

Measurement of Perforated Tile Airflow in Data Centers

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ABSTRACT

The cooling performance of a raised-floor data center is dominated by its ability to deliver adequate cooling airflow to IT equipment through perforated tiles. Although many studies have focused on the design of plenum systems and the use of computational fluid dynamics (CFD) to predict tile airflow rates, very little has been published regarding accurately measuring airflow in existing facilities. We investigate the ability of a commercially available passive flow hood to accurately measure data center tile airflow and find that the built-in back pressure compensation functionality does not work well. Further, two previously proposed correction methods for data center applications provide reasonable accuracy when applied to fairly restrictive tiles, but are inaccurate for large-open-area tiles. As a solution, we propose a new family of semiempirical two-measurement correction methods for passive flow hoods. Additionally, we construct a high-accuracy powered flow hood (PFH) to validate all of the passive-hood methods. The existing and proposed methods are evaluated against both CFD simulations and measurements made in an actual data center. The correction method proposed in and recommended by this study yields the highest accuracy—less than 5% error for scenarios considered.

INTRODUCTION

The great majority of existing data centers, and many new ones, use a raised floor plenum to distribute cooling airflow to IT equipment. Air is pumped into the floor plenum by computer room air handler (CRAH) units and exits the plenum through perforated floor tiles that are generally located in front of and aligned with each rack. The challenge in efficiently cooling IT equipment is to provide sufficient,

but not excessive, airflow to each rack. Tile airflow is typically measured to establish a baseline performance of the data center as part of a routine cooling assessment, in preparation for infrastructure changes, or in troubleshooting hot spots.

Several research studies have used passive flow hoods to measure tile airflow. Many such measurements have even formed the basis for validating computational fluid dynamics (CFD) models. Schmidt et al. presented the first true tile-by-tile airflow prediction model, which used a 2-D (depth-averaged) CFD model. CFD results were compared against measurements made with a commercially available flow hood for all 60 tiles in a small test section of a data center. Schmidt et al. (2001) noted that the flow resistance of the device must be low relative to the tile resistance to obtain an accurate measurement. They also proposed a correction to the airflow indicated by the flow hood based on the ratio of resistances of the measurement device and tile being measured. This correction is referred to here as the one measurement uniform pressure (1MUP) method and is explained in more detail in the 1MUP Method section of this paper. Radmehr et al. (2005) used a commercial flow hood to measure tile airflows in developing a model for leakage flow through raised-floor systems. They measured tile airflows using their hood's built-in back pressure compensation (BPC). While the present study finds that the built-in BPC method may not significantly improve measurement accuracy, little correction was likely needed in Radmehr et al. (2005) given the very high resistance 16% nominal open-area tiles involved.

More recent research related to tile airflow measurement has focused on the creation of calibrated CFD models for data

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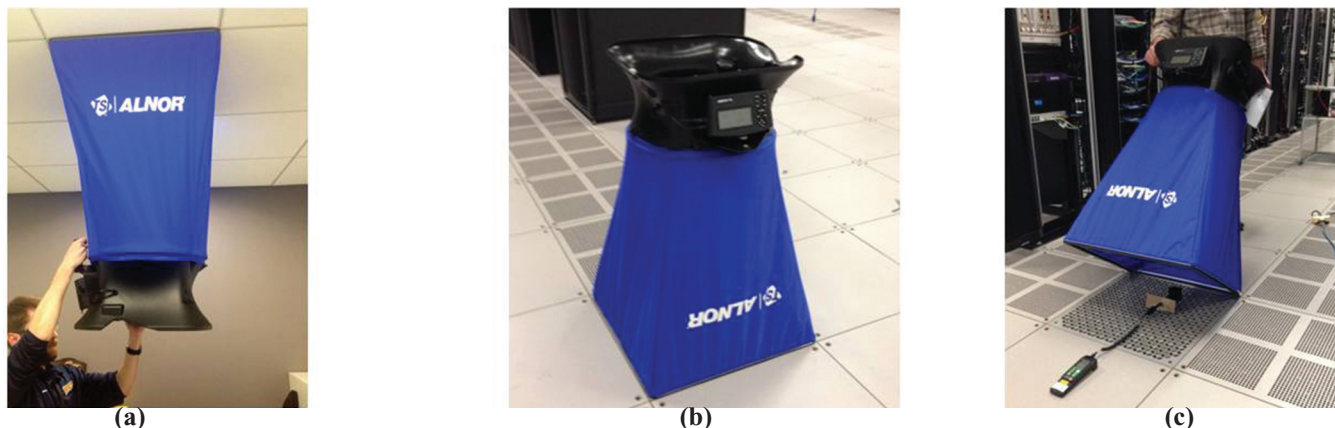


Figure 1 Airflow measurements with a passive flow hood: (a) HVAC application, (b) Data center application, (c) Data center application (with anemometer).

centers. Seymour (2014); Renner and Seymour (2015); and Pardey et al. (2015) have all focused on this subject with the goal of creating trustworthy models with which the effect of cooling-equipment changes, an evolving IT population, and other changes can be predicted. Even if perforated tile airflow rates are computed in CFD, rather than specified as boundary conditions, it is necessary to measure the airflow from many or all tiles to validate the model.

Perforated tile airflow is most commonly measured with a passive (fanless) flow hood, as shown in Figures 1 and 2, and as described with detail in the “Passive Flow Hood Characterization” section of this paper. Such hoods were originally developed and are primarily used today for general HVAC applications as shown in Figure 1a. In the typical HVAC application, diffusers or supply grilles are located downstream of one or more central air handlers, through a tree-shaped network of ductwork. The data center application (Figure 1b) is quite different in that all perforated tiles are fed by the same large plenum—ideally, at one uniform pressure. Consequently, when a flow hood of large resistance (relative to the perforated tile’s resistance) is placed over a perforated tile, airflow may simply bypass the tile with the extra flow-hood resistance and exit the plenum through other ways such as perforated tiles or cable cutouts. When high-flow (e.g., 56% and 68% open area) tiles are employed, the uncorrected passive flow hood may indicate only half or less of the actual, unmeasured flow rate. While many commercial hoods do include some type of BPC as mentioned above, we speculate (based on the poor performance for data centers observed in this study) that this functionality was developed with general HVAC applications and not data centers in mind.

Seymour (2014) suggested and Pardey et al. (2015) employed the correction method shown in Figure 1c. Following this method, referred to here as the anemometer method, a velocity is recorded with and without the hood in place. The airflow measured by the hood is then scaled up by the ratio of the no-hood-to-hood velocities. However, to date, no informa-

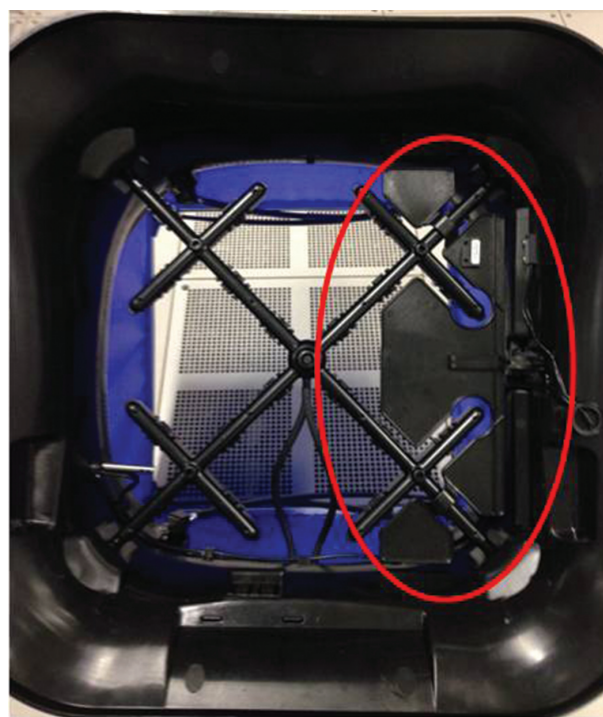


Figure 2 Pitot tube array and BPC flap.

tion has been published with regard to the accuracy of this (or any other) correction method.

This study investigates the accuracy of existing methods (1MUP, anemometer, and BPC) and proposes a new family of semiempirical two-measurement (2M) methods that can be implemented on any commercial hood that features built-in BPC. Additionally, a powered flow hood (PFH) was constructed specifically for this study and used to validate the (more practical) passive hood methods.

Our findings are based on CFD simulations, laboratory tests, and actual data center tile airflow measurements. With no correction, the passive flow hood significantly underesti-

mates tile airflow and built-in BPC only marginally improves accuracy. For example, based on measurements with 25% open perforated tiles, BPC reduces the error from about 17% to 11%. The 2M methods proposed here provide the highest accuracy and are practical to implement, as demonstrated by CFD simulations and actual data center measurements.

PASSIVE FLOW HOOD CHARACTERIZATION

Before discussing specific measurement correction methods, we consider some aspects that apply generally to all of the passive flow hood methods. For this study, an Alnor Model EBT 731 flow hood was employed, as shown in Figure 1. The hood measures airflow rate indirectly through pressure measurements taken from the pitot tube array shown in Figure 2. The instrument is supplied with a built-in BPC mode that requires two successive measurements, the second of which is with the back-pressure flap (Figure 2) in the closed position. As delivered, the hood does not provide intermediate airflow-rate readings when in BPC mode; only the final back-pressure-corrected value is displayed. As our proposed methods require the two measured flow rates (flap open and flap closed) as input, we bypassed a flap-position switch on the hood so that it could be used in normal (non-BPC) mode either with or without the flap in position. Although the hood does not directly provide the true airflow rate with the flap down, we manually calibrated the hood.

We calibrated the flow hood against a laboratory flow bench for both flap-open and flap-closed configurations, using the setup shown in Figure 3. The flow bench includes a control fan with integrated airflow-rate measurement that may be set to a prescribed airflow rate independent of pressure drop. (The flow bench fan itself was first calibrated against another similar unit and found to be within $\pm 1\%$ over the range of interest.) A duct of approximately 16 in. (0.4 m) in diameter and 10 ft (3 m) in length connects the fan to a box, at the entrance of

which is located an aluminum air-mixing screen. The length of duct and the mixing screen together ensure a relatively uniform downstream airflow profile. The outlet of the box, just upstream of the flow hood, may be left open or fitted with a standard 2×2 ft (0.6×0.6 m) perforated tile. The flow hood was first calibrated without a tile and the indicated values were scaled (by modifying a calibration factor in the hood's firmware) by a factor of 0.94 to achieve an accuracy of $\pm 1.5\%$ over the range of 250–1240 cfm (425–2107 m³/h). The calibration study was then repeated with 25% and 56% open area tiles; indicated airflow rates were not measurably affected by the presence of the tiles (and attendant changes to the airflow pattern presented to the hood).

All of the 2M correction methods (that use the back-pressure flap) require the hood resistance in both open and closed-flap configurations as input. The flow resistance coefficient is defined by

$$\Delta P = \alpha Q^2 \quad (1)$$

where ΔP is the pressure drop across the resistance and Q is the flow rate. Note that Equation 1 can be alternatively stated in terms of a dimensionless loss coefficient f :

$$\Delta P = \frac{f}{2} \rho V^2 \quad (2)$$

where ρ is the density of air and V is the average velocity over the area A of the device that is normal to the flow direction. For floor tiles and the flow hood, the reference area A is always taken here as 4 ft² (0.37 m²). The loss coefficient and flow resistance coefficient are related by

$$\alpha = \frac{f\rho}{2A^2} \quad (3)$$

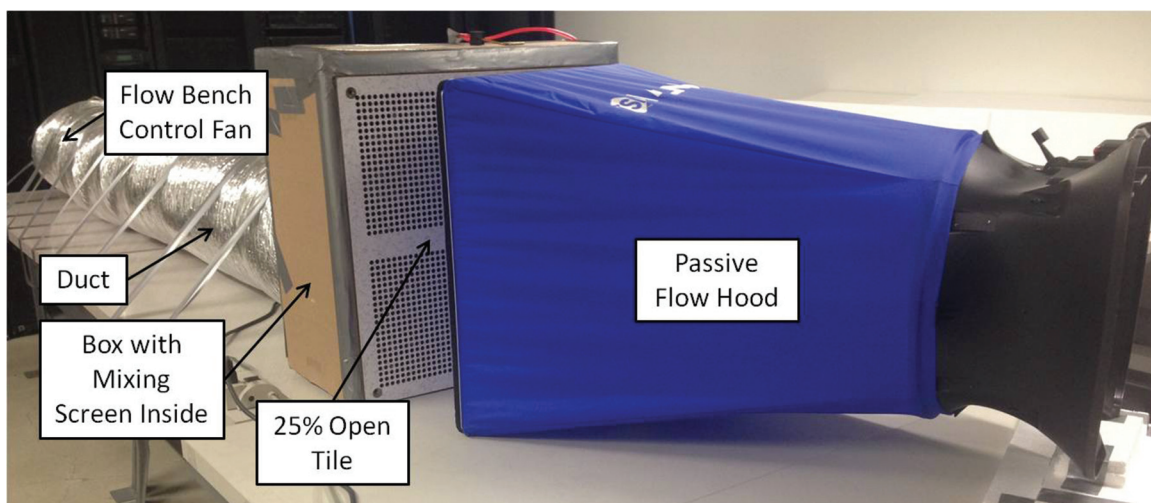


Figure 3 Calibration of flow hood with flow bench.

Figure 4 shows the flow resistance characterization of the flow hood with and without the back-pressure flap. The hood loss coefficients f were determined to be 14.9 and 32.1 for the two scenarios, respectively. For comparison purposes, note that the loss coefficients for 25% and 56% open tiles are $f_T = 47$ and $f_T = 5.9$, respectively, based on manufacturer-published data.

POWERED FLOW HOOD

Figure 5 shows a PFH that was constructed for the present study. The PFH features a 2×2 ft (0.6×0.6 m) box that is placed over a tile and connected to a metered fan by a 16 in. (0.4 m) diameter duct. The PFH fan compensates for the resistance of the measurement instrument and, just as the laboratory flow bench, can be set to operate at a specific airflow rate that is independent of pressure drop. For the current study, a manometer reading inside the box is used to automatically drive the fan until the pressure inside the box equals that of the surrounding room.

As is discussed further in the Data Center Airflow Measurements section, the PFH is used as a benchmark comparison for the passive flow hood methods discussed here. However, PFH measurements could only be obtained for the more restrictive and lower flow rate 25% open area tiles as the airflow of the 56% and 68% open area tiles in the subject data center exceeded the range of the PFH. Furthermore, with large-open-area tiles, the airflow dynamics between the plenum and room become much more intimately coupled and the zero-pressure box environment likely no longer preserves the true unmeasured tile environment.

METHODS FOR ESTIMATING PERFORATED TILE AIRFLOW IN DATA CENTERS

We now discuss in detail several methods for correcting passive flow hood measurements for data center applications.

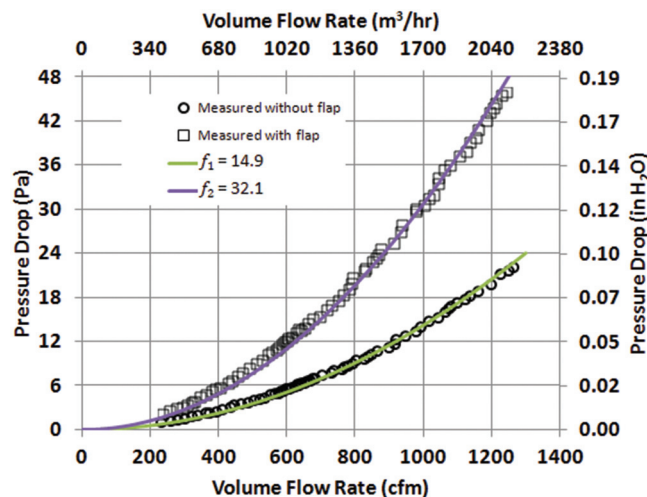


Figure 4 Flow resistance characterization of flow hood.

Two of the methods were previously described in the literature (1MUP and anemometer) and the remaining methods are proposed here for the first time. We also consider the built-in BPC functionality in our study; however, as the details of the method are proprietary to the hood manufacturer, we have no theoretical basis for this method. In sections that follow, we discuss CFD simulations and actual data center measurements that were performed in order to compare and evaluate the various tile airflow measurement methods.

1MUP Method

We denote the method proposed by Schmidt et al. (2001) as the 1MUP method because it involves a single flow hood measurement and may be derived assuming that the pressure drop originally across the tile alone also exists across the combined tile and flow hood. Following this assumption, we can write

$$f_T Q_T^2 = (f_h + f_T) Q_h^2 \quad (4)$$

where Q_T is the “true” tile airflow and Q_h is the tile airflow indicated by the hood. The corrected tile airflow Q_T may then be computed as

$$Q_T = Q_h \sqrt{1 + \frac{f_h}{f_T}} \quad (5)$$

The loss coefficient of the tile f_T can be easily determined from manufacturer-supplied data; however, the flow hood loss coefficient f_h typically must be measured as discussed previ-



Figure 5 Powered flow hood (PFH).

ously. Clearly, when the hood resistance is relatively small, the indicated airflow will equal the true tile airflow. This method performs better with more-restrictive tiles, when both the quantity $f_h / f_T = \alpha_h / \alpha_T$ is close to zero and the resulting deviation from a uniform and consistent (with the hood versus without the hood) driving pressure is small.

Anemometer Method

Seymour (2014) proposed the anemometer method shown in Figure 1c. This method employs a simple empirical scaling of anemometer (vertical velocity) readings taken with and without the hood in place as a basis for a correction of the indicated hood airflow. For example, if the anemometer reads half of its original velocity after the hood is placed over the tile (and anemometer), indicated hood airflow is scaled up by a factor of two.

The anemometer method requires two instruments and a great deal of bending to floor level for large data centers. The correction theoretically provides the “true” tile airflow as long as the velocity at the location of the anemometer scales uniformly with the tile airflow rate from “without hood” to “with hood” scenarios. We test this assumption by considering the accuracy of the method as determined by CFD simulations and data center measurements. Additionally, to investigate the repeatability of the anemometer method, we made repeated measurements on selected tiles in an actual data center as is discussed in the Data Center Airflow Measurements section.

2M Methods

We propose a family of correction methods that require two airflow measurements made in succession—each with a different hood resistance. Some commercial flow hoods already feature built-in 2M methods in which an alternative (higher) hood resistance is created by virtue of a back-pressure flap that may be rotated into a position so that it obstructs some of the airflow through the hood, as shown in Figure 2. We designate the 2M method built-into the flow hood employed in this study as the *BPC method*.

The physical interpretation of the 2M methods is demonstrated in Figure 6. Airflow measurements Q_1 and Q_2 are made using the lower (base) hood resistance, and the higher hood resistance f_2 (with the back-pressure flap in the open and closed positions, respectively). The corrected airflow measurement that we seek is the airflow that would be measured if the hood resistance approached zero; this corresponds to the intercept of the curves with the vertical axis of Figure 6. The reason for, and the differences among, the four extrapolation curves is explained in this section.

One approach to determining the extrapolation curves of Figure 6 is to empirically fit, for example, linear, exponential, or other functions to the measured data points (f_1, Q_1) and (f_2, Q_2) . An alternative is to derive a curve based on a physical model applicable to data center applications. Starting with the

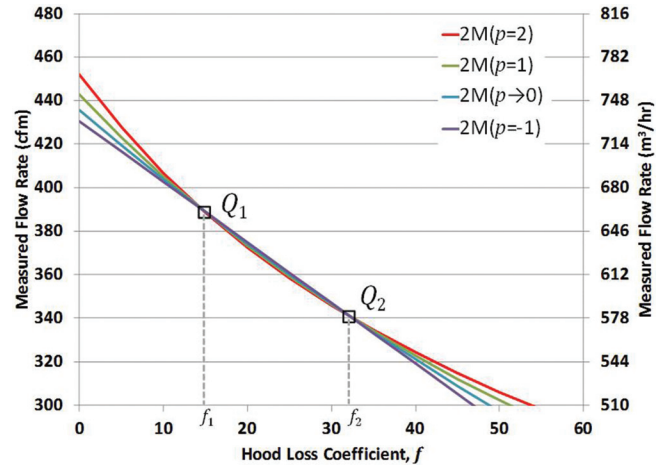


Figure 6 Measured tile airflow as a function of hood resistance.

physical approach, we assume that the overall effective pressure drop across both tile and hood combinations is the same as that for the tile alone (as was assumed with the 1MUP method):

$$f_T Q_T^2 = (f_1 + f_T) Q_1^2 = (f_2 + f_T) Q_2^2 \quad (6)$$

Eliminating f_T from Equation 6 yields the correction

$$Q_T = \text{Sign}(Q_1) |Q_1| |Q_2| \sqrt{\frac{1 - \beta}{Q_2^2 - \beta Q_1^2}} \quad (7)$$

where $\beta = f_1 / f_2$ is the ratio of hood resistances in the “no flap” and “flap” configurations. Note that the correction embodied in Equation 7 is designated as the 2M($p = 2$) method and an example of its implementation is shown in Figure 6 where Equation 7 represents the vertical-axis intercept of the 2M($p = 2$) curve. Although the 2M($p = 2$) correction provides the highest accuracy of the methods considered here, it may not provide a correction for all possible measured airflows Q_1 and Q_2 , as the quantity in the denominator can theoretically be negative. We note that this was not observed in the CFD simulations or data center measurements performed as part of this study; the 2M($p = 2$) method always produced a sensible numerical estimate. However, alternative extrapolation curves can be defined that are theoretically more robust. It can be shown that replacing the squared exponent of Q_T^2 , Q_1^2 , and Q_2^2 in Equation 6 with the variable p leads to the following general tile airflow correction (and vertical-axis intercept of Figure 6):

$$Q_T = \text{Sign}(Q_1) |Q_1| |Q_2| \left(\frac{1 - \beta}{|Q_2|^{p-\beta} |Q_1|^p} \right)^{\frac{1}{p}} \quad (8)$$

where p can be any real value. Further, it can be shown that in the most typical scenario, in which $|Q_T| > |Q_1| > |Q_2|$, Equation 8 always produces a numerical value while

$$p < \frac{\ln \beta}{\ln \frac{|Q_2|}{|Q_1|}} \quad (9)$$

Equation 8 provides the desired correction and the vertical intercept of the curves in Figure 6. The curves themselves are given by

$$Q(f) = \text{Sign}(Q_1)|Q_1||Q_2| \left(\frac{1 - \beta(f)}{|Q_2|^p - \beta(f)|Q_1|^p} \right)^{\frac{1}{p}} \quad (10)$$

in which β becomes a generalized function of f :

$$\beta(f) = \frac{f_1 - f}{f_2 - f} \quad (11)$$

The other curves shown in Figure 6 correspond to $p = -1$, $p \rightarrow 0$, and $p = 1$. It can be shown that these values of p represent extrapolation curves that are linear in f , exponential in f , and vary with $1/f$, respectively. The three methods, denoted $2M(p = -1)$, $2M(p \rightarrow 0)$, and $2M(p = 1)$ are well-behaved for any value of Q_1 or Q_2 and are viable (empirical) alternatives to the $2M(p = 2)$ method if the latter method cannot produce a numerical value for Q_T .

CFD SIMULATIONS

CFD simulations were performed to compare the tile airflow measurement techniques in a consistent manner free of experimental uncertainty. For reference, a Reynolds-averaged Navier-Stokes (RANS) CFD tool was employed with the standard $k-\varepsilon$ turbulence model. The 42×28 ft (12.8×8.5 m) data center layout considered is shown in Figure 7; the arrangement

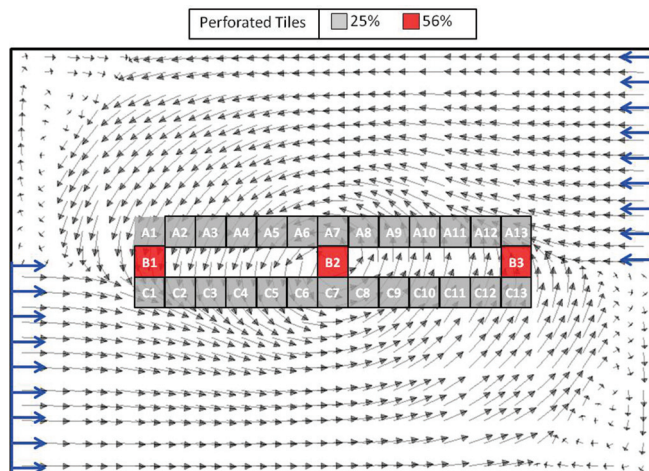


Figure 7 Simulated layout with 24 in. (0.6 m) plenum.

of tiles mirrors that of the portion of a real data center studied here and discussed in the following section. However, for convenience, only a small portion of the data center is modeled in CFD; results cannot be directly compared to the experimental measurements. For our purposes, the choice of CFD modeling details employed and the typical accuracy of such tools are likely not as important as the ability of the tool to correctly distinguish between the scenarios considered. Therefore, all CFD settings and, most importantly, the computational grid, were kept consistent for all scenarios representing “no flow hood” or “with a flow hood.”

The two outer rows of 13 tiles each are of 25% open area while the middle three tiles are of 56% open area. In order to intentionally create complex pressure and airflow variations in the plenum, the cooling airflow is introduced into the plenum at diagonally opposite corners, which creates the recirculation pattern shown in Figure 7. The return airflow is distributed uniformly across the ceiling, which is located 10 ft above the raised floor. Although not shown in Figure 7, IT racks facing each perforated tile along the outer rows were included in the simulations. The racks were specified with a 6 kW power dissipation and airflow based on 125 cfm/kW [$213 \text{ m}^3/(\text{h} \cdot \text{kW})$]. However, it was found that rack power and airflow rate did not significantly affect the simulated hood airflow measurements. Both 6 and 24 in. (0.15 and 0.60 m) deep plenums were considered with the former generating much larger pressure variations in the plenum.

To compare measured tile airflow rates to actual tile airflow rates, the flow hood was explicitly modeled, with some simplifications. The hood was modeled simply as a box, open on each end, with the total flow resistance specified across a plane at mid-hood height. The no-flap and flap hood resistance values used correspond to those of the actual hood resistances measured in this study and discussed previously. For each scenario considered, one CFD simulation, which does not include the flow hood at all, provided the actual tile airflow rates. Subsequently, the flow hood with a no flap resistance was placed on one tile at a time in successive CFD simulations. The flow hood simulations were then repeated with the flap hood resistance. A total of 59 CFD simulations were thus required for each scenario with our 29-tile layout.

Normally, some type of tile airflow straightening should be included in data center CFD simulations that feature simplified two-dimensional representations of tiles—as in the present study—because the three-dimensional structure of actual perforated tiles largely removes the horizontal velocity components from the airflow passing through them. However, because the simple straightening technique used here that forces all airflow emerging from the tile to be in the perpendicular direction substantially affected the results for the anemometer method scenario, both straightening and no straightening results are included. The velocity values used in the anemometer method were taken as the normal component of the velocity immediately above and in the (arbitrarily

Table 1. Relative RMSE of Methods from CFD Measurements

Methods	6 in. (0.3 m) plenum (straightening)		24 in. (1.2 m) plenum (straightening)		24 in. (1.2 m) plenum (no straightening)	
	25% tiles	56% tiles	25% tiles	56% tiles	25% tiles	56% tiles
Raw	10.8%	47.4%	7.1%	45.3%	6.3%	43.5%
1MUP	9.8	29.2%	8.1%	17.5%	9.9%	19.6%
Vane Anemometer	19.3%	11.1%	3.0%	3.7%	87.2%	21.3%
2M($p = -1$)	4.5%	30.9%	1.8%	31.1%	2.9%	29.1%
2M($p \rightarrow 0$)	4.6%	26.4%	1.5%	26.9%	2.9%	24.9%
2M($p = 1$)	4.6%	18.7%	1.1%	19.2%	2.9%	17.1%
2M($p = 2$)	2.7%	1.5%	0.7%	1.4%	3.0%	4.5%

selected) upper-right quadrant of each tile, as viewed in Figure 7.

Table 1 shows the results of the CFD simulations in terms of average relative root mean square error (RMSE) broken down by tile type where relative RMSE is defined by

$$\text{Relative RMSE} = \sqrt{\sum_{i=1}^n \left(\frac{Q_{i,est} - Q_{i,true}}{Q_{ave,true}} \right)^2} \quad (12)$$

where $Q_{i,est}$ is the tile airflow rate computed by the method, $Q_{i,true}$ is the actual tile airflow rate as provided by the CFD simulation, and $Q_{ave,true}$ is the average airflow rate of the tiles considered. The raw values refer to the airflow measured by the no flap flow hood without any correction applied. The other methods include the corrections as detailed previously. Note that the hood's built-in BPC method was not evaluated in CFD as details of this correction method are unknown.

Uncorrected flow hood measurements are poor, particularly for large-open-area tiles. Relative RMSE for 56% open-area tiles ranges from 43.5% to 47.4%. The best (lowest) relative RMSE values for each scenario and tile type are colored in darker gray with the second best values colored in lighter gray. The 2M($p = 2$) method is substantially better than the other methods for the scenarios considered with no relative RMSE greater than 4.5%. The second-best accuracy is achieved with the 2M($p = 1$) method. The anemometer method performs much better with straightening; however, it is not clear if this is an artifact of the CFD modeling of the anemometer location or a real effect. Both the 1MUP and 2M($p = 1$) methods are reasonably accurate for 25% open area tiles; these methods have relative RMSEs near 20% for all 56% open area tile scenarios, but still offer a substantial improvement over uncorrected measurements.

Figure 8 shows the relative RMSE for the 2M methods as a function of p for all 25% and 56% open area tiles for the 24 in. (0.6 m) plenum (straightening) scenario. For reference, the relative RMSE achieved with raw measurements and the

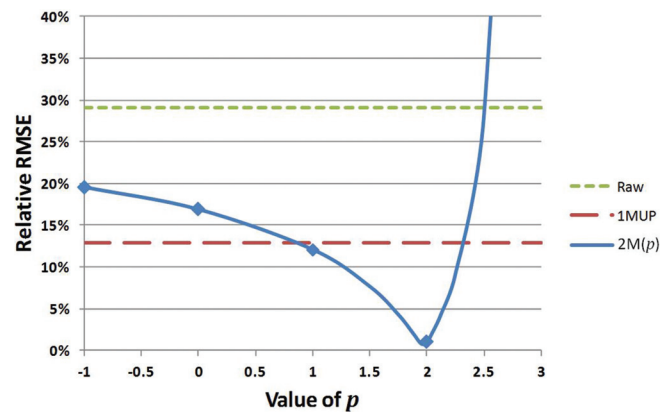


Figure 8 Relative RMSE for different values of p for all tiles (CFD with 24 in. [0.6 m] plenum).

1MUP method are also included. Clearly, an exponent near 2, which corresponds to the physics-based model, provides the best accuracy. However, smaller values of p , which may provide greater robustness, provide reasonable accuracy over the range shown. Error increases quickly for values of p larger than 2.

DATA CENTER AIRFLOW MEASUREMENTS

Tile airflow measurements were taken in a 7400 ft² (687 m²) data center located in Shrewsbury, MA. Figure 9 shows the layout of the 138-rack facility (Figure 9a) and the primary area of focus for measurements discussed here (Figure 9b). Cooling airflow is supplied along the left and right sides of the room as shown. Because all cabling is run overhead and there is no piping or other infrastructure under the raised floor, the 2 ft (0.6 m) deep plenum is empty (except for floor stanchions) and there are no cable cutouts or other significant openings in the floor. There are six 56% open area tiles and one 68% open area tile (Tile I25); the remaining 185 tiles are 25%

open area. Additional details of the data center can be found in the publication by Pardey et al. (2015).

The airflow rate of each tile in the cold aisle, except for A12 and A13, shown in Figure 9b, was measured using the methods discussed previously. Tiles A12 and A13 were partially obstructed by rack hardware so that neither passive nor PFH measurements could be taken.

Additionally, tiles I17 (25% open area), B1 (56% open area), and I25 (68% open area) were selected for additional study in order to evaluate the variability of velocity measurements associated with the Anemometer Method. Specifically, “no hood” and “hood” measurements were recorded at nine locations on each tile, as shown in Figure 10. The variation in the velocity ratio, normalized by the average velocity ratio over each tile, is indicated by shade. The variation is small for

the 25% open area tile, but increases moving from the 56% to 68% open tiles, respectively. In the latter case, the normalized ratio increases by 64% going from the lowest to highest reading. For the same 68% open area tile, if we consider only the four intermediate measurement locations (as might be a more typical and better anemometer placement), the spread is 29%. As the normalized velocity ratio scales directly with the predicted tile airflow in the anemometer method, large tile airflow measurement error is expected when this method is employed with large-open-area tiles.

Returning to the comparison of measured airflow rates for the cold aisle studied, Table 2 shows the RMSE and bias for each method relative to PFH for all 25%-open tiles. For relative RMSE, we use the Equation 12 approach, where PFH

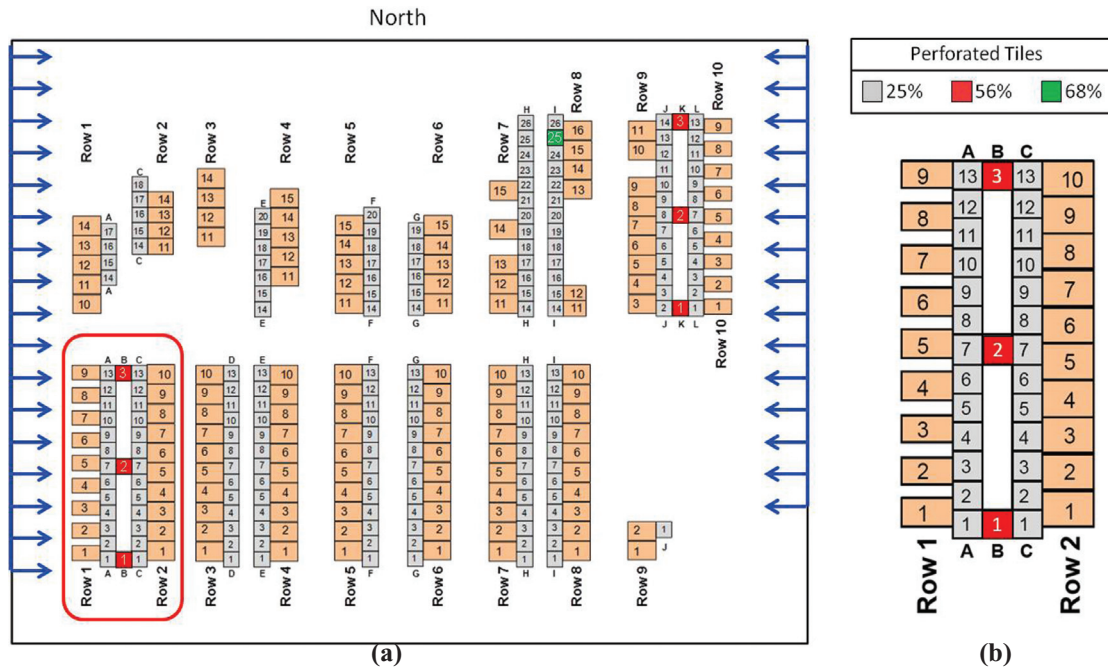


Figure 9 Data center layout: (a) Entire data center and (b) cold aisle studied.

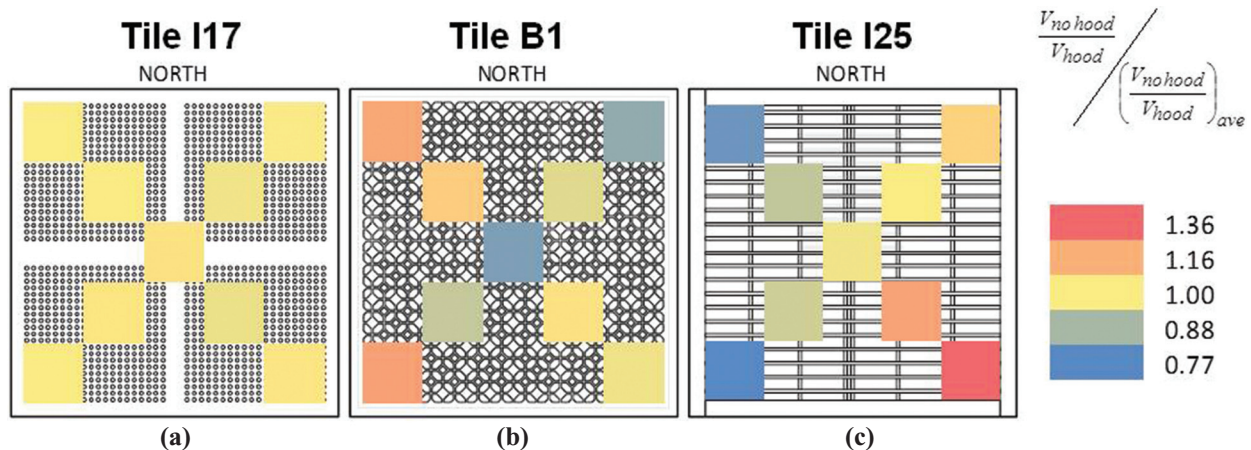


Figure 10 Variation in measurements with anemometer placement: (a) 25% open, (b) 56% open, and (c) 68% open.

measurements are used for the values of $Q_{i,true}$ and $Q_{ave,true}$. The relative bias is defined as

$$\text{Relative Bias} = \frac{1}{n} \sum_{i=1}^n \frac{Q_{i,est} - Q_{i,PFH}}{Q_{ave,PFH}} \quad (13)$$

where $Q_{i,est}$ is the tile airflow rate computed by the method, $Q_{i,PFH}$ is the tile airflow rate provided by the PFH, and $Q_{ave,PFH}$ is the average airflow rate of the tiles measured by the PFH. (As discussed, PFH results were not obtained for the 56% and 68% open tiles.) As with the CFD simulations, we include the raw (uncorrected) flow hood measurement for comparison. Owing to its flow resistance and as expected, the uncorrected flow hood reports measured airflows that are consistently lower than the PFH value. We also note that all methods underpredict the PFH value by at least 2.5%. It is not clear how much of the bias of the methods is due to the correction methods themselves or measurement error associated with the PFH. While the PFH is believed to provide the most accurate measurements obtained, it is reasonable to conclude that it has some nonnegligible measurement error. Even with the fairly restrictive 25% open area tiles, the ideal, room-pressure environment created just above the tile by the PFH is likely not a perfect representation of the environment just above the tile in its normal, unmeasured state. Additionally, the relative bias is very large compared to the overall relative RMSE for most methods. This suggests that most of the relative RMSE is due to underprediction.

The $2M(p=2)$ method again provides the highest accuracy with an RMSE of 3.5%, which is very similar to the RMSE determined in the CFD simulations. The next best agreement is provided by the 1MUP method, which achieves an RMSE of 5.4%—better than that achieved in the CFD simulations. The vane anemometer method provides a meaningful improvement over the raw measurements with an RMSE of 7.6%. The worst accuracy is achieved using the hood's built-in BPC method. The RMSE of 11.3% is still a

meaningful improvement relative to no correction; however, the other methods perform substantially better and, in the case of the 1MUP method, are slightly simpler. As stated earlier, we assume that the BPC algorithm was selected with general HVAC and not data center applications in mind.

Figure 11 shows a tile-by-tile airflow comparison between the $2M(p=2)$ and PFH methods. Note that the $2M(p=2)$ method largely captures the trend of the PFH measurements with a small bias towards underprediction, as also indicated in Table 2.

Figure 12 compares the measured tile airflow rates of the three 56% open area tiles estimated by four selected methods. The PFH could not be used with 56% open tiles. As explained previously, the built-in BPC produced only an error message with 56%-open tiles and all other 2M methods are excluded in favor of the expected best, the $2M(p=2)$ method. All of the correction techniques predict much larger

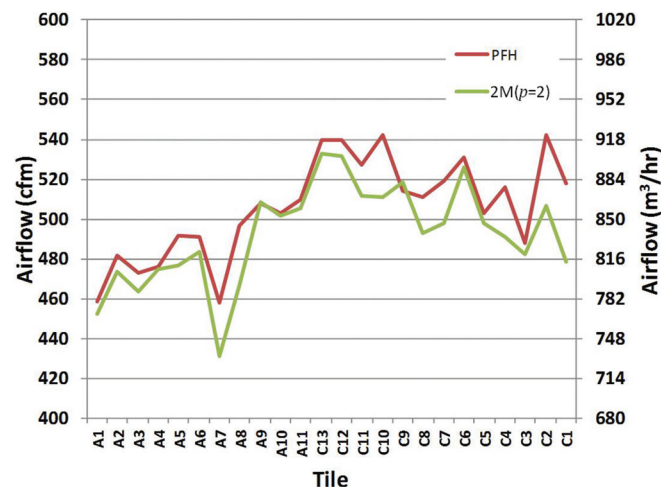


Figure 11 Comparison of $2M(p=2)$ method and PFH measurements for 25%-open tiles.

Table 2. Relative RMSE and Bias of Methods versus PFH Airflow for 25%-Open Tiles

Method	RMSE	Bias
Raw	17.4%	-17.3%
1MUP	5.4%	-5.1%
Vane Anemometer	7.6%	-7.4%
BPC	11.3%	-10.7%
$2M(p=-1)$	8.2%	-8.0%
$2M(p \rightarrow 0)$	7.0%	-6.8%
$2M(p=1)$	5.4%	-5.0%
$2M(p=2)$	3.5%	-2.6%

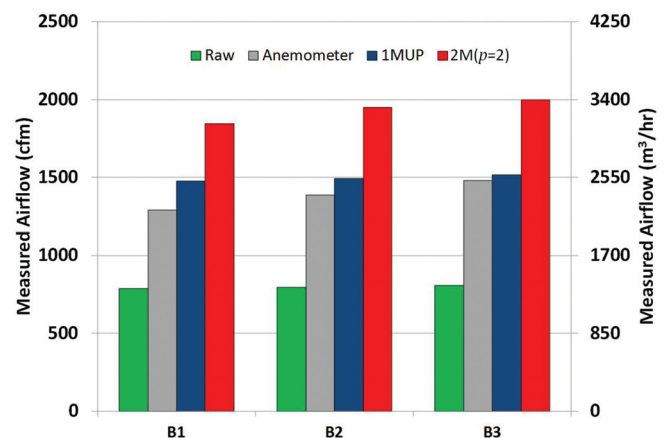


Figure 12 Comparison of tile airflow predictions for 56%-open tiles.

tile airflow than the raw hood measurement. The 1MUP and vane anemometer predictions are similar to one another and are both substantially lower than those of the 2M($p = 2$) method.

SUMMARY OF TILE AIRFLOW MEASUREMENT METHODS

Table 3 provides a summary and comparison of the tile airflow measurement methods discussed in the study. The PFH is impractical for measuring many or all tiles in a typical data center. The vane anemometer method is moderately less convenient and slower than the other passive-hood methods because additional measurements are required at floor level. It may also provide inconsistent accuracy that is affected by the choice of where to locate the anemometer on the tile. The remaining methods are all fast and simple; however, the 2M methods may require a minor modification to commercially available hoods in order to be implemented.

In terms of accuracy, all of the methods provide reasonable improvements relative to raw hood measurements for more-restrictive 25% open tiles. However, as predicted in the CFD simulations, the 2M($p = 2$) method is capable of much better accuracy than the other methods for larger open area tiles.

CONCLUSION

We investigated the accuracy of perforated tile airflow measurements in data centers. Several methods for correcting for the error-inducing effect of a passive flow hood were considered, including two methods already in the literature which and referred to in this manuscript as the 1MUP and anemometer methods. Additionally, we constructed a purpose-built PFH that is believed to be capable of measuring airflow from more restrictive tiles (e.g., 25% open area) more accurately than all passive-hood methods, though it is impractical for large-scale data center assessments. We propose a family of two-measurement passive-hood correction techniques and investigate the accuracy of all methods using CFD studies, laboratory experiments, and actual data center measurements. We find that some type of correction for flow-hood resistance is required and that existing correction methods are in error by about 5% to 10% for 25% open and upwards of 30% for 56% open area tiles based on relative RMSE for our CFD simulations and data center measurements. The best-performing correction method is the 2M uniform pressure (2M[$p = 2$]) method, which can be implemented with existing commercially available flow hoods. This method reduces the error for 25% and 56% open area tiles to 1% to 3% and 1% to 5%, respectively, in the examples considered in this study.

Table 3. Summary of Tile Airflow Measurement Methods

Method	Measurements	Ease of Use	Observed Range of Relative RMSE			Comments
			25% Tiles CFD	56% Tiles CFD	25% Tiles Measured	
1MUP	1 hood airflow	Easy	8% to 10%	18% to 29%	5%	
Vane Anemometer	1 hood airflow 2 anemometer velocities	Moderate	4% to 87%	4% to 21%	8%	Inconsistent accuracy
BPC	2 hood airflows	Easy	N/A	N/A	11%	Unpublished algorithm May fail with large-open-area tiles
PFH	1 hood airflow	Difficult	N/A	N/A	N/A	No correction required May fail with large-open-area tiles Practical for research only
2M($p = -1$)	2 hood airflows	Easy	2% to 5%	29% to 31%	8%	May require slight modification to commercial hood
2M($p \rightarrow 0$)	2 hood airflows	Easy	2% to 5%	25% to 27%	7%	May require slight modification to commercial hood
2M($p = 1$)	2 hood airflows	Easy	1% to 5%	17% to 19%	5%	May require slight modification to commercial hood
2M($p = 2$)	2 hood airflows	Easy	1% to 3%	1% to 5%	4%	May require slight modification to commercial hood May fail with some combinations of measured airflows

NOMENCLATURE

A	=	area normal to flow direction for perforated tile or hood
f_T	=	loss coefficient of perforated tile
f_h	=	loss coefficient of hood (in general)
f_1	=	loss coefficient of hood in normal “no flap” mode
f_2	=	loss coefficient of hood with flap closed
n	=	number of tiles
$Q_{ave,PFH}$	=	volumetric airflow rate measured by PFH, average for all tiles
$Q_{ave,true}$	=	volumetric airflow rate, average for all tiles as predicted by CFD
$Q_{i,est}$	=	volumetric airflow rate of tile estimated by correction method
$Q_{i,PFH}$	=	volumetric airflow rate measured by PFH
$Q_{i,true}$	=	volumetric airflow rate of tile estimated, “actual” as predicted by CFD
Q_T	=	volumetric airflow rate of perforated tile
Q_1	=	volumetric airflow rate measured by hood with f_1
Q_2	=	volumetric airflow rate measured by hood with f_2
V	=	velocity
α_h	=	flow resistance coefficient of hood (in general)
α_T	=	flow resistance coefficient of perforated tile
α_1	=	flow resistance coefficient of hood in normal “no flap” mode

α_2	=	flow resistance coefficient of hood with flap closed
ΔP	=	pressure drop
ρ	=	density of air

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