THE ZERO PRESSURE PARADOX

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ABSTRACT

The zero pressure compensation method has proven to be the best method to measure air flow rates accurately although it also has be shown that the accuracy depends on the type of air terminal device and how and where the pressure to be compensated is measured in the instrument. Although the principal of the zero pressure method implies universal applicability, in practice this does not seem to be case. This has lead us to develop the 'extended' zero pressure method

KEYWORDS

Zero-pressure method, Powered Flow Hood, compensation

1 INTRODUCTION

To balance ventilation systems accurate measurements of supply and return flows at a large spectrum of air terminal devices (ATD's) are needed. Measuring supply flows is challenging because of the uneven velocities coming from different parts of the registers. Passive flow hoods and anemometers can be extremely inaccurate because they typically mistake turbulence for additional flow and tend to read high. Return flows are much easier to read but if return pressures are low, the presence of a passive flow hood can increase the pressure in the return and force flow to divert to other registers where it is unmeasured causing low readings.

A Powered Flow Hood (PFH) is supposed to solve both of these problems by reducing the pressure next to the register to ´zero´ by a powered fan whose speed is adjusted automatically. This measuring method is therefore often referred to as the zero pressure compensation method. It has proven to be the best method to measure flow rates accurately ((Caillou, 2014), (Stratton, Walker, & Wray, 2012), (Wray, Walker, & Sherman, 2002) and (Knights & Gilbert, 2015)).

In theory 'zero pressure compensation' refers to an amount of compensation as if the PFH is no longer 'visible' for the flow. But of course it **is** present. In practice, the pressure to be compensated is difficult to measure and on top of that the needed pressure compensation for supply and return flow is fundamentally different. We refer to this tension between theory and practice as the 'zero pressure paradox'.

In order to understand better what is needed for accurate measurements independent of ATD, flow direction and flow intensity, fundamental research using Computational Fluid Dynamics (CFD) and measurements on a variety of ATD's have been carried out. This resulted in an enhanced understanding of the complexity of different flows encountered by a PFH and what ´zero pressure´ in fact should be.

2 ZERO PRESSURE METHOD

The zero-pressure method was originally developed in the early seventies by Bitter & Detzer (Bitter & Detzer, 1974) as a means to accurately measure air flow quantities at inlets and outlets.

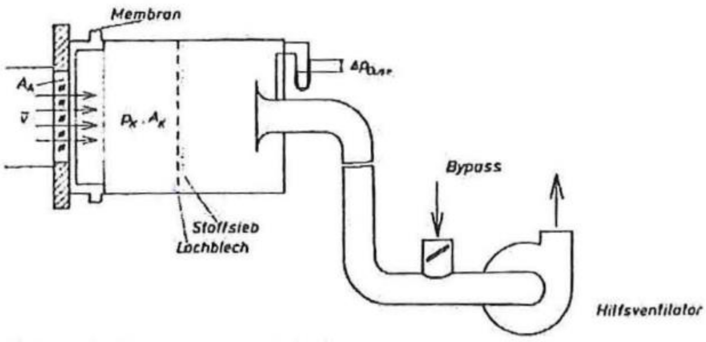

Figure 1: Original schematic for the zero-pressure method (Bitter & Detzer, 1974).

Figure 1 shows a schematic of the setup to apply the zero-pressure method at an outlet. The outlet is fully covered by a big box which is connected to a duct and a fan. When introducing the device, the pressure in the box will rise due to its induced resistance and consequently the flow will decrease. Powering the fan decreases the upstream pressure until the pressure measured in the box is compensated for. The supply flow from the ATD is now supposed to be as if no flowhood is present and can be measured accurately. The same principle applies for return flow.
Next the influence on a ventilation system of a non-powered flow hood will be discussed

2.1 System dependent impact of an intrusive device

Figure 2 shows a schematic of a hypothetical ventilation system, where air is sucked into the system by a fan on the left hand side. The air will flow through the main branch and exits over branch 1 and branch 2.
The fan characteristic can be given in a fan performance curve, showing the pressure difference dp_T - the flow quantity Q_T relation assuming a constant fan speed.

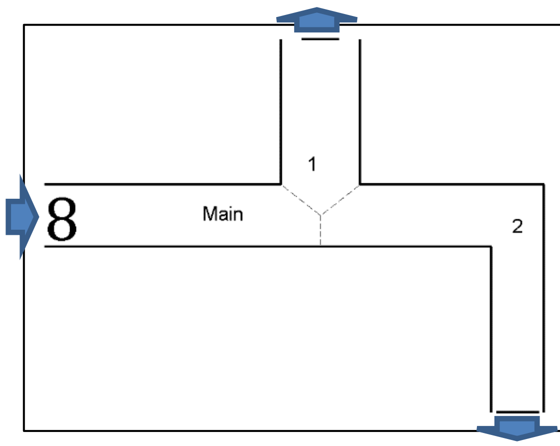

Figure 2: System Layout

The resistance curve of the independent branches can be estimated by equation (1)where Q is the air flow quantity in the branch and R the resistance factor. This resistance factor relates to the flow parameters and duct geometry (Rolloos, 1975), but will be assumed only to depend on the geometry in this example.

$$dp = R \cdot Q^2 \quad (1)$$

Figure 4 and Table 1, Situation 1, show the equilibrium state of the undisturbed system providing 250 m^3/hr at a total pressure difference of 78.2 Pa.

Table 1: Equilibrium of the undisturbed ventilation system (1).
Equilibrium with an intrusive device at branche 1 (2).

Parameter	**Flow [m^3/hr]**		**Pressure [Pa]**	
Situation	1	2	1	2
Main Branch	250	228	31.3	25.9
Branch 1	153	120	46.9	57.8
Branch 2	97	108	46.9	57.8
Total System	250	228	78.2	83.7

With a device at the outlet of branch 1, an additional resistance is induced reducing both the flow in branch 1 and the total flow (Table 1, Situation 2).

The flow reduction depends on the resistance of the intrusive device relative to the resistance of the measured branch and the total system. In particular in high performance buildings, where high efficiency, low noise and therefore low resistance are key, the flow reduction due to the use of flowhood can be considerable. Due to the large variety of ATD's and ventilation systems it is not possible to account for the resistance of an intrusive instrument by a single calibration factor (Roper, 2013). The zero pressure method seems to be the best alternative.

Some design challenges for a PFH to be dealt with will be discussed in the following section.

2.2 Design challenges for an easy-to-use PFH

The first measuring systems applying zero pressure compensation were quite big.

Figure 3: two zero pressure measuring systems, one from 1978 and one as it is being built today (top left)

These days a PFH should be easy to use, lightweight and compact. These are conflicting demands for proper zero pressure compensation.
Air supply flow will expand fast after exiting the ventilation system, causing a quick equalization of the surrounding static pressure to environmental pressure. When limiting the space at the exit with a flowhood, the flow is unable to expand freely and a non-homogeneous static pressure field is formed. This pressure field strongly depends on the flow pattern and

magnitude. Even in a PFH with a big hood (Presser, 1978) the pressure to be compensated to zero pressure depended on the measuring location in the hood.

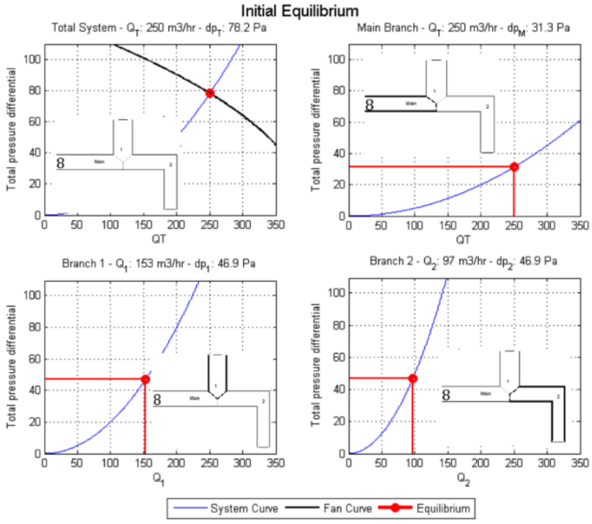

Figure 4: System equilibrium, undisturbed. From top left clockwise: total system curve and fan curve; main branch system curve; Branch 2 system curve which is steep indicating a high resistance duct and/or valve and branch 1 system curve which is less steep, therefore indicating a lower resistance and receiving a larger part of the total flow. The pressure differences over branch 1 and branch 2 are equal being the difference between the pressure in the main branch at the split and the environmental pressure.

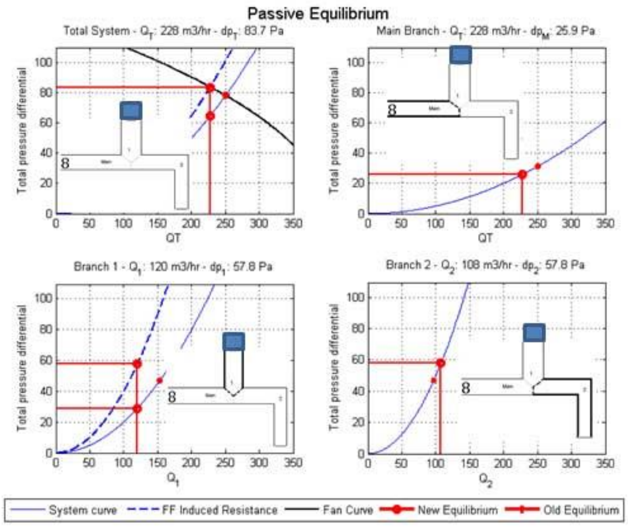

Figure 5: System equilibrium with intrusive device. The additional resistance induced by the intrusive device is visible by the steeper total system curve, resulting in a new equilibrium with the fan curve with a decreased flow and an increased pressure difference. The main branch system curve does not change, therefore the equilibrium shifts over this line to a lower air flow. A similar process applies to branch 2 where the equilibrium shifts over the system curve to a higher value due to an increased pressure difference at the splitting point. Finally, the additional resistance as induced by the device resulting in a steeper resultant system curve of branch 1 plus the device, thereby lowering the flow and increasing the pressure drop

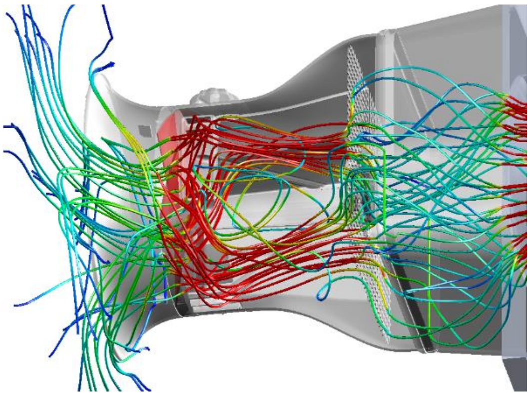

Figure 6: The non-homogeneous pressure field in a small and easy-to-handle PFH. Which pressure should be compensated for?

3 RETURN- AND SUPPLY FLOW COMPARED

Until now, supply and return flow have been treated the same way and no clear distinction is made. To show that these two flows are fundamentally different, the 1-dimensional pressure gradient will be investigated using a simple model.

3.1 Pressure Diagrams

The set-up for the model consists of a single duct with a fan in the center and a valve at both ends of the ducts. The fan induces a flow from left to right and measurements take place on the right hand side of the duct for supply flow (Figure 8) and on the left hand side for return flow (Figure 10).

Figure 9 and Figure 11 show the influence of a non-powered flow hood on the static pressure in the system.

When looking at supply flow (Figure 9), the theoretical pressure measured in front of the instrument is equal to the pressure to be compensated to restore the original condition. With a small PFH this pressure is hard to determine due to the non-homogeneous pressure field in the PFH. An additional practical problem in a small PFH is the large fluctuations of the pressure at higher flows.

In case of return flow, the non-homogeneity of the pressure field is less important since the flow is disturbed by the ATD downstream of the PFH. However, the pressure measured differs greatly from the pressure to be compensated as shown in Figure 11. This difference is equal to the dynamic pressure. The dynamic pressure can be estimated using the conditions measured with a 'passive' PFH.

When measuring return flow, theoretically, 'zero-pressure' means an equalization of the sum of static and dynamic pressure (the total pressure) with respect to the atmospheric pressure. This is fundamentally different from supply flow.

3.2 Supply flow challenges

Figure 7 shows a non-homogeneous pressure field resulting from a typical ATD. In order to compensate the resistance of the PFH, the question to be asked is: at which point should zero-pressure be applied? Since an instrument should be accurate independent of the ATD or flow measured, determining the perfect location for the pressure measurement proved to be impossible since it will differ for every ATD and air flow quantity. Therefore an 'extended' zero pressure method is developed that uses the pressure in the 'passive' PFH in relation to the passive flow amount for a wide variety of ATD’s at a large range of air flow quantities.

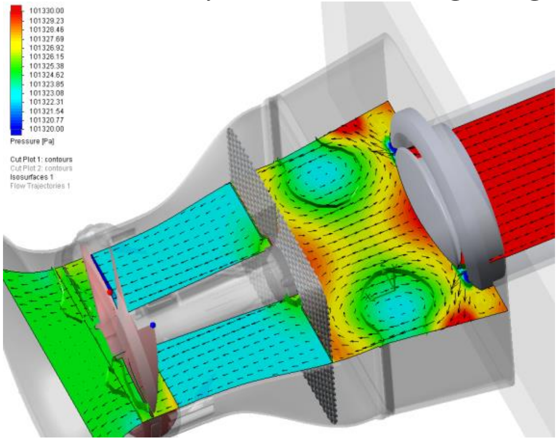

Figure 7: Valve dependent flow and pressure patterns at the pressure measurement area.

4 CONCLUSIONS

To summarize the main findings, the following conclusions can be drawn.

- The effect of a flow hood on low resistance ventilation systems is large and must be compensated for.
- The use of a compact PFH induces several challenges:
  - Supply and return flow are fundamentally different due to non-zero dynamic pressure.
  - Return flow measurement is independent of the ATD and system measured, though an invariable error is made being equal to the dynamic pressure of the flow.
  - Supply flow measurement strongly depends on the ATD measured introducing pressure variations around the point of measurement.
- The ‘Zero-pressure paradox’: in theory it seems straightforward to compensate the pressure caused by an intrusive device, in practice it is not.
- With the ‘extended’ zero pressure method the pressure to be compensated can be more accurately determined both for supply and return flows.

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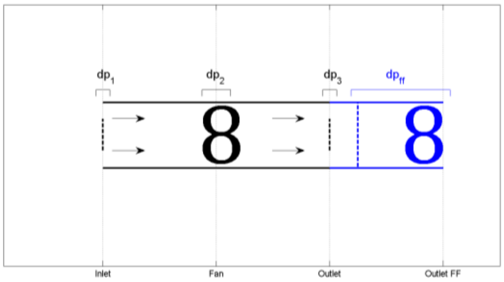

Figure 8: Layout supply flow

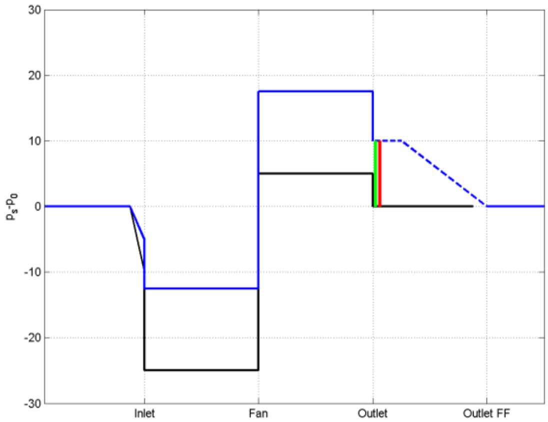

Figure 9: Supply flow pressure diagram. The black line represents the static pressure in the system without a PFH (FF); the blue solid line shows the static pressure including a PFH and the blue striped line the static pressure inside the FF. The red bar represents the pressure measured and the green bar the pressure to be compensated

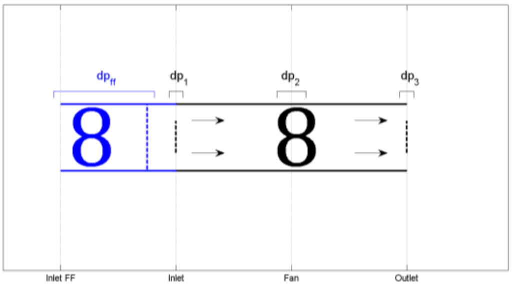

Figure 10: Layout return flow

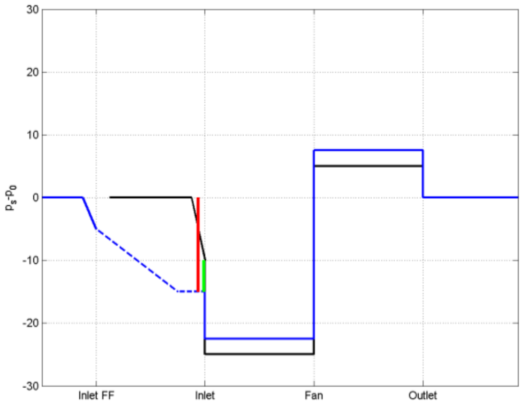

Figure 11**:** Return flow pressure diagram. The black line represents the static pressure in the system without a PFH (FF); the blue solid line shows the static pressure including a PFH and the blue striped line the static pressure inside the FF. The red bar represents the pressure measured and the green bar the pressure to be compensated.