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Using Measured Exponents and Fractional Leakage Flow to Characterize Air Leakage in Ducts

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ABSTRACT

The issue of leakage in air distribution systems is one of the most underappreciated efficiency improvement opportunities, even though potential improvements are significant. One of the main reasons for this lack of awareness lies in the current ineffective practices for determining and addressing the energy consequences due to poor tightness. The airtightness of ducts is generally only evaluated based on classifications (e.g. leakage class or seal class), which are in turn used to estimate duct leakage flows during normal operation using an assumed operating pressure, and a fixed flow exponent of 0.65. This paper presents technical arguments for why it is worthwhile to characterize duct leakage with measured flow exponents and operating pressures, or with direct measurements of leakage flows during normal operation. The focus is non-residential buildings, for which there are large variations in duct pressures during normal operation, between different duct sections, and between testing/classification values and normal-operation values. Experimental evidence is presented to show that the current methodology is not suitable for describing the real behavior of ductwork. Two alternative methodologies are described, both related to expressing leakage in percentage terms with respect to the nominal flow rate. The difference between the two indexes is that the first one utilizes the estimated leakage at operating pressure, the other indicates the maximum pressure at which the system can operate in order to not exceed a predetermined percentage leakage value.

INTRODUCTION

The subject of air duct sealing has been considered from a quantitative point of view from the first half of the 1960s. During the 1970s and 1980s [1] some measurement campaigns were carried out by SMACNA, ASHRAE and TIMA [2,3] and because of those campaigns, a number of reference standards were published in the United States and Europe (SMACNA HVAC Air Duct Leakage Test Manual, EUROVENT 2/2, DW/143 [4–6]. At that time, the critical aspect was not so much the quantification of the leakage impacts, but rather the classification of leakage normalized by duct surface area (L/s per m^2 or cfm/100ft). These documents contain the basis of the classification procedures in current Standards, although there have been some classification and nomenclature changes [7–9]. The general reference model for air flow through duct leaks is the power law expressed in the following form:

$$m_{exf} = f \times A \times \Delta P^n \quad (1)$$

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Where A is the surface area of the duct section, and the leakage parameters f and n depend on the fluid dynamics at the leak. In the original references the value of the exponent n was found to vary between 0.5 and 0.93 with an average value of 0.65 [1,3] consistent with the 0.5 to 1 range predicted by fluid dynamics theory. Presumably to simplify test procedures, it has become common practice to use a constant exponent (n) of 0.65. By assuming a constant exponent, the other leakage parameter f can be determined by means of a single-pressure measurement. On the basis of this simplified model, a test called the Duct Air Leakage Test (DALT) was defined, which consists of isolating a section of the duct system with end caps and shut-off plates, bringing the section to a predefined pressure (positive or negative) by inflating or extracting air by means of an external fan, and measuring the compensation air flow rate. Knowing the duct surface area, the parameter f can be calculated according to the model (1). Nevertheless, issues arise if the actual flow exponent of the system is not equal to 0.65; it can be analytically proven [10] that in this case the f-value obtained by the measurement is not constant but rather is dependent on the test pressure value. Thus, the standards adopting the constant-n model must also specify reference pressure values at which to perform the classification tests.

However, even the original Standard [4] gives examples of how to quantify leakage in terms of absolute flow and/or in terms of percentage of design flow. The need to quantify leakage during normal operation is also made explicit in more recent references such as the 2024 ASHRAE Systems And Applications Handbook [11] which, in the chapter dedicated to duct tightness, states that "*to enable proper accounting of leakage related impacts on fan energy use and space conditioning loads, the design engineer should establish the maximum allowable system leakage for each fan system as a percentage of fan airflow at a reference system operating condition. Recommended test pressures and maximum system leakages fractions are indicated in Table 2.*"; the cited table lists types of air systems for which a maximum leakage rate of 5% is generally required, with exceptions for ducts located outdoors (2%) and air handling units (1%).

The need to quantify the energy lost due to air leakage is also mentioned in the 2024 revision of the European Directive referred to as the Energy Performance of Building Directive (EPBD); Standard EN 16798-5 [12] indicates a simplified procedure for estimating the absolute exfiltrated or infiltrated flow rate through the application of the simplified model where the f-value is assumed to be constant and taken from the classification test's results, the pressure is assumed to be the average operating pressure of the system, and the duct surface area is calculated using a simplified procedure according to EN14239 [13].

In 2018 ASHRAE published Standard 215:2018 [14], which standardizes direct measurement of leakage during operation from the difference between the flow rate measured at the terminals and the flow rate measured at the fan or an upstream duct cross section. The focus within the standard on quantifying the uncertainty in the measured value illustrates the technical care and proficiency required by this methodology [15], suggesting that this method may be difficult to apply on a large scale. The purpose of this paper is to examine some additional possibilities for quantifying and reporting the expected leakage under operating conditions based upon applying the leakage model with a measured flow exponent.

METHODOLOGIES

Simplified Leakage Model versus Extended Leakage Model

The current simplified model requires only a single measurement; however, it is limited by the fact that its accuracy drops off whenever the operating pressure varies from a small range around the test pressure used for that measurement. On the other hand, the proposed extended model requires multiple measurements but provides more accurate estimates of leakage flow over a large operating pressure range.

To illustrate the differences between the two models, measurements on a circular-metal primary air system serving two-story office building are investigated. For the purposes of these comparisons the factor f for the simplified model will be called $f_{0.65}$, while the factors for the extended model will be called f_{ext} and n_{ext}. A branch of the system was tested through a series of measurements performed at 10 different pressures from 50Pa up to 500Pa. From these measurements, the logarithmic linear regression technique [16] was applied and the proportionality coefficient $(f \times A)$ as well as the exponent n were calculated, as was the coefficient of determination R^2, resulting in $(f_{ext} \times A) = 4.99$, $n_{ext} = 0.51$, and $R^2 = 0.999$. It should be noted that it is not necessary to split the proportionality coefficient (fxA) into the two factors f and A when estimating leakage flow for the test section. This split is only necessary when the leakage factor f is to be determined for classification purposes, and for extrapolation to other duct sections.

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As 500Pa is the test pressure required for the classification of circular sheet metal ducts by means of single-point measurement [17], the leakage factor corresponding to the simplified model was calculated at that pressure resulting in $(f_{0.65} \times A) = 2.13$, and the leakage flow over the entire pressure range was estimated. Figure 1 shows the different behavior of the two models at low pressures.

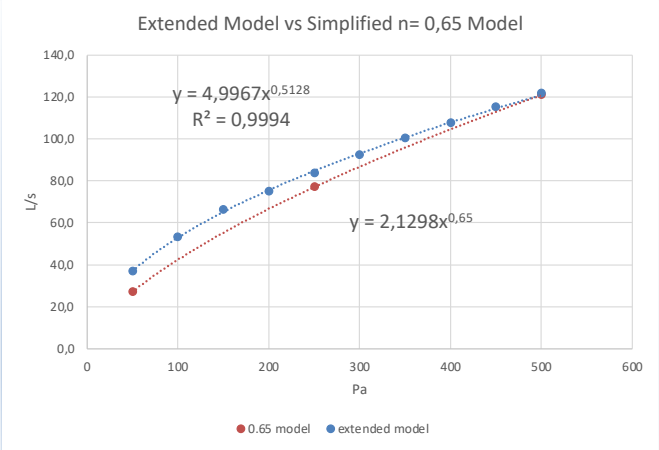

Figure 1: Differences between the extended model characterized by a multi-point measurement and the simplified model characterized by a single-point measurement at 500Pa.

It is important to note that the value of $(f \times A)$varies by almost a factor of 2 between the two models. This is very important, since A is the same for both, and thus the value of f is very different, which means that applying the result to other parts of a duct system will be very different for the two models. The comparison at lower pressure between leakage estimated by the 0.65 model and the extended model shows that the simplified model underestimates the leakage by 8% at 250 Pa and by 27% at 50 Pa. More generally, this different behavior was analyzed in a paper from 2024 [10] which illustrates how the calculation of the leakage reported at 100 Pa using the simplified model with $f_{0.65}$ referred to a 500Pa test leads to an underestimation of leakage by 19% when applied to a system whose actual exponent is 0.5, and overestimates leakage by 47% when applied to a system whose actual exponent is 0.93.

Moving away from the 0.65-Model

When it is necessary to estimate leakage under operating conditions, it must be considered that the pressure inside the ducts varies from the maximum value (at the fan) to the minimum value (at the grilles or diffusers) due to friction losses. Although the pressure variation upstream of VAV (Variable Air Volume) boxes in a variable flow system is in general modest, the pressure across leaks downstream of those boxes varies dramatically. It is self-evident that a suitable model for this type of calculation must be applicable over the entire pressure range, and therefore the extended model is much more suitable for determining these leakage flows.

The impact of an assumed versus a measured exponent can be compared using the f factor. The test-section surface area was estimated from construction drawings, resulting in $A = 10.3\ m^2$, and $f_{n=0.65}$ can be calculated using:

$$f_{n=0.65} = \frac{m_{exf}}{A \times \Delta P^{0.65}} \qquad (2)$$

The parameters f_{ext} and n_{ext} are obtained through the multiple measurements and linear logarithmic regression analysis. In this case, Equation (2) can be rewritten using n_{ext} instead of 0.65 and f factor can then calculated at any pressure:

$$f_{n=n_{ext}} = \frac{m_{exf}}{A \times \Delta P^{n_{ext}}} \qquad (3)$$

The f-factors obtained using Equations (2) and (3) are plotted in Figure 2. Figure 2a plots Equation (2), showing the

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value of $f_{0.65}$ that would be obtained by making the single point measurement at pressures lower than the classification pressure in EN 12337 (500Pa). Figure 2b plots Equation (3) over the same pressure range. It is clear that $f_{0.65}$ varies at each pressure point while f_{ext} remains essentially constant. It should also be noted that the range in which $f_{0.65}$ varies (0.21 to 0.28) does not include the f_{ext} value (0.48).

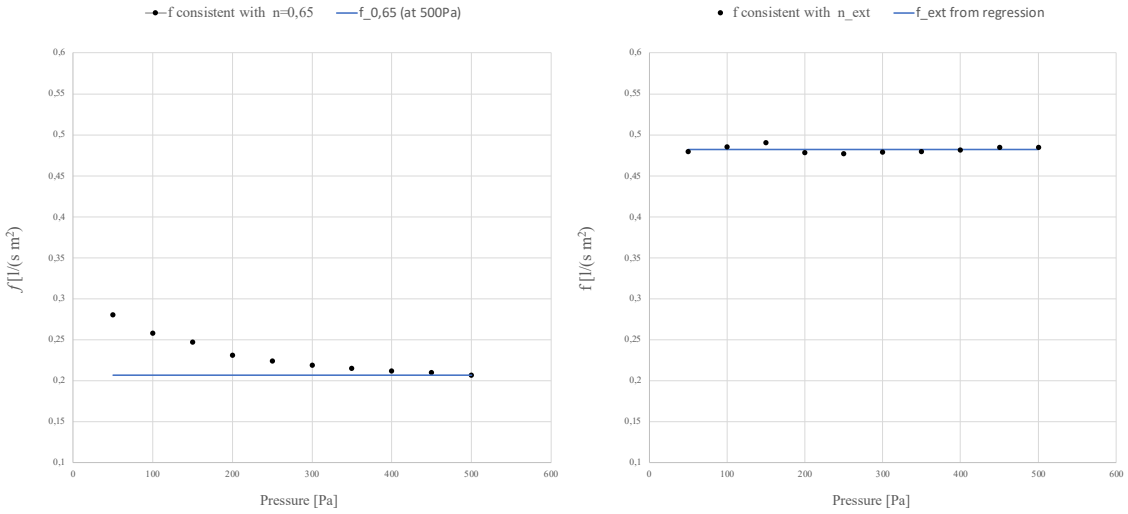

Figure 2: Impact of pressure on $f_{n=0.65}$ (2a) and $f_{n=n_ext}$ (2b)

The differences in Figure 2 are due to the models and can be seen analytically:

$$f_{ext} = \frac{m_{exf}}{A \times \Delta P^{(n_{ext}-0.65+0.65)}} = \frac{m_{exf}}{A \times \Delta P^{0.65} \times \Delta P^{(n_{ext}-0.65)}} = f_{0.65} \times \Delta P^{(0.65-n_{ext})} \quad (4)$$

$$f_{0.65} = \Delta P^{(n_{ext}-0.65)} \times f_{ext} \qquad or \qquad f_{0.65}/f_{ext} = \Delta P^{(n_{ext}-0.65)} \quad (5)$$

The adoption of the 0.65-model results in the determination of a coefficient that is not the proportionality coefficient of the power law alone, but a value influenced by a pressure contribution whenever $n_{ext} \neq 0.65$.

After having explored the issue of 0.65-model reliability, it can be concluded that:

1. The only advantage of the model based on n=0.65 is providing a value of $f_{0.65}$ with a single measurement.
2. Leakage classifications based on $f_{0.65}$ values only make sense if the system and all its components to be compared are tested at the same pressure, but European Standards provide inconsistent and non-unique pressure-test values for different system geometries and/or components.
3. The main reason for the inconsistency of the 0.65 model is that measurements made on a system at different pressures lead to different values of $f_{0.65}$, resulting in different Air Tightness Classifications.

After analyzing the differences between the two models, the authors propose the use of the extended model.

PROPOSED ALTERNATIVE METHODOLOGIES

The adoption of a more reliable leakage model opens the way to some interesting features, one of which is finding a way to make the visualization of leakage more practical and reliable. The proposed methods are based on appropriate management of information that is normally available and stipulating that leakage can be represented by quantifying the maximum system leakage expressed in terms of relative flow rate. To obtain a more reliable model it is necessary to combine information already known by the duct designer (such as design flow rate and design fan pressure for the air distribution system) with the extended model's parameters f_{ext} and n_{ext}.

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Method 1 proposal

The first method consists of using the extended model to calculate the value of the expected percentage leakage rate $R\%$ at the design pressure.

$$m_{exf} = f_{ext} \times A \times \Delta P_{ref}^{n_{ext}} \quad (6)$$

$$R\%_{\Delta P_{fan}} = \frac{f_{ext} \times A \times \Delta P_{ref}^{n_{ext}}}{m_{design}} \times 100 \quad (7)$$

Equation (7) provides relative air leakage assuming that the same reference pressure is applied to the entire duct area, which means that a representative operating reference pressure would need to be determined. The reference pressure may be chosen from the following options:

$\Delta P_{ref} = \Delta P_{fan}$, pressure at the fan (meaning the Static Pressure at the connection between AHU and ducts), in which case the amount of leakage indicated will be an overestimate.

$\Delta P_{ref} = \Delta P_{average}$, this should provide for a more accurate leakage estimate, as already indicated in EN 16798-5. In this case the error is significantly smaller, although this pressure will overestimate leakage for systems with a uniform distribution of leakage sites, due to linear averaging of pressures versus averaging of the pressure raised to the power n_{ext}

$\Delta P_{ref} = \Delta P_{EO}$, the equivalent operating pressure is, by definition, the pressure at which a DALT test provides the same leakage flow rate as the regularly operating system and therefore provides the most accurate estimate. It is possible to calculate the equivalent operating pressure starting from the design papers, i.e. according to the so-called 4-bands Method proposed by Pedranzini et al. in 2024[15].

Method 2 proposal

The second method proposed involves using an a priori definition of a maximum percentage leakage to be considered acceptable, and can be described as follows:

1) Define a maximum percentage air leakage with respect to the nominal system flow (e.g. 3% or 6%)

2) Use the extended model to calculate the maximum pressure consistent with this level of percentage leakage (e.g. if the accepted leakage is 6% or 3% respectively, the parameter could be called $\Delta P_{6\%}$ or $\Delta P_{3\%}$.

In other words, the $\Delta P_{6\%}$ (or $\Delta P_{3\%}$) value is the maximum static pressure value at which the system can operate to avoid air leakage greater than 6% (or 3%) of the design flow rate. This pressure level can be compared with the design operating pressure in order to check if the maximum allowed leakage can be consistent with the operating pressure of the system. Knowing the duct surface area and the operating pressure of the system, this approach facilitates calculation of the leakage factor f_{ext} consistent with the accepted percentage leakage.

NUMERICAL CASE STUDY

The examples below are for the supply ductwork of a primary air system characterized by a leakage factor that places it in sealing class A and a leakage exponent n_{ext} of 0.53. The system and geometric data refer to an existing system that was tested and analyzed for research purposes.

$m_{design} = 2390\ m^3\ h^{-1}$	$(1407 cfm)$	nominal system flow rate
$A = 83.1\ m^2$	$(895\ ft^2)$	leaking surface

expected class A (ATC5) with $f_{0.65} = 0{,}027 \times 10^{-3}\ m^3 s^{-1} m^{-2}$ at $500\ Pa$ $(2\ inH_2O)$.

By design, the total pressure at the fan exit is $P_{Tot\,fan} = 124 Pa$ $(0.5\ inH_2O)$, with air velocity $v = 6\ m\ s^{-1} (19.7\ ft\ s^{-1})$, so that the velocity pressure $P_{V\,fan} = \rho \frac{v^2}{2} = 21.6\ Pa$ $(0.086\ inH_2O)$, and the Static Pressure $P_{S\,fan} =$

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102.4 Pa (0.4 inH_2O). The design pressure at the diffusers is $P_{S\ terminals} = 14Pa$ (0.056 inH_2O).

Method 1

Considering the leakage specification of Class A (ATC5) at $\Delta P = 500\ Pa$ defined by the Standards, in this case EN 12237 - circular channels, as well as given the value of $n_{ext} = 0.53$, it is possible to calculate f_{ext} using Equation (5):

$$f_{ext} = f_{0.65} \times \Delta P^{(0.65-n_{ext})} = 0.027\ \times 10^{-3} \times 500^{(0.65-0.53)} = 0.057\ \times 10^{-3}\ m^3 s^{-1}\, m^{-2}$$

Method 1 involves estimating the leakage at the reference pressure and has three cases:

1. $\Delta P_{ref} = \Delta P_{fan}$ where $\Delta P_{S\,fan} = 102.4\ Pa$ (0.4 inH_2O) and, via the (6):

   $$m_{exf} = f_{ext} \times A \times \Delta P_{ref}^{n_{ext}} = 0.057\ \times 10^{-3} \times 83.1\ \times 102.4^{0.53} = 198\ \ m^3 h^{-1}\,(117\ cfm)$$

   And in percentage terms:

   $$R\%_{(\Delta P_{fan})} = \frac{198\ \ m^3 h^{-1}}{2390\ \ m^3 h^{-1}} = 8.3\%$$

2. $\Delta P_{ref} = \Delta P_{Average}$ with $\Delta P_{S\,Average} = 58.6\ Pa$ (0.23 inH_2O)

   By referring the calculation to the average pressure, a leakage value of $R\%_{(\Delta P_{average})} = 6.1\%$ is obtained.

3. $\Delta P_{ref} = \Delta P_{EO}$ In the case of the tested system, the value of the Equivalent Operating Pressure was obtained by means of the 4-Band Method and is equal to $\Delta P_{EO} = 62\ Pa$ (0.25 inH_2O), thus resulting in $R\%_{(\Delta P_{EO})} = 6.3\%$.

The results for the three cases are summarized in Table 1.

Table 1, Air Leakage Expressed as a Percentage at Different Reference Pressures

Reference Static Pressure [Pa]		M_{leak} [m^3h^{-1}]	R%
$\Delta P_{S\ fan}$	102 Pa	198.0	8.3%
$\Delta P_{S\ average}$	58 Pa	146.7	6.1%
$\Delta P_{S\ OE}$	62 Pa	151.7	6.3%

It is interesting to note that the values found in this case are quite similar, nevertheless the choice of reference pressure could be crucial. Setting the reference to the equivalent operating pressure provides the most accurate estimate.

Method 2

Method 2 proposes to identify a maximum *R*% air leakage value deemed acceptable and calculates the pressure value at which a possible DALT test can be performed to find that leakage. The evaluation of the tightness of the system can be based upon *i)* the pressure at the fan, *ii)* the average pressure or, better, *iii)* the equivalent operating pressure.

The example is developed using three different percentage leakage levels (9%, 6%, 3%) applied to three different tightness classes (Class A, Class B, Class C according to EN 12237 or ATC5, ATC4, ATC3 according to EN16798-3). The application of the extended model requires knowledge of the values of n_{ext} and f_{ext}. For the A Class case, the exponent is assumed to be $n_{ext} = 0.53$ and the leakage factor is calculated in the same way as for Method 1, obtaining $f_{ext} = 0{,}057\ \times 10^{-3}\ m^3 s^{-1}\, m^{-2}$. For classes B and C, the same procedure was followed, using the DALT test pressure values given in EN 12237, which are 1000Pa and 2000Pa respectively.

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The extended calculation for the A class system at a leakage R% = 9% (test pressure according to EN12237 for supply systems $\Delta P_{DALT} = 500Pa$):

$$m_{exf\,R\%} = \frac{R\%}{100} \times m_{design} = \frac{9}{100} \times 2390 = 215\ m^3h^{-1} = 0.06\ m^3s^{-1}\ (127\ cfm)$$

$$\Delta P_{R\%} = \left[\frac{m_{exf\,R\%}}{A \times f_{ext}}\right]^{\frac{1}{n_{ext}}} = \left[\frac{R\%}{100}\right]^{\frac{1}{n_{ext}}} \left[\frac{m_{design}}{A \times f_{ext}}\right]^{\frac{1}{n_{ext}}} = \left[\frac{9}{100}\right]^{\frac{1}{0.53}} \left[\frac{2390 \div 3600}{83{,}1 \times 0{,}057 \times 10^{-3}}\right]^{\frac{1}{0.53}} = 120Pa\ (0.48\ inH_2O)$$

Table 2 shows the $\Delta P_{R\%}$ values for all cases considered:

Table 2: Calculation of the Limit Pressure Levels for Compliance with the Maximum Percentage Leakage Referring to Classes A, B, C Defined According to EN 12237

Class		ΔP_{DALT} Test	$f_{0.65}$	f_{ext} (n_{ext}=0.53)	Maximum Allowed Pressure		
EN 12237	EN 16798-3	for Supply	[$10^{-3} \times m^3 s^{-1} m^{-2}$]	[$10^{-3} \times m^3 s^{-1} m^{-2}$]	$\Delta P_{3\%}$	$\Delta P_{6\%}$	$\Delta P_{9\%}$
Class A	ATC5	500 Pa	0,027	0,057	15 Pa	56 Pa	120 Pa
Class B	ATC4	1000 Pa	0.009	0.021	102 Pa	379 Pa	814 Pa
Class C	ATC3	2000 Pa	0.003	0.007	695 Pa	2572 Pa	5526 Pa

DISCUSSION

There are several important issues associated with the way that duct leakage is currently addressed and reported. The key issue is the fact that tightness classes do not address calculation of the implications of duct leakage during normal operation. Related to this, it has to be considered that pressures often change with time and between different parts of the duct system, and therefore the calculation of leakage flows at different pressures and different flow rates is required. The use of the extended model allows for greater accuracy over the entire operating pressure range.

Another issue is that rather than only considering absolute leakage flows, it is useful to refer to the maximum percentage duct leakage relative to the design flows under different operating conditions. This paper has shown how the extended model works well in conjunction with percentage duct leakage. In the case when only one branch is tested, the leakage percentage associated with the rest of the system can be better estimated utilizing the extended model. For this model, multiple pressure values must be tested by the operator and linear logarithmic regression is needed to obtain the parameters. Although most Standards currently do not provide procedures for a test like this, they describe how to do the DALT test at a certain pressure. All that is needed is a protocol for picking the other test pressures, and the common procedure for obtaining n_{ext} and f_{ext} from a logarithmic fit to the measurements.

Finally, it is worth noting that a realistic estimate of the value of n_{ext} would be very useful at the design phase. As a last resort, the value 0.65 can be used at that point, although the authors believe that the default value should be closer to 0.6. A better scouring of existing data (both published and unpublished), as well as experimental characterization of a significant number of systems through the extended model in the near future could provide a better default value for the leakage exponent.

CONCLUSIONS

Based upon the results presented herein, we can draw several conclusions. The first conclusion is that using the simplified model with an assumed exponent to characterize duct leakage can provide confusing, misleading or simply incorrect results for the characterization of leakage under normal operating conditions, or even for the characterization of ductwork when the leakage test is not performed at the exact reference pressure for whatever reason (e.g. equipment that is not capable of reaching that pressure for the section being tested). The extended (or multi-point) model that utilizes the actual flow exponent does not suffer from these drawbacks.

Finally, we also conclude that the two alternative methods proposed in this paper (Method 1 and Method 2) could be substituted for the current methodology, and by utilizing the multi-point test results, can provide immediate, understandable feedback on the assessment of the relative leakage that can be expected when the system operates at a specified operating

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pressure.

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NOMENCLATURE

m = volumetric air flow rate [$m^3\ h^{-1}$, $L\ s^{-1}$, cfm]
ΔP = Static Pressure difference [Pa, in.w.g]
f = Leakage coefficient [$L\ s^{-1}\ m^{-2}\ Pa^{-n}$]
A = Leaking duct surface area [m^2, ft^2]
n = Leakage exponent

Subscripts

$_{leak}$ = leakage
$_{0.65}$ = refers to the simplified model with n=0.65
$_{ext}$ = refers to the extended model

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