will decrease.

Generally, the pressure drop at a fixed ATD providing the hygienic flowrate is around 10 Pa. Self-adjusting ATD's generally need 50-70 PA to function properly.

2.3 Calculation method

To estimate the additional energy used to overcome ductwork leakage the additional flowrate and the additional pressure drop shall be calculated using the calculation model developed by (Leprince & Carrié, 2018), which is based upon EN 16798-5-1 (CEN, 2016).
If the fan compensates for leakages the flowrate at the fan shall be:

$$q_{v;ahu} = q_{v;dis;req} + q_{lea} \quad (6)$$

$$q_{lea} = A_{du} * c_{lea} * \Delta P_{du}^{ep} * 3600 \quad (7)$$

With

$q_{v;ahu}$	m^3/h	Required flowrate at Air Handling Unit
$q_{v;dis;req}$	m^3/h	Sum of required flowrates at Air Terminal Devices
q_{lea}	m^3/h	Flowrate through leakages
A_{du}	m^2	Area of the ductwork
c_{lea}	$m^3/s/m^2$ at 1Pa	Airtightness factor of the ductwork
ep	-	pressure difference exponent, default value: 0.65
ΔP_{du}	Pa	Average pressure difference between inside and outside the ductwork

Leakage only creates an additional pressure drop in the ductwork (not at the ATD), so to estimate the additional pressure drop due to ductwork leakage the pressure drop at the ATD's shall be deduced from the total pressure drop.

$$\Delta P_{fan} = \Delta P_{ATD;req} + \left(\frac{q_{v;dis;req} + q_{lea}}{q_{v;dis;req}}\right)^2 * \Delta P_{du;noleak} \quad (8)$$

ΔP_{fan}	Pa	Required pressure at fan to provide required pressure at ATD
$\Delta P_{ATD;req}$	Pa	Required pressure at ATD to provide required flowrate
$\Delta P_{du;noleak}$	Pa	Pressure drop in the ductwork when there are no leakages (when flowrate in the ductwork is the hygienic flowrate). This pressure drop does not include pressure drop at ATD.

To simplify the calculation and avoid cross-references, it can be assumed that ΔP_{du} is constant whatever the leakage is and equal to:

$$\Delta P_{du} = \Delta P_{ATD;req} + \frac{\Delta P_{du;noleak}}{2} \quad (9)$$

In this study, any leakage inside the AHU itself and the heat exchanger have been neglected to only show the impact of leakage in the ductwork system.

2.4 Impact of leakages on sound level

The more flowrate and pressure the fan is producing the more casing radiation (sound power) is produced by the fan.
For a given Air Handling Unit, the sound Power transmitted to the dwelling is measured in laboratory according to EN ISO 5135:1999 (Determination of sound power levels of noise from air-terminal devices, air-terminal units, dampers and valves by measurement in a reverberation room). Data are provided by manufacturers.
Usually in single-houses the fan is directly connected to a silencer, the sound attenuation of this device is laboratory tested according to ISO 7235 (Acoustics -- Laboratory measurement procedures for ducted silencers and air-terminal units -- Insertion loss, flow noise and total pressure loss).

The flow then goes through an air distribution box where the sound is split. The sound reduction depends upon the number of habitable rooms. If there are 2 rooms a reduction of 3 dB is assumed, if there are 4 rooms the reduction is of 6 dB.

To estimate the sound pressure level in the room the following equation is used:

$$L_p = L_w + 10 * \log\left(\frac{Q}{4\pi r^2} + \frac{4}{A}\right) \qquad (10)$$

With:

L_p	dB(A)	Sound pressure in the room
L_w	dB (A)	Sound power after the distribution box (neglecting the attenuation of ductwork)
Q	-	Coefficient depending of the angle of radiation, Q=2 for an Air Terminal Device on a wall
r	m	Distance to the source (r=1.5 m in the following example)
A	m²	Reference sound absorption area (20 m² Sabin for a furnished living-room and 8.5 m² Sabine for a small furnished bedroom)

3 RESULTS AND DISCUSSIONS : CASE STUDY

3.1 Fan energy use

Hypothesis

The following four scenarios have been simulated:

- House 1 is a medium-sized house with a central mechanical ventilation system with heat recovery. The ductwork system is a radial air distribution system using semi-rigid plastic ductwork. The diameter of the ductwork is 75mm and the total length is 125m. It is assumed that the ductwork is equally split between supply and extract.
- House 2 is also a medium sized house with a central mechanical ventilation system with heat recovery. The ductwork system is a trunk and branch air distribution system using metal or rigid plastic ductwork with 6m of ductwork DN160mm and 40m of ductwork DN125mm. It is assumed that the ductwork is equally split between supply and extract.
- House 3 is a large house with a central mechanical ventilation system with heat recovery. The ductwork system is a radial air distribution system using semi-rigid plastic ductwork. The diameter of the ductwork is 75 mm and the total length is 200 m. It is assumed that the ductwork is equally split between supply and extract.
- House 4 is a large house with a humidity-based extract only ventilation system, with self-adjusting ATD. The average flowrate is 100 m^3/h. The required pressure at the ATD is 70 Pa. The ductwork area is assumed to be 7.4m² (radial air distribution system).

Table 1 summarises the hypothesis of the ventilation system in each house used for the calculation.

Table 1: Hypothesis for cases studies

	House 1	House 2	House 3	House 4 (specific to the French market)
Hygienic flowrate (m^3/h)	**225**	**225**	**300**	**100**
Required pressure at ATD's (Pa)	**10**	**10**	**10**	**70**
Ductwork area of each airflow (m²)	**14.72**	**9.36**	**23.6**	**7.4**
Pressure drop in ductwork (without leakages) (Pa) for each airflow	**100**	**100**	**150**	**80**

Results

Table 2 shows the required flowrate and pressure of ***each*** airflow (supply and extract) and fan in each house and for the various airtightness classes. The required pressure at the fan includes the pressure drop in the ductwork plus the required pressure at the ATD's.

Table 2: Required pressure and flowrate for each fan according to the ductwork leakages for the 3 houses tested

Required flowrate of each fan (m^3/h)				
	House 1	House 2	House 3	House 4 (specific to the French market)
3*class A	286	264	424	146
1.5*class A	256	245	362	123
Class A	245	238	341	115
Class B	232	229	314	105
Class C	227	226	305	102
Class D	226	225	302	101
No leakage	225	225	300	100
Required pressure at each fan (Pa)				
3*class A	172	148	309	240
1.5*class A	139	128	228	191
Class A	129	122	204	176
Class B	116	114	174	158
Class C	112	111	165	153
Class D	111	110	162	151
No leakage	110	110	160	150

The fan power consumed to produce this pressure and flowrate can either be calculated by assuming a constant efficiency or read in the fan curves provided by the ventilation unit manufacturer.
The annual fan energy use shall be estimated assuming that the fans in both airflows work continuously that is to say 8,760 hours per year.
Table 3 shows the annual energy use of both fans assuming a constant fan efficiency of 0.27.

Table 3: Annual energy use of both fan (kWh) assuming an efficiency of 0.27

Annual energy use of both fans (kWh)				
	House 1	House 2	House 3	House 4 (specific to the French market)
3*class A	888	703	2359	315
1.5*class A	641	565	1488	211
Class A	571	523	1255	183
Class B	485	471	984	150
Class C	459	454	904	140
Class D	450	449	878	137
No leakage	446	446	865	135

Figure 2 and Figure 3 compare the annual energy use of both fans in the 4 houses according to the various airtightness classes. It shows that fans connected to leaky ductwork (3*Class A) in the four houses use 57-169% more energy than fans connected to very airtight ductwork (Class D), if they ventilate to provide the hygienic flowrate at Air Terminal Devices.

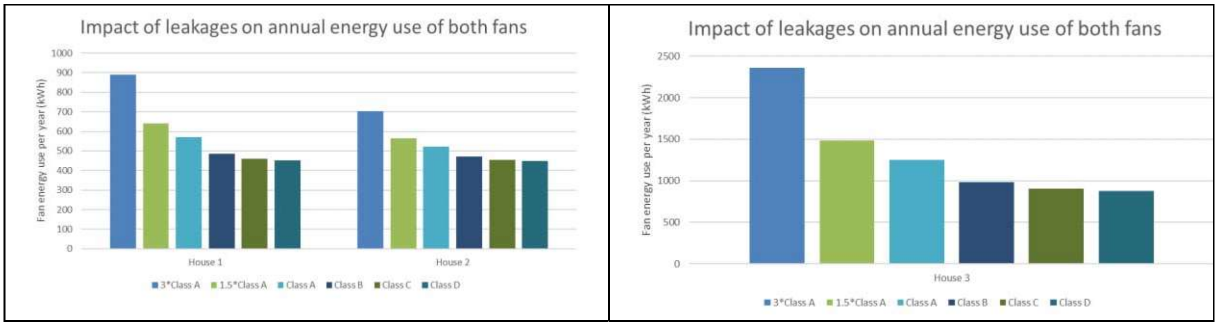

Figure 2: Annual energy use of both fans in houses 1 and 2 (left) and in house 3 (right) (estimated assuming a fixed fan efficiency of 0.27)

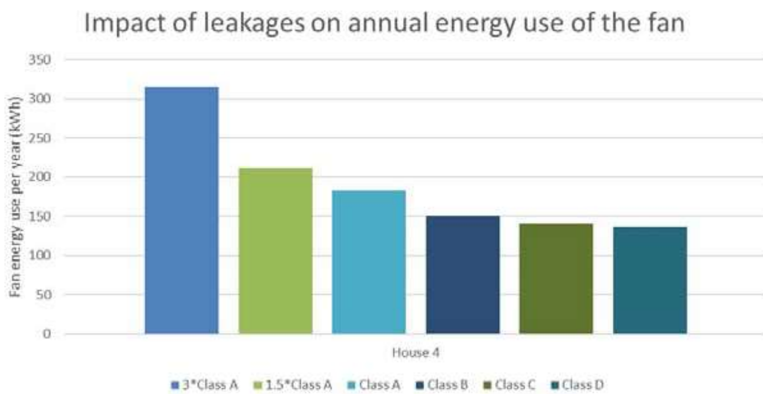

Figure 3: Annual energy use of the fan in house 4 (estimated assuming a fixed fan efficiency of 0.27)

3.2 Sound pressure

The sound pressure in bedrooms and living rooms has been calculated for houses 1 to 3 (with heat recovery system).

The sound power to dwelling (supply) measured according to ISO 5135 for the Residential Air Handling Unit Excellent 400 of Brink has been used (provided by manufacturer). The values are given in Table 4.

The sound reduction (or attenuation) of the silencer Brink (ISO AKS 1m, diam 160 mm) is provided for each frequency by the manufacturer. The impact on the total sound power is given in Table 4.

The sound power to each room and the sound pressure in bedrooms and living-rooms are calculated according to assumptions explained in § 2.4.

Table 4: Sound power in the ductwork and sound pressure in the rooms for the 3 houses for leaky and airtight ductworks

	HOUSE 1		HOUSE 2		HOUSE 3	
	3*Class A	Class D	3*Class A	Class D	3*Class A	Class D
Required flowrate (m^3/h)	286	226	264	225	424	302
Required pressure (Pa)	172	111	148	110	309	162
Sound power to dwelling (dB)	78.7	73.1	75.5	73.1	80.4	78.7
Sound power to dwelling with A correction and silencer dB(A)	43.9	38.5	41	38.5	46.8	43.9
Sound power to each room (- 6dB)	37.9	32.5	35	32.5	40.8	37.9

Sound pressure in bedrooms dB(A) according to equation (9)	35.2	29.8	32.3	29.8	38.1	35.2
Sound pressure in living rooms dB(A) according to equation (9)	32.2	26.8	29.3	26.8	35.1	32.2

<30 dB(A)	Quiet
30-35 dB(A)	Audible sound
>35 dB(A)	Loud

The impact is of 5.4 dB in the first house, 2.5 dB in the second and 2.9 dB in the third one, making the sound pressure from quiet to loud in house 1. In dB a difference of 3 dB corresponds to twice the noise level: it is equivalent to have two identical systems working at the same time.

4 CONCLUSION

The first part of this study has demonstrated the impact of ductwork leakage on both the flowrate and pressure drop at the fan. It has provided equations to calculate the impact according to

- the required hygienic flowrate
- the required pressure at ATD
- ductwork properties (surface area, leakage coefficients and pressure drop without leakage).

In, the second part of this study these equations were applied to central mechanical ventilation systems with heat recovery in three houses and with a single exhaust DCV system in one house. It has shown that fans connected to leaky ductwork (3*Class A) can use 57-169% more energy than fans connected to very tight ductwork (Class D) to produce the required hygienic flowrate.
If the fans have to work harder to produce the required hygienic flow rate, then they will produce more sound through the casing and in the ductwork and therefore noise hindrance. Calculation made from fan manufacturer data have shown a difference from 2.5dB(A) and up to 5.4 dB(A) in the habitable rooms according to the ductwork airtightness.

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6 REFERENCES

ASHRAE. (2013). *2013 ASHRAE Handbook: Fundamentals.*
Bailly, A., Duboscq, F., & Jobert, R. (2014). Impact of a poor quality of ventilation systems on the energy efficiency for energy-efficient houses. *35th AIVC Conference "*

Ventilation and airtightness in t (pp. 108-118). Poznań, Poland, 24-25 Septembe: AIVC.

Berthault, S., Boithias, F., & Leprince, V. (2014). Ductwork airtightness: reliability of measurements and impact on ventilation flowrate and fan energy consumption. *35 TH AIVC-4 TH TIGHTVENT & 2 ND VENTICOOL CONFERENCE , 2014* (pp. 478-487). Poznań, Poland, 24-25 September: AIVC.

Carrié, F., Bossaer, A., Andersson, J., Wouters, P., & Liddament, M. (2000). Duct leakage in European buildings: Status and perspectives. *Energy and Buildings, 32*(3), 235-243.

CEN. (2016). *EN 16798-5-1:2016 Energy performance of buildings - Ventilation of buildings - Part 5-1: Calculation methods for energy requirements of ventilation and air conditioning systems.*

Crowe, K. (1974). *A History of the Original Peoples of Northern Canada.* Montreal: McGill/Queen's University Press for the Arctic Insitute of North America.

D.F., D. (2011). Case study: effect of excessive duct leakage in a large pharmaceutical plant. *32th AIVC Conference. Towards Optimal Airtightness Performance* (pp. 55-56). Brussels, Belgium: AIVC.

Krishnamoorthy, S., & Modera, M. (2016). Impacts of duct leakage on central outdoor-air conditioning for commercial-building VAV systems. *Energy and Buildings, 119*, 340-351.

Leprince, V., & Carrié, F. (2018). Impact of ductwork airtightness on fan energy use: Calculation model and test case. *Energy and Buildings, 176*, 287-295.

Leprince, V., Carrié, F., & Kapsalaki, M. (2017). Building and ductwork airtightness requirements in Europe – Comparison of 10 European countries. *38th AIVC Conference. Ventilating healthy Low-energy buildings* (pp. 192-201). Nottingham, UK: AIVC.

Levinson, R., Delp, W., Dickerhoff, D., & Modera, M. (2000). Effects of airflow infiltration on the thermal performance of internally insulated ducts. *Energy and Buildings, 32*(3), 345-354.

Moujalled, B., Leprince, V., & Mélois, A. (2018). Statistical analysis of about 1300 ductwork airtightness measurement in new French buildings: impact of the type of ducts and ventilation systems. *39th AIVC conference.* Juan les pins.

Soenens, J., & Pattijn, P. (2011). Feasibility study of ventilation system air-tightness. *32th AIVC Conference. Towards Optimal Airtightness Performance* (pp. 51-54). Brussels, Belgium: AIVC.

Stroo, P. (2011). Class C air-tightness: Proven roi in black and white. *32th AIVC Conference. Towards Optimal Airtightness Performance2* (pp. 96-97). Brussels, Belgium: AIVC.

Zaslow, M. (1988). The Northward Expansion of Canada. *The Journal of Canada, 2*(3), 216-222.