Impact of ductwork airtightness on fan energy use: calculation model and test case

Valérie Leprince[*1], François Rémi Carrié[2]

1 PLEIAQ
84 C Avenue de la Libération
69330 Meyzieu, France
**Corresponding author: valerie.leprince@pleiaq.net*

2 ICEE
93 rue Molière
69003 Lyon, France

ABSTRACT

This paper proposes a methodology to assess fan energy use savings when improving ductwork airtightness. This methodology is based on new standard FprEN 16798-5-1:2016. Unlike the classical "cube law", it considers pressure drops at air terminal devices separately from the pressure drops in the rest of the system. The calculation tool based on this methodology: a) gives the fan energy use before and after airtightness improvements with various inputs depending on the initial state; b) indicates whether or not the required airflow rates are met at the air terminal devices; c) gives a range of energy savings assuming on the one hand perfect fan adjustment, no errors in input data and constant fan efficiency, and on the other hand, safe-side estimates of deviations from these assumptions.

Using experimental data from an earlier study on a scale 1 ductwork built in a laboratory, the tool gives results in good agreement with measured energy savings. The deviation without pressure adjustment between experimental and theoretical savings lies in the region of 5 percentage points. With ideal pressure adjustment and when required airflow rates were met before tightening, the maximum fan energy savings by tightening the ductwork from 1.5 times Class A to Class C reaches 51% according to the tool versus 46% according to measurement data.

KEYWORDS

Ductwork airtightness, fan energy use impact.

1 INTRODUCTION

There are a number of studies that demonstrate significant energy use impacts of ductwork leakage (Soenens & Pattijn, 2011), (Stroo, 2011), (Berthault, Boithias, & Leprince, 2014), (Dyer, 2011) and (Bailly, Duboscq, & Jobert, 2014) .

Nevertheless, there remain fundamental issues regarding the methods used to evaluate the energy savings of better ductwork airtightness. Designers and consultants commonly use the cube-law (or slight variants) to obtain estimates of the fan energy savings due to reduced airflow rates resulting from lower leakage rates. This law simply states that the fan power demand scales with the cube of the airflow rate. It is based on similarity principles assuming the fan serves a ductwork system with a constant opening. In this case, the pressure drop through the system scales with the square of the airflow rate. In practice, leakages only have an impact on the ductwork pressure drop and not on air terminal devices pressure drop therefore, in calculation, it is necessary to consider separately pressure drop in ductwork and pressure drop at ATD.

Another common implicit assumption when using the cube-law for duct tightening applications is that the fan is adjusted to deliver the required airflow rate at the air terminal devices. This may not be true with the fan in place if it is not strong enough and/or properly adjusted to compensate for the leaks in the system.

To our knowledge, this can result in very optimistic estimates of energy savings, which can be problematic both for the customer with much longer Return On Investment and for the reputation of the service provider. To avoid these problems, this paper details a methodology to give realistic but conservative energy-savings estimates.

2 OBJECTIVES

The main objectives of this study are:

a) to develop a methodology that gives realistic and conservative estimates of fan energy use impacts of duct airtightness improvements;
b) to develop a tool implementing the methodology;
c) to test the tool on real test case.

This methodology is consistent with standard FprEN 16798-5-1:2016.

3 THEORETICAL APPROACH

The methodology is structured around 4 methods outlined in Table 1. In short, while method 1 is firmly based on FprEN 16798-5-1:2016, method 2 considers some measured input data, rather than manufacturer data, which can be more accessible in a field study. Method 3 builds on method 2 but adjusts the fan to meet the required airflow rate and pressure needed at fan. Method 4 modifies the input data within confidence bounds defined by the user to give safe-side estimates. **Error! Reference source not found.** helps the user choose between the methods depending on the access to input data and the objective of the calculation.

Table 1: Outline of the 4 methods

Method	Short description	Pre-requisites	Limits / issues
Method 1	Based on the new standard FprEN 16798-5-1 methodology. Re-calculates fan and ductwork pressure (not ATD pressure) to calculate leakage airflow rate.	Requires knowledge of the fan properties and pressure at ATD (measurement or design value).	Does not include an adjustment of the fan (either pressure or flowrate): depending on the system, flowrate may exceed required flowrate at ATDs after refurbishment.
Method 2	Builds on method 1 but uses measured instead of manufacturer's data for the fan.	Requires measurements of pressure, flowrate and power at fan (also pressure measurement at ATD).	Likely less accurate than method 1 (measurement uncertainty) but easier to gather information. Does not include adjustment of the fan.
Method 3	Builds on method 2 but includes an adjustment of the fan to adapt flowrate and pressure set-point. Considers separately pressure losses at ATDs and pressure losses in ductwork.	Requires measurements of pressure, flowrate and power at fan (also pressure measurement at ATD).	Calculated set-point may not be consistent with the installed fan.
Method 4	Builds on method 3 but includes uncertainty in input data and conservative estimates of fan efficiency variations between operating points.	Requires measurements of pressure, flowrate and power at fan and estimation of uncertainty (also pressure measurement at ATD).	Method 4 is likely not relevant with large uncertainty estimates.

Table 2: Choice of the method

	Method 1	Method 2	Method 3	Method 4
Access to input data				
You have manufacturers data on installed fan	X			
You can perform measurement on site of flowrate, pressure and power at fan		X	X	X
Objective of the calculation				
You want to perform a calculation that comply with the standard	X	X		
You want to take into account gain due to pressure and flowrate adjustment			X	X
You want to estimate the maximal gain with air tightening (this may include fan change hypothesis)			X	
You want to perform a safe-side calculation	X	X		X

3.1 System representation and variables

The approach is based on the system representation proposed in Figure 1

Extract ductwork Supply ductwork

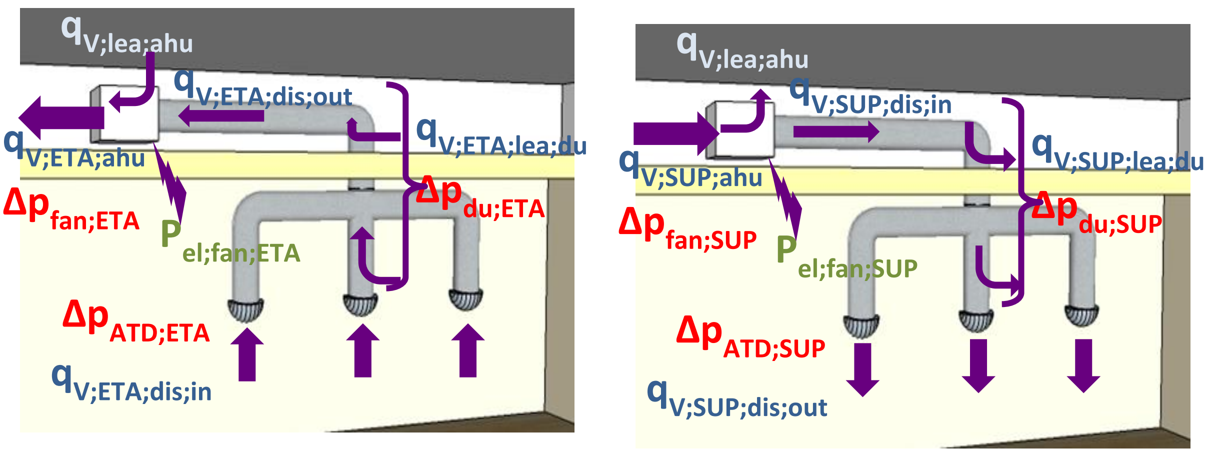

Figure 1: System representation

Table 3: Variables and subscripts used in the paper

Variable		Subscript	
Symbol	Meaning	Symbol	Meaning
A	Area	ATD	Air Terminal Device
c	leakage coefficient	ahu	Air handling unit
Pel	Electrical power	des	Design
qV	Volume airflow rate	dis	Distribution system
u	uncertainty	du	Duct
		el	Electrical
η	fan efficiency	ETA	Extract air
Δp	Pressure difference	fan	at fan
$\Delta p_{fan;step}$	Pressure step between two fan setting	in	entering the system
		lea	leakage
		mes	measured
		new	after tightening
		old	before tightening
		out	leaving the system
		req	required
		SUP	Supply air

Identifiers for the control of the fan are given in Table 4, it shall not be mingled with the control of the volume airflow rate in the ventilation system (Table 5) which is not necessarily made at fan.

Table 4: Identifiers for FAN_CTRL

SYS_TYPE	Identifier	Meaning
SINGLE_ZONE or MULTI_ZONE	NO_CTRL	No control: the fan reacts on the variable flowrate according to its characteristics
SINGLE_ZONE or MULTI_ZONE	CONST_PRES	The fan is controlled to achieve a constant pressure difference between a specific point in the distribution system and the surrounding
MULTI_ZONE	MIN_PRES	The fan is controlled to the minimum pressure difference necessary in the system
SINGLE_ZONE	DIRECT	The fan is controlled directly according to the variable flowrate

Table 5: Identifiers for AIR_FLOW_CTRL

Identifier	Meaning
NO_CTRL	No flow rate control, continuous operation
ON/OFF_CTRL	Time dependent flowrate control, continuous operation during occupancy time
MULTI_STAGE	Multi stage variable flowrate control
VARIABLE	Continuously variable flowrate control

3.2 Basic principles for all methods

All methods shown in this paper are firmly based on FprEN 16798-5-1:2016 whose aim is to evaluate the energy performance calculation of mechanical ventilation and air conditioning systems. The methods give estimates, before and after tightening the ductwork, of:

- the fan energy use;
- the overall airflow rates at the ATDs.

Yearly calculation

FprEN 16798-5-1 gives basic equations for one ventilation stage of operation. To assess the yearly performance of a system, the calculation has to be iterated for several stages of operation. Each of these stages is characterized by:

- A subset of input values common to all stages
- Another subset of input values specific to that stage (Table 6).

Annual cost estimates assume a fixed cost of electricity and without consideration for peak service subscription.

Table 6: Specific input values for each stage

Function of the dependence of the efficiency of the supply/extract fan on the volume flowrate derived from manufacturer's data, provided according ISO 5801	$f_{\eta}(q_V)$
Function for the dependence of the pressure difference over the supply/extract fan on the volume flowrate, derived from manufacturer's data, provided according ISO 5801	$f_{\Delta p}(q_V)$
Supply fan design pressure difference	$\Delta p_{fan;SUP;des}$
Extract fan design pressure difference	$\Delta p_{fan;ETA;des}$
Total required supply air volume flowrate (at air terminal devices) (sum for each ventilation zone)	$q_{V;sup;dis;req}$

Total flowrate to be extracted (at air terminal devices) (sum for each ventilation zone)	$q_{V;ETA;dis;req}$
Operation requirement signal	$f_{OP;V}$

Flowrate at air terminal devices
If there is either no control of the airflow rate or an on/off control (Table 5), the standard method does not adjust the fan according to duct leakages. Therefore, the power of the fan remains the same but the airflow rate at the air terminal devices may change. Note that in this case, the required flowrate may not be met. This requires both to compare energy use and to check actual airflow rate at ATDs before and after air tightening of the ductwork.

Use of default values
The methods use default values consistently with FprEN 16798-5-1 regarding the airtightness of the heat exchanger and of the air handling unit.

Pressure in the ductwork to calculate the leakage airflow rate
The leakage airflow rate is calculated with the ductwork airtightness class and pressure according to clause 6.3.2.2.2 of FprEN 16798-5-1. The duct pressure is:

$$\Delta p_{du;SUP/ETA} = \frac{\Delta p_{fan;SUP/ETA} + \Delta p_{ATD;SUP/ETA}}{2} \qquad (1)$$

$\Delta p_{fan;SUP/ETA}$ Pa Supply/Extract fan pressure difference
$\Delta p_{ATD;SUP/ETA}$ Pa Pressure drop at air terminal device at maximum flowrate (the farthest from the fan in terms of resistance to flow passage), on the supply/extract ductwork.

$\Delta p_{fan;SUP/ETA}$ and $\Delta p_{ATD;SUP/ETA}$ can be either a design value (method 1) or a measured value (methods 2, 3, 4).

3.3 Method 1
Apart from minor adjustments mentioned above, method 1 almost fully reproduces calculation performed in FprEN 16798-5-1:2016.

3.4 Use of measured values (methods 2, 3, 4)
Method 2, 3, 4 perform the same calculation as method 1 but with the measured input data described in Table 7 for each ventilation stage instead of detailed characteristics of air handling units.

Table 7: Measured values specific for Method 2, 3 and 4

Electrical fan power of the supply fan	$P_{el;fan;SUP;mes}$	Joule/h
Electrical fan power of the extract fan	$P_{el;fan;ETA;mes}$	Joule/h
Volume airflow rate supplied to the ventilation system (at ahu)	$q_{V;SUP;dis;in;mes}$	m^3/h
Volume airflow rate extracted from the ventilation system	$q_{V;ETA;dis;out;mes}$	m^3/h
Supply fan pressure difference	$\Delta p_{fan;SUP;mes}$	Pa
Extract fan pressure difference	$\Delta p_{fan;ETA;mes}$	Pa

Calculations before air tightening
The fan airflow rate and fan energy use are directly inferred from the measured values.
The difference between the measured fan airflow rate and the calculated leakage airflow rate gives the airflow rate at the ATDs, which is compared to the required airflow rate.

Calculation of the airflow rate after air tightening

The volume airflow rate supplied/extracted to the distribution system ($q_{V;SUP/ETA;dis;in}$) is calculated as follows:

If AIR_FLOW_CTRL = NO_CTRL (the airflow rate remains set to the measured airflow rate)

$$q_{V;SUP;dis;in} = q_{V;SUP;dis;in;mes} \quad (2)$$

$$q_{V;ETA;dis;out} = q_{V;ETA;dis;out;mes} \quad (3)$$

If AIR_FLOW_CTRL = ON/OFF_CTRL

$$q_{V;SUP;dis;in} = f_{op;V} * q_{V;SUP;dis;in;mes} \quad (4)$$

$$q_{V;ETA;dis;out} = f_{op;V} * q_{V;ETA;dis;out;mes} \quad (5)$$

If AIR_FLOW_CTRL = MULTI_STAGE

The standard's algorithms remain the same

If AIR_FLOW_CTRL = VARIABLE (the airflow rate adapts to the required airflow rate within the bounds of the fan curve, measured values are not used)

$$q_{V;SUP;dis;in} = f_{op;V} * min\left(max\left(q_{V;SUP;ahu;min}; q_{V;SUP;dis;in;req}\right); q_{V;SUP;ahu:max}\right) \quad (6)$$

$$q_{V;ETA;dis;out} = f_{op;V} * \min\left(max\left(q_{V;ETA;ahu;min}; q_{V;ETA;dis;out;req}\right); q_{V;ETA;ahu:max}\right) \quad (7)$$

where

$q_{V;SUP;ahu;min/max}$	m^3/h	minimum/maximum flowrate to be supplied by air handling unit
$q_{V;ETA;ahu;min/max}$	m^3/h	minimum/maximum flowrate to be extracted by air handling unit
$q_{V;SUP;dis;in;req}$	m^3/h	required airflow rate to be supplied at air handling unit
$q_{V;ETA;dis;out;req}$	m^3/h	required airflow rate to be extracted at air handling unit

Efficiency of the supply /extract fan before and after air tightening (methods 2 and 3)

The efficiency of the fan is:

$$\eta_{fan;SUP} = \frac{q_{V;SUP;dis;in;mes}}{P_{el;fan;SUP:mes}} * \Delta p_{fan;SUP;mes} \quad (8)$$

$$\eta_{fan;ETA} = \frac{q_{V;ETA;dis;out;mes}}{P_{el;fan;ETA;mes}} * \Delta p_{fan;ETA;mes} \quad (9)$$

3.5 Additional specificities of method 2

Supply/extract fan pressure difference after air tightening

This method does not include setting of the fan (see method 3 to include pressure and flowrate setting)

If FAN_CTRL=NO_CTRL

$$\Delta p_{fan;SUP/ETA} = \Delta p_{fan;SUP/ETA;0} - \left(\frac{q_{V;SUP/ETA;dis;in/out}}{q_{V;SUP/ETA;dis;in/out;mes}}\right)^2 * \left(\Delta p_{fan;SUP/ETA;0} - \Delta p_{fan;SUP/ETA;mes}\right) \quad (10)$$

where:

$\Delta P_{fan;SUP/ETA;0}$ Pa Pressure difference over the supply /extract fan if the air volume flow is 0 (1st key parameter determining the fan characteristic curve). If it is unknown it can be determined with a two-point measurement using equation 10

$\Delta P_{fan;SUP/ETA;mes}$ Pa Measured pressure difference over the supply/extract fan before air tightening

Note that this formula is not in the standard, but is included in FprEN 16798-5-1 excel sheet.

If FAN_CTRL= DIRECT (only for SINGLE_ZONE)

$$\Delta p_{fan;SUP/ETA} = \left(\frac{q_{V;SUP/ETA;ahu}}{q_{V;SUP/ETA;dis;in/out;mes}}\right)^2 * \Delta p_{fan;SUP/ETA;mes} \quad (11)$$

If FAN_CTRL= CONST_PRES

$$\Delta p_{fan;SUP/ETA} = \left[\left(1 - f_{\Delta p;SUP/ETA;ctrl}\right)\left(\frac{q_{V;SUP/ETA;ahu}}{q_{V;SUP/ETA;dis;in/out;mes}}\right)^2 + f_{\Delta p;SUP/ETA;ctrl}\right] * \Delta p_{fan;SUP/ETA;mes} \quad (12)$$

If FAN_CTRL=MIN_PRES

$$\Delta p_{fan;SUP/ETA} = \left[\left(1 - f_{\Delta p;SUP/ETA;ctrl}\right)\left(\frac{q_{V;SUP/ETA;ahu}}{q_{V;SUP/ETA;dis;in/out;mes}}\right)^2 + f^2_{\mathrm{V;max;SUP/ETA}} * f_{\Delta p;SUP/ETA;ctrl}\right] * \Delta p_{fan;SUP/ETA;mes} \quad (13)$$

Where

$f_{\Delta p;SUP/ETA;ctrl}$ - Controlled portion of the total supply/extract design pressure difference

$f_{V;max;SUP/ETA}$ - Maximum part load factor of the zone air volume airflow rates

3.6 Additional specificities of method 3

Method 3 adjusts fan pressure and flowrate set-point after air tightening to provide the required pressure and flowrate at air terminal devices.

Air volume flow rate at air handling unit after air tightening

Assuming perfect adjustment of the airflow rate is equivalent to assume that the AIR_FLOW_CTRL is VARIABLE with no maximum limit. Therefore, in this method, for all values of AIR_FLOW_CTRL:

$$q_{V;SUP;dis;in} = f_{op;V} * max\left(q_{V;SUP;ahu;min}; q_{V;SUP;dis;in;req}\right) \quad (14)$$

$$q_{V;ETA;dis;out} = f_{op;V} * max\left(q_{V;ETA;ahu;min}; q_{V;ETA;dis;out;req}\right) \quad (15)$$

Supply/extract fan pressure difference

Air leakages increase the ductwork pressure drop. Therefore, after air tightening, the fan pressure can be reduced. The new setting for fan pressure is calculated for the maximum required flowrate at the ATDs for all stages of operation.

$$\Delta p_{fan;SUP/ETA;set} = \Delta p_{ATD;SUP/ETA;req} + \left(\frac{q_{V;SUP/ETA;dis;in/out;max}}{q_{V;SUP/ETA;dis;in/out;mes;max}}\right)^2 * \left(\Delta p_{fan;SUP/ETA;mes;max} - \Delta p_{ATD;SUP/ETA}\right) \quad (16)$$

Where:

$q_{V;SUP/ETA;dis;in/out;mes;max}$ m^3/h maximum measured flowrate maximum $q_{V;SUP/ETA;dis;in/out;mes}$ in the variable inputs) for supply/extract

$\Delta p_{fan;SUP/ETA;mes;max}$ Pa Corresponding measured pressure at fan

$q_{V;SUP/ETA;dis;in/out;max}$ m^3/h Corresponding maximum flowrate at air handling unit after refurbishment

$\Delta p_{fan;SUP/ETA;req}$ Pa Required pressure drop at ATD

$\Delta p_{fan;SUP/ETA}$ Pa Measured pressure drop at ATD

Therefore,

If FAN_CTRL=NO_CTRL

$$\Delta p_{fan;SUP/ETA} = \Delta p_{fan;SUP/ETA;0} - \left(\frac{q_{V;SUP/ETA;dis;in/out}}{q_{V;SUP/ETA;dis;in/out;max}}\right)^2 * \left(\Delta p_{fan;SUP/ETA;0} - \Delta p_{fan;SUP/ETA;set}\right) \tag{17}$$

If FAN_CTRL= DIRECT (only for SINGLE_ZONE)

$$\Delta p_{fan;SUP/ETA} = \left(\frac{q_{V;SUP/ETA;dis;in/out}}{q_{V;SUP/ETA;dis;in/out;max}}\right)^2 * \Delta p_{fan;SUP/ETA;set} \tag{18}$$

If FAN_CTRL= CONST_PRES

$$\Delta p_{fan;SUP/ETA} = \left[\left(1 - f_{\Delta p;SUP/ETA;ctrl}\right)\left(\frac{q_{V;SUP/ETA;dis;in/out}}{q_{V;SUP/ETA;dis;in/out;max}}\right)^2 + f_{\Delta p;SUP/ETA;ctrl}\right] * \Delta p_{fan;SUP/ETA;set} \tag{19}$$

If FAN_CTRL=MIN_PRES

$$\Delta p_{fan;SUP/ETA} = \left[\left(1 - f_{\Delta p;SUP/ETA;ctrl}\right)\left(\frac{q_{V;SUP/ETA;dis;in/out}}{q_{V;SUP/ETA;dis;in/out;max}}\right)^2 + f_{\mathrm{V;max;SUP/ETA}}^2 * f_{\Delta p;SUP/ETA;ctrl}\right] * \Delta p_{fan;SUP/ETA;set} \tag{20}$$

3.7 Additional specificities of method 4

Method 4 performs the same calculation of method 3 but modifies the following input parameters to obtain a conservative estimate of energy savings:

- Measured inputs
- Fan efficiency
- Fan pressure setting

The user give a range of uncertainty for these input values, which are re-calculated as follows:

$$A_{du;SUP;met4} = A_{du;SUP} * (1 - u_{Adu}) \tag{21}$$
$$A_{du;ETA;met4} = A_{du;ETA} * (1 - u_{Adu}) \tag{22}$$
$$c_{lea;du;SUP;old;met4} = c_{lea;du;SUP;old} * (1 - u_{Cleaold}) \tag{23}$$
$$c_{lea;du;ETA;old;met4} = c_{lea;du;ETA;old} * (1 - u_{Cleaold}) \tag{24}$$
$$c_{lea;du;SUP;new;met4} = c_{lea;du;SUP;new} * (1 + u_{Cleanew}) \tag{25}$$
$$c_{lea;du;ETA;new;met4} = c_{lea;du;ETA;new} * (1 + u_{Cleanew}) \tag{26}$$
$$\Delta p_{ATD;SUP;met4} = \Delta p_{ATD;SUP} * \left(1 + u_{pATD}\right) \tag{27}$$
$$\Delta p_{ATD;ETA;met4} = \Delta p_{ATD;ETA} * \left(1 + u_{pATD}\right) \tag{28}$$
$$q_{V;SUP;dis;req;met4} = q_{V;SUP;dis;req} * \left(1 + u_{qV;dis;req}\right) \tag{29}$$
$$q_{V;ETA;dis;req;met4} = q_{V;ETA;dis;req} * \left(1 + u_{qV;dis;req}\right) \tag{30}$$
$$P_{el;fan;SUP;mes;met4} = P_{el;fan;SUP;mes} * \left(1 - u_{Pelfan}\right) \tag{31}$$
$$P_{el;fan;ETA;mes;met4} = P_{el;fan;ETA;mes} * \left(1 - u_{Pelfan}\right) \tag{32}$$
$$q_{V;SUP;dis;in;mes;met4} = q_{V;SUP;dis;in;mes} * \left(1 - u_{qV;mes}\right) \tag{33}$$
$$q_{V;ETA;dis;out;mes;met4} = q_{V;ETA;dis;out;mes} * \left(1 - u_{qV;mes}\right) \tag{34}$$
$$\Delta p_{fan;SUP;mes;met4} = \Delta p_{fan;SUP;mes} * \left(1 - u_{pfan;mes}\right) \tag{35}$$
$$\Delta p_{fan;ETA;mes;met4} = \Delta p_{fan;ETA;mes} * \left(1 - u_{pfan;mes}\right) \tag{36}$$

37)

Efficiency of the supply/extract fan

$$\eta_{fan;SUP} = \left(\frac{q_{V;SUP;dis;in;mes;met4}}{P_{el;fan;SUP:mes;met4}} * \Delta p_{fan;SUP;mes;met4}\right) * \left(1 - u_{fan;eff}\right) \tag{38}$$

$$\eta_{fan;ETA} = \left(\frac{q_{V;ETA;dis;out;mes;met4}}{P_{el;fan;ETA;mes;met4}} * \Delta p_{fan;ETA;mes;met4}\right) * \left(1 - u_{fan;eff}\right) \quad (39)$$

Supply and extract fan pressure difference

$$\Delta p_{fan;SUP/ETA;set} = \Delta p_{ATD;SUP/ETA;req} + \left(\frac{q_{V;SUP/ETA;dis;in/out;max}}{q_{V;SUP/ETA;dis;in/out;mes;max} * \left(1 - u_{qV;mes}\right)}\right)^2 * \left(\Delta p_{fan;SUP/ETA;mes;max} * \left(1 - u_{pfan;mes}\right) - \Delta p_{ATD;SUP/ETA;met4}\right) + \Delta p_{fan;step} \quad (40)$$

4 LABORATORY EXPERIMENTS

Figure 2: Laboratory replication of real ductwork system in Autun (France). Source : (Berthault, Boithias, & Leprince, 2014)

Berthault et al. (Berthault, Boithias, & Leprince, 2014) built an extract air duct system similar in size and ventilation characteristics to ductwork systems commonly found in France in multi-family buildings to study, among other things, the impact of ductwork airtightness on fan energy use. The facility consists in an 18.7 m² ductwork connected to a constant pressure extract fan, eight self-adjusting Air Terminal Devices. Four of them can be manually set to a minimum and maximum airflow rate. The minimum and maximum airflow rates are 260 m^3/h and 425 m^3/h, respectively. The ductwork was initially class C but holes were intentionally drilled to reach 1.5 class A. Power, flowrate, and pressure measurement at fan were performed before and after drilling.

The fan is na EC motor, with an impeller with forward blades and direct drive. Fan power for pressure and airflow rates used in the study are given in Table 8

Table 8: Fan properties

Airflow rate	Pressure	Power
425 m3/h	120 Pa	63 W
260 m3/h	120Pa	40 W

We have compared the results obtained by measurements with those obtained with our methodology. Note that in our calculation, consistently with the fan characteristics, the fan control is set to "constant pressure"; the airflow control is set to "variable" because the self-adjusting ATDs are maintaining the flowrate on a large scale of pressure.

5 RESULTS

Table 9 shows the annual operating energy cost savings. Absolute cost savings are low as it is a small ductwork, but methods 3 shows that tightening the ductwork from 1.5 · Class A to class C can reduce energy use up to 53% (with a suitable fan adjustment).

Table 9: Theoretical and experimental cost savings on test case, improving the airtightness from 1.5 · Class A to Class C. Maximum and minimum airflow rates are used 2 and 22 hours per day, respectively

Results based on	Cost before retrofitting	Required airflow rate met before retrofitting	Cost after retrofitting	Required airflow rate met after retrofitting	Cost savings in €	Cost savings in %
Fan power measurements (no fan adjustment)	68€	Yes	61€	Yes	- 7€	-11%
Method 1	71 €	Yes	60 €	Yes	- 11 €	- 15%
Method 2	68 €	Yes	57 €	Yes	- 11 €	- 16%
Method 3	68 €	Yes	32 €	Yes	- 36 €	- 53%
Method 4	61 €	Yes	47 €	Yes	- 14 €	- 23%
Fan power measurements (with fan adjustment)	68€	Yes	47€	Yes	- 21€	- 31%
Cube-law	68€	Yes	40€	Yes	-28€	-59%

Figure 3 and Figure 4 compare the economy planed by the tool for minimum and maximum flowrate with measurement results with and without pressure adjustment. The darkest part of each bar represents the power difference between 1.5 · Class A and Class C. The left bar represents measurement data with no pressure adjustment (pressure is maintained at 120 Pa, to be compared with methods 1 and 2), whereas the right bar gives measurement data with pressure adjustment: the pressure is at 120 Pa for 1,5 · Class A and 90 Pa for Class C (to be compared with methods 3 and 4).

At the maximum airflow rate (Figure 3), the results of Method 1 and 2 are in good agreement with measurement data with no fan pressure adjustment (120 Pa). The measurement data with fan pressure adjustment (90 Pa) for the tightest system shows that the measured energy savings are very close to the energy savings estimated with method 3, which is likely close to the maximum savings achievable given its assumptions. Conversely, method 4 deviates more from the experimental results with a conservative estimate of 15%.

At the minimum airflow rate (Figure 4), the deviations are larger due to the very low differences in fan power demand induced by air tightening: an absolute difference of 1 W corresponds to a relative difference of 2 percentage points. Note also that the fan pressure could not be adjusted below 90 Pa although less pressure was needed. Therefore, method 3 overestimates the savings, but the measured data remains within the range given by method 4 and 3.

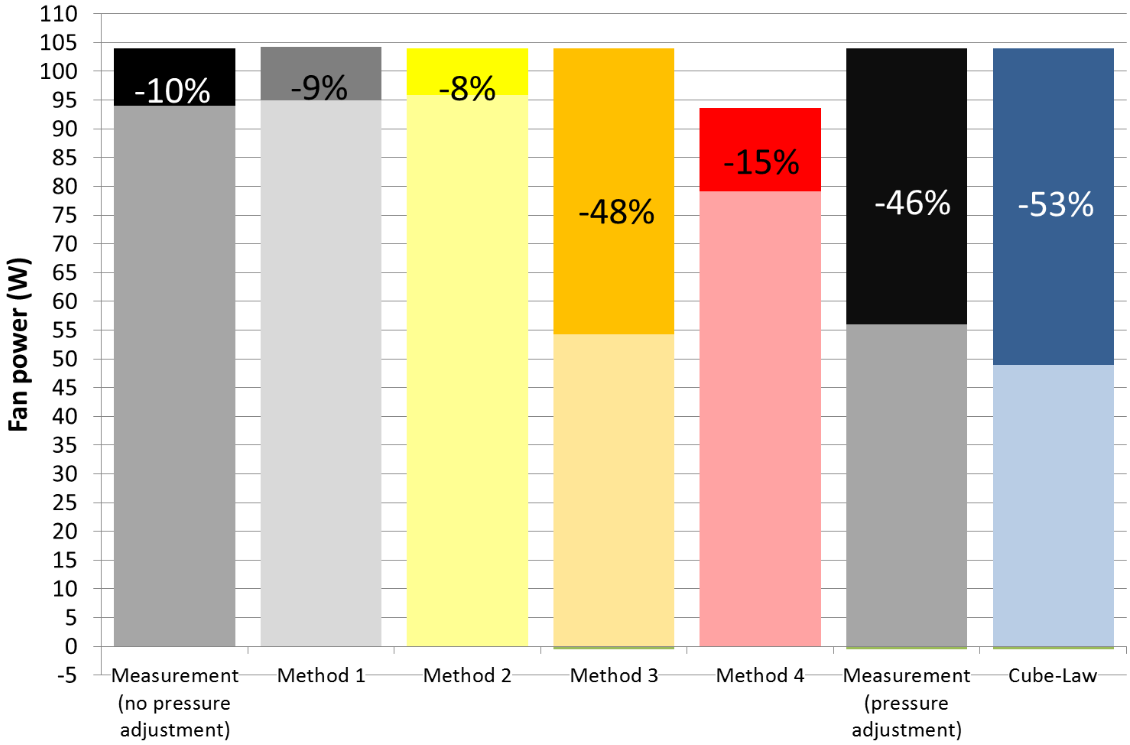

Figure 3: Theoretical and experimental cost savings on test case, improving the airtightness from 1.5 · Class A to Class C. Results for maximum airflow rate.

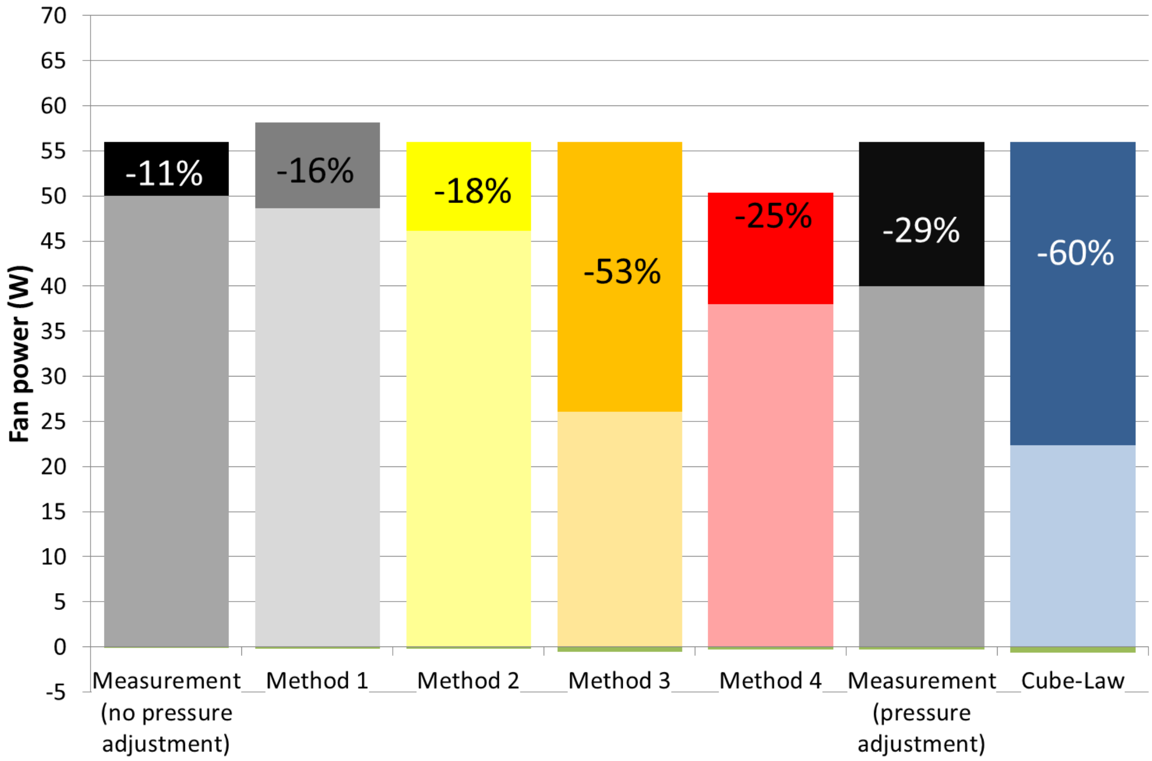

Figure 4: Theoretical and experimental cost savings on test case, improving the airtightness from 1.5 · Class A to Class C. Results for minimum airflow rate.

6 DISCUSSION

The methodology includes 4 methods that can be used depending in particular on the accessibility to the fan characteristics input data and the objective of the calculation.

Method 1 uses theoretical input data, however, when dealing with existing installations, it may be difficult to obtain accurate information on the air handling unit (nominal efficiency of the fan, function of the dependence of the pressure difference over the supply extract fan, etc.). Method 1 assumes that the fan has been tested according ISO 5801, and that characteristics obtained with this standard are available. If not, the user could guess this input data to use method 1, but experience shows that this can be very challenging and inappropriate for the desired level of accuracy of the fan energy savings, in particular with old installations. In our experience, method 2 is more appropriate in such cases.

Method 1 includes airflow rate adjustment only with a variable or multistage airflow control. Therefore, when there is either no control or an on/off control of the airflow rate in the ductwork system, leakages reduce the airflow rate at the ATDs but have no impact on the fan energy use. This is not strictly true because the fan sees a slightly different pressure drop before and after sealing if the leaks are close to the fan; however, we expect this has a minor impact and only fan adjustment can give significant energy savings.

To overcome these limitations, methods 2-3-4 introduce a subset of measured input data substituting the theoretical input data. Method 2 has the same limitations as method 1 regarding the adjustment of the fan. Because fan adjustment after air tightening (new speed, new pressure set point, etc.) is desirable and may be suggested by a consultant to his client, method 3 assumes a “perfect” fan adjustment to meet the required airflow rates at the ATDs. To avoid unpleasant surprises in terms of actual energy savings, method 4 introduces factors on some of the input data to perform a "safe-side" calculation. For instance, it assumes that the fan efficiency decreases with decreasing airflow rates (unlike method 1, 2, 3 that assumes constant fan efficiency).

Note that, similarly to the cube-law, methods 3 and 4 adjust the fan set-point considering the impact of leakages both on airflow rate and pressure drop; however, the fundamental difference are that methods 3 and 4:

- take into account separately the pressure drop in the ductwork and the pressure drop at air-terminal devices. Only the post-tightening pressure drop in the ductwork is assumed to be proportional to square of the airflow rate (whereas the cube law assumes the pressure at the fan is proportional to the square of the airflow rate). The pressure at the ATD is set after tightening independently of the leakages;
- the energy use is calculated to ensure required airflow rates and pressures are met after refurbishment. If the measured initial airflow rates are insufficient, the tool adjusts these boundary conditions. In such cases, air tightening and fan adjustment may result in no or even increased energy use to obtain the required airflow rates at the air terminal devices.

Because method 3 calculates energy use assuming the fan is set at the optimum calculated set-point, it gives the maximum savings achievable with ductwork air tightening with the fan in place. Note, however, that the optimum set-point is calculated based on the equations provided in FprEN 16798-5-1. As a result, this set-point may not be reachable with the existing fan, for instance, because it is outside its range of operation or because of the fan behaviour deviates from the assumed behaviour in the standard.

Method 4 calculates the minimum economy assuming that "everything that can go wrong will go wrong", i.e., using the worst possible value of each input within a range defined by the user. Therefore, it is overly pessimistic in terms of achievable energy savings, in particular, if the user cannot define reasonable deviation ranges for the input values. It would be interesting to refine this analysis with error propagation or a stochastic approach to give more realistic results, although the selection of proper ranges of deviations would remain an issue.

Nevertheless, on the test case, theoretical and field results are in good agreement both with and without fan adjustment. Also, method 1 (based on manufacturer's data) and method 2 (based on measurement) give consistent results. The energy savings in our test case reach 53%. These significant energy savings are possible because the required flowrate were met before retrofitting.

As expected, the cube-law overestimates the energy savings with 53% at maximum flowrate (versus 46% measured) and 60% at minimum flowrate (versus 29% measured). In this specific experiment, because we had the required airflow rates at the ATDs, the deviations between the cube law and method 3 are rather small. Otherwise, they would be much larger because the airflow rate would be first transferred from the leakages to the ATDs before any reduction of the fan airflow rate can occur.

One limitation of this study is that the methodology does not account for the impact of ductwork leakages on heating, cooling and humidity losses. Energy losses due to over-ventilation can be estimated based on:

- The leakages located in the conditioned zone;
- The increased ventilation airflow rates due to leakages, using equations in FprEN 16798-5-1. If the required airflow rates at ATD are not met pre-tightening, the leakage airflow rate needs to be first transferred from the leakages to the ATDs before any reduction of the ventilation airflow rate can occur in the conditioned zone;
- The indoor temperature during the heating period.

Nevertheless, when air is preconditioned, the impact of leakage is difficult to calculate as it depends in particular on:

- whether the heat losses through the leaks are completely, partly or not recoverable;
- the part of the energy demand for space conditioning the building that has to be provided by the ventilation system.

7 CONCLUSIONS

The methodology developed in this paper allows one to assess fan energy savings resulting from ductwork airtightness improvement scenarios, because of the decrease of the airflow rate and pressure drops as the ductwork system gets tighter. The methodology does not require inputs related to the building energy performance, except for those characterizing the ventilation system energy use. Although consistent with FprEN 16798-5-1:2016, the methodology includes a number of new equations to substitute fan manufacture's input data with field data and to take into account pressure drops at air terminal devices separately from the pressure drops in the rest of the system. The methods developed assess energy savings with various sub-sets of input data depending on the initial state, including whether or not the required airflow rates were met. The methodology provides a range of energy savings resulting from different scenarios (e.g., no or perfect fan adjustment) including the accessibility and reliability of the input data (e.g., uncertainties in measured pressures, airflow rates, and fan power demand) to have safe-side estimates.

Measurement data on an experimental ductwork with an airtightness class of 1.5 · Class A and Class C is in the range of values given by our methodology. The deviation without pressure adjustment between experimental and theoretical savings lies in the region of 5 percentage points. With ideal pressure adjustment and when required airflow rates were met before tightening, the maximum fan energy savings by tightening the ductwork from 1.5 · Class A to Class C reaches 53% according to the tool versus 46% according to measurement data.

The results obtained are promising and thereby support using this methodology for assessing fan energy use impacts of ductwork leakage. This could be useful either to designers or consultants willing to explore different ductwork airtightness scenarios in new systems or the relevance of duct-sealing measures in existing systems. Ductwork leakages have also a significant impact on heating, cooling and humidity losses; however, a more global approach to the building-system interaction seems necessary to address this issue. It would be worth investigating how this could done without a complex energy performance of building simulation.

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