AIRFLOW AND EQUIPMENT COOLING

WHITE PAPER

The Effects of Doors on Airflow and Equipment Cooling in IT Equipment Cabinets

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The ability to run an efficient data center can be a challenge especially when you are dealing with legacy installations while also planning for future applications. The principles of data center design for effective thermal management with high density data communications equipment heat loads are frequently violated, and data center managers typically suffer through these violations through no fault of their own. More often than not these violations either come via an acquisition of previously developed space or habitation of mature space for yesterday's heat loads. Fortunately there are a number of standard practices and a few creative patches and band aids to which data center managers have access for minimizing or neutralizing the resultant hot spots from these violations. These patches range from adding high static pressure blowers to the bottom spaces of equipment cabinets to plugging all sources of bypass air to creating barriers to hot air re-circulation, such as internal cabinet air dams, cabinet top return air isolation panels, and closed duct return air paths, to adding floor fans to deliver more cold air to the fronts of cabinets. While there is situational merit to all these approaches, removing high-flow perforated cabinet front doors or tasking cabinet vendors to deliver even greater percent-open mesh doors do not represent viable patches for improving the thermal performance of a data center.

As counter-intuitive as it may seem, there appears to be no realizable value in improving the percent-open beyond the 63% that most electronic equipment manufacturers have specified in the guidelines for deploying their equipment in third party cabinets. According to the information on pressure drops through different percent-open metal mesh materials developed by the material vendors, those server OEMs did not just make up that 63% open requirement – they settled on the optimum balance between maximum physical security and maximum airflow.

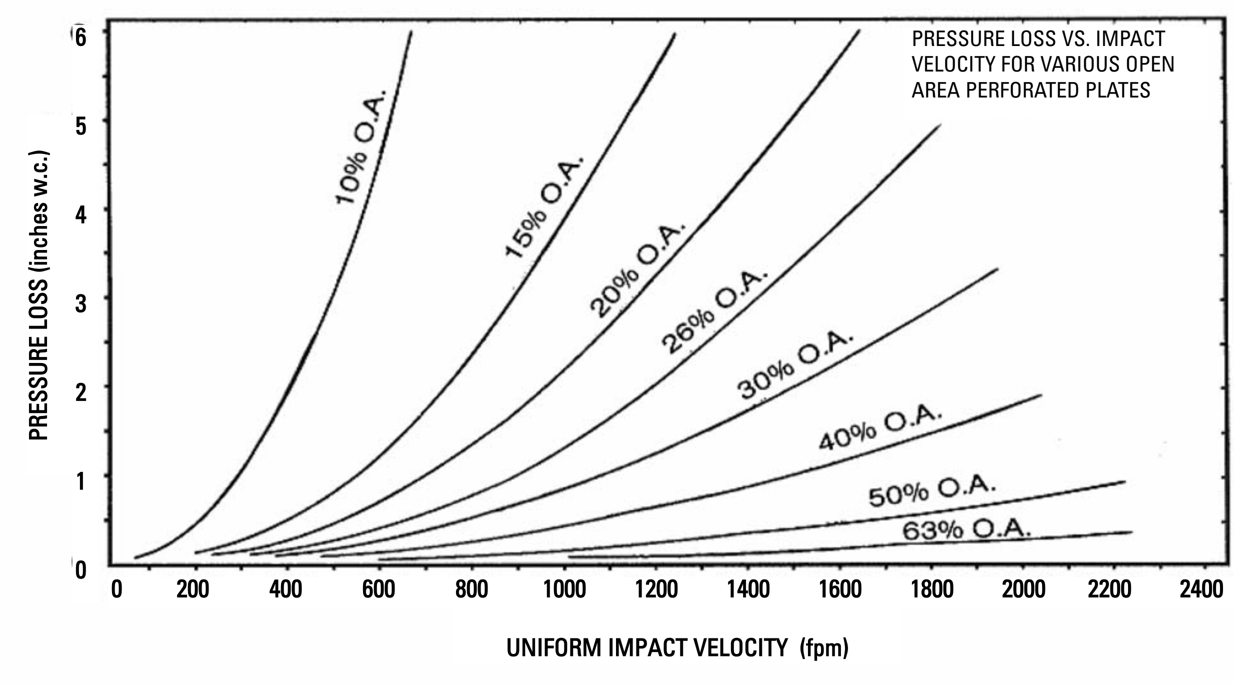


Figure 1, Source: Designers, Specifiers and Buyer's Handbook for Perforated Metals, The Industrial Perforators Association, 1993

Figure 1 from The Industrial Perforators Association clearly shows significant improvement in pressure loss over the cubic feet per minute (CFM) curves seen in most data center cabinets, up to the 63% open level where the pressure-drop curve is practically a straight line.

As counter-intuitive as it is that more open space would not necessarily allow for greater airflow, Chatsworth Products, Inc. took advantage of an opportunity to verify this anomaly in a customer's data center during a confirmation audit to confirm various corrective actions had actually delivered the anticipated cooling benefit, by monitoring airflow through equipment and through a cabinet with various percent-open door configurations. A little background on this particular data center problem and a general review of the principles for cooling servers will assist in understanding the data and its importance.

First, there are various ways to describe the cooling that happens with server equipment in cabinets. One description is the equation for forced air convection heat transfer:

Convection heat transfer from a surface to an airflow is governed by the equation $Q = hA(T_w - T_f)$

Where Q = the heat transfer rate
h = the convection heat transfer coefficient
A = the surface area
T_w = the temperature of the surface
T_f = the temperature of the fluid

Another way to describe this cooling is the equation for sensible cooling:

$$BTU = (\Delta T \cdot CFM \cdot 1.08)$$

Where ΔT = the difference between source air and return air

This answer can be converted into kW merely by dividing by 12,000

And finally, the equation for CFM describes this same relationship:

$$Q = \frac{1.67W}{C_t}$$

Where Q = CFM
C_t = Temperature rise across the equipment in Celsius

It is instructive to review what these equations are actually describing. In the heat transfer equation, every factor except the Tf is basically a constant defined by the manufacturer, so the only variable over which the data center manager has any control is the temperature of the input air. In the sensible cooling equation, every factor is controlled by the equipment performance specification – the CFM is controlled by the fans in the servers (this will only vary to the degree those fans are variable speed or to the degree that those fans are choked by an inadequate air supply) and the ΔT is a factor of the equipment and the air crossing it. While there are effects the data center manager can apply to sensible cooling at the room level, there is nothing variable about sensible cooling at the equipment level. In the CFM equation, again, that temperature rise is a constant defined internal to the server. In summary, server fans will pull air from somewhere and the only variable over which the user has any control is the temperature of the air those fans will be drawing.

In the data center under study, a site audit revealed that the high density blade PCs were consuming more air than the room air handlers were producing and hot spots were being created in cabinets because hot exhaust make-up air was being drawn over the tops of cabinets and introduced into the blade servers located toward the tops of those cabinets. The patches that were deployed included sealing off bypass air in the room and replacing the perforated floor tiles with 50% open floor grates to increase the amount of chilled air that was delivered into the cold aisles and then building a barrier between the hot and cold aisles so that any make-up air that the servers drew would at least not be from the hottest air in the room. The audit, which included actual microprocessor temperatures, revealed the patches were successful.

Furthermore, the audit provided an opportunity to test the validity of the "63% Solution." The test was run in a raised floor data center to quantify the significance of a server cabinet door percent open area relative to the net flow through the cabinet. Temperature and air velocity data were acquired at the exhaust of Clear Cube blade PC's loaded in the cabinet. Intake air temperature was also acquired.

At the flow rates tested (~2800 CFM through a 7' MegaFrame cabinet), the results from the acquired data suggested that increasing the percent open area of the door perforation above 63% does not significantly affect the net airflow rate through the blade server.

CPI M-Series MegaFrame® Cabinets are used to house Clear Cube R-series blade PCs. The R-Series Clear Cube system allows 8 blade PCs to be installed into a 3 RMU chassis. For this customer's installation, up to 12 of the 8-blade chassis were installed into a 7' MegaFrame cabinet. These units require a peak inrush current of 10A, and a nominal draw between 4.5-6.5A is typical for each chassis. The actual electrical current supplied to the cabinet could not be measured; however, based on the product literature, the cabinet load is estimated to be within the range of 6.5kW and 9.4kW.

The test cabinet selected had the blade PC chassis installed in RMU 10 through 45.

Airflow and temperature measurements were acquired using a turbine type blade anemometer at the rear of the cabinet and at the server inlets.

Figure 2

Results: The data collected during the test is summarized in Tables 1 and 2 below. An unexpected result observed in the measurement was that the server exhaust temperatures actually decreased with the 63% door closed, which was attributed to the timing differential between tests coinciding with both the heat of the day and the heavier transaction load on the PCs.

Table 1

			BLADE CHASSIS FAN				
			1	2	3	4	Chassis Total
Case 1 U 43-45	Exh. Fan	°C	31.9	33.0	32.4	31.4	
		°F	89.5	91.4	90.3	88.6	
		FT/Sec	21.4	21.9	22	22.1	
		CFM	53.93	55.19	55.44	55.69	220.25
	Intake Surf. Temp	°C	22.8	21.1		22.2	
		°F	73.0	70.0		72.0	

			1	2	3	4	Chassis Total
Case 6 U 30-32	Exh. Fan	°C	21.3	22.7	26.2	28.8	
		°F	70.4	72.8	79.1	83.9	
		FT/Sec	22.1	21.1	21	21.5	
		CFM	55.69	53.17	52.92	54.18	215.97
	Intake Surf. Temp	°C	13.9	13.3		17.2	
		°F	57.0	56.0		63.0	

			1	2	3	4	Chassis Total
Case 12 U 10-12	Exh. Fan	°C	21.2	22.9	23.4	22.0	
		°F	70.2	73.3	74.2	71.6	
		FT/Sec	21.5	23.1	22.5	23.1	
		CFM	54.18	58.21	56.70	58.21	227.31
	Intake Surf. Temp	°C	12.8	13.3		15.6	
		°F	55.0	56.0		60.0	

Table 1. 100% Open Area Measurements (Door Removed)

Table 2

			BLADE CHASSIS FAN				
			1	2	3	4	Chassis Total
Case 1 U 43-45	Exh. Fan	°C	30.7	31.7	32.2	30.7	
		°F	87.3	89.1	89.9	87.3	
		FT/Sec	21.2	21.2	21.4	22.1	
		CFM	53.42	53.42	53.93	55.69	216.47
	Intake Surf. Temp	°C	25.6	23.3		23.9	
		°F	78.0	74.0		75.0	

			1	2	3	4	Chassis Total
Case 6 U 30-32	Exh. Fan	°C	20.7	21.8	24.6	26.7	
		°F	69.3	71.3	76.3	80.1	
		FT/Sec	21.5	21.8	20.2	21.7	
		CFM	54.18	54.94	50.90	54.68	214.71
	Intake Surf. Temp	°C	15.0	13.9		20.0	
		°F	59.0	57.0		68.0	

			1	2	3	4	Chassis Total
Case 12 U 10-12	Exh. Fan	°C	21.7	23.3	23.4	22.6	
		°F	71.0	74.0	74.1	72.6	
		FT/Sec	22.1	22.6	22.1	22.1	
		CFM	55.69	56.95	55.69	55.69	224.03
	Intake Surf. Temp	°C	15.6	15.0	16.1	17.2	
		°F	60.0	59.0	61.0	63.0	

Table 2. 63% Open Area Measurements (Door closed)

CPU temperatures and fan speeds were monitored directly though the PCs operating system and showed no improvement (i.e., operating temperature decrease) with the change from 63% open doors to a 100% open situation (i.e., no door).

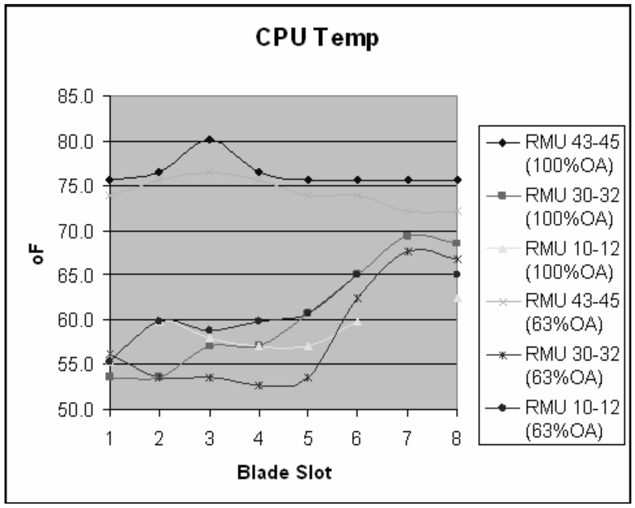


Figure 3

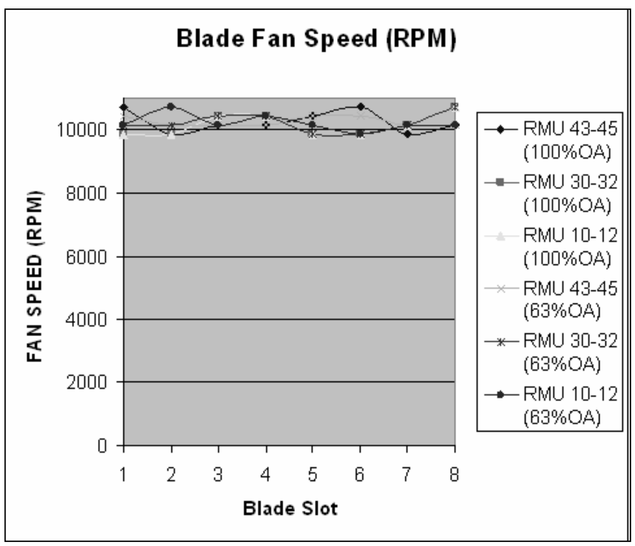
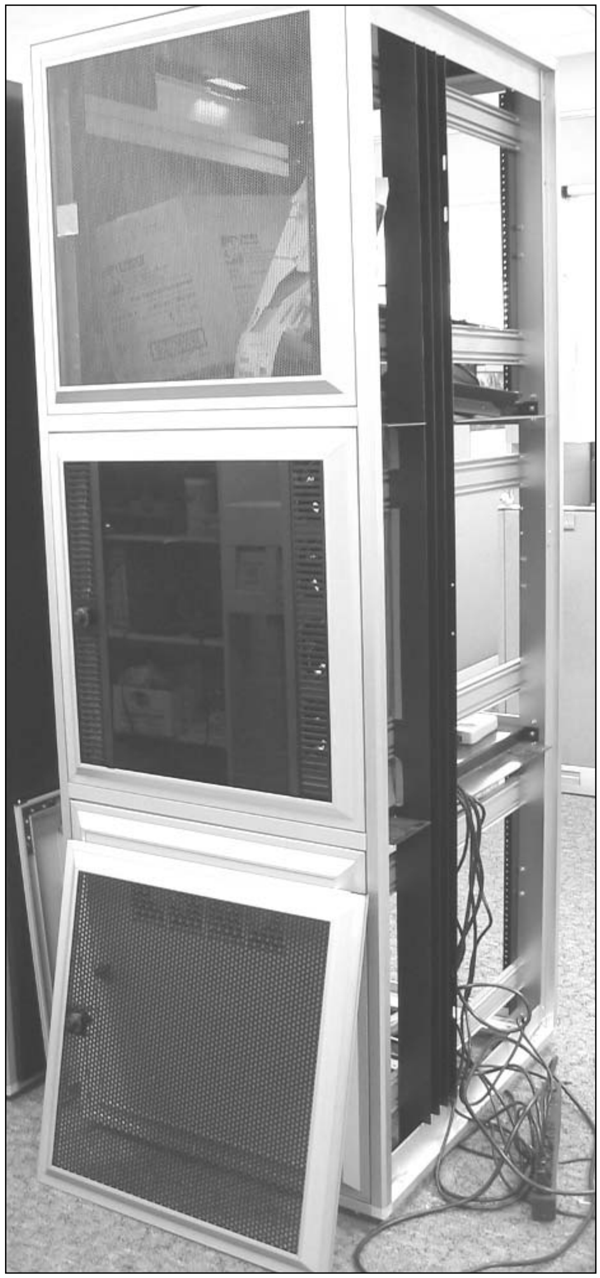


Figure 4

A subsequent controlled test was designed and executed to verify the anticipated poorer performance at less than 63% and to double check the results from the data center test. In these tests, only restriction to airflow was considered, having already established CPU temperatures are only affected by input air temperature, assuming the server fans are not choked.

Test Approach

The test environment was the center section of an empty multi-compartment co-location cabinet, shown in the picture below. Four different door configurations were tested: 8% open (vented plexiglass), 40% open (perforated metal), 63% open (perforated metal), and 100% open (door off).



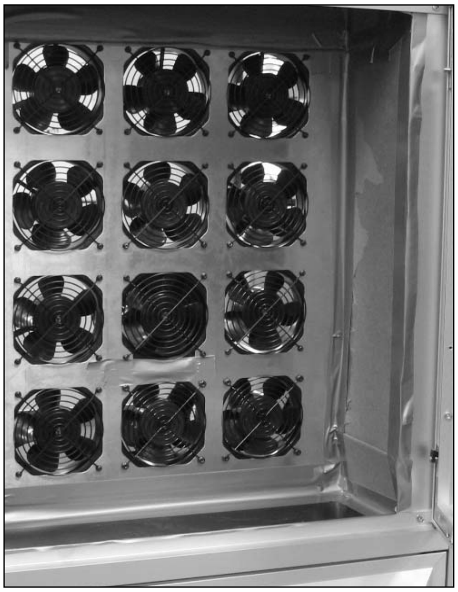


Figure 5

A plate was fabricated to hold 12 fans within the 14 RMU available in the center cabinet section. Cardboard and duct tape were used to seal off all of the leaks (See Figure 2). Each of the fans was capable of ~100 CFM in a free condition.

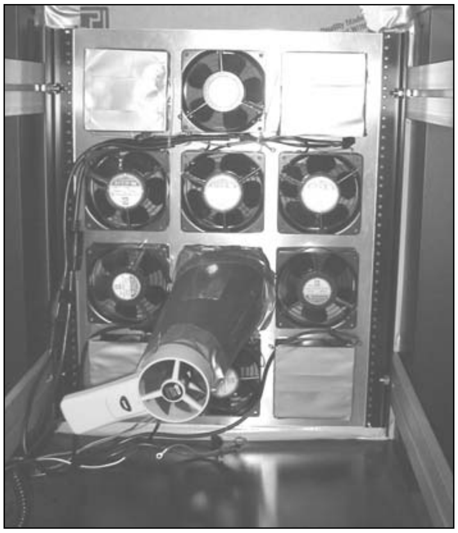


Figure 6

In the rear of the cabinet we attached a horn to a turbine anemometer so that we could measure flow through one of the fans (See Figure 6). Data was taken with 12, 10, 8 and 6 fans active in the cabinet to simulate flows of ~3600, 3000, 2400, and 1800 CFM through a 45 RMU cabinet. The inactive fans were duct taped over to prevent leakage for the tests with fewer than 12 fans active.

Fans Active		12			10			8			6		
		Open	Closed	% Flow Reduction	Open	Closed	% Flow Reduction	Open	Closed	% Flow Reduction	Open	Closed	% Flow Reduction
Vented Plexi	FT/Sec	37.5	16.4	56.3%	37.3	18.4	50.7%	38.4	23.2	39.6%	37.7	25.0	33.7%
(8% Open)	FT/Sec	37.5	16.2	56.8%	37.3	18.0	51.7%	38.2	23.0	39.8%	37.4	25.0	33.2%
Est Flow/RMU	CFM	79.5	34.6	56.5%	65.9	32.2	51.2%	54.2	32.7	39.7%	39.8	26.5	33.4%
Simulated Flow/ 45 RMU CAB	CFM	3580	1556		2967	1448		2437	1470		1792	1193	
40% Open	FT/Sec	37.5	34.6	7.7%	37.2	34.8	6.5%	37.8	36.7	2.9%	37.3	36.5	2.1%
Perf	FT/Sec	37.8	34.8	7.9%	37.0	35.0	5.4%	37.8	36.7	2.9%	37.0	36.4	1.6%
Est Flow/RMU	CFM	79.9	73.6	7.8%	65.6	61.7	5.9%	53.5	51.9	2.9%	39.4	38.7	1.9%
Simulated Flow/ 45 RMU CAB	CFM	3594	3312		2951	2776		2406	2336		1773	1740	
63% Open	FT/Sec	37.5	37.0	1.3%	36.8	36.2	1.6%	37.7	37.4	0.8%	37.0	36.8	0.5%
Perf	FT/Sec	37.5	36.8	1.9%	36.8	36.2	1.6%	37.7	37.3	1.1%	37.0	36.8	0.5%
Est Flow/RMU	CFM	79.5	78.3	1.6%	65.1	64.0	1.6%	53.3	52.8	0.9%	39.2	39.0	0.5%
Simulated Flow/ 45 RMU CAB	CFM	3580	3522		2927	2880		2399	2377		1766	1756	

Table 3 shows the data collected during the test.

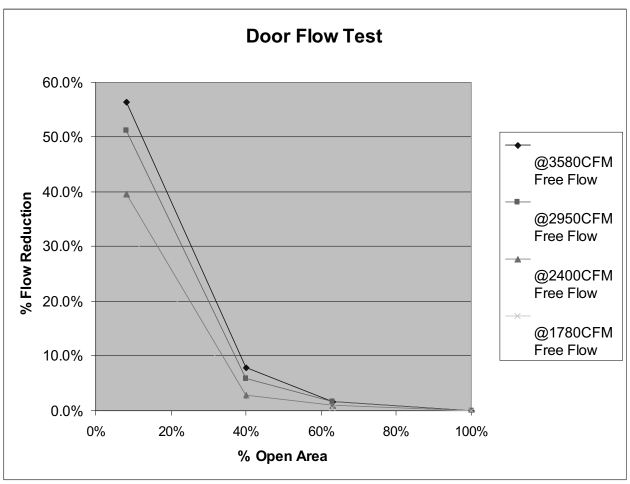


Figure 7

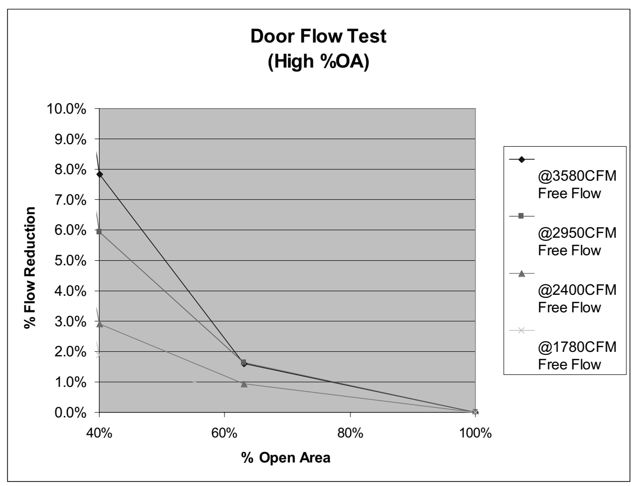


Figure 8

Figures 7 and 8 show the relative effect of percentage flow reduction versus door percentage open area at various free flow conditions.

Results:

The data confirms the findings from the less controlled data center tests and specifically indicates that we can expect to see a maximum flow efficiency improvement of only 1.6% in going from 63% open to 100% open in moving 3600CFM through a 45 RMU cabinet. At 1800 CFM, the improvement drops to 0.5%. As a point of reference, a 7' cabinet fully populated with Dell 1855 PowerEdge blade chassis would require 3120 CFM of flow capacity (6 - 7 RMU chassis @, 520 CFM each). If that reduction in airflow actually represented choking the server fans, it would represent a total airflow of 3070 CFM. At the maximum PowerEdge configuration load of 26.6kW in a cabinet, using the airflow equation of Q=1.67•W/Tc, the 63% open door would deliver "hypothetical" cooling of only 0.23ºC less than the 100% open solution – statistically and practically inconsequential. Or, that difference for a 70% open door would only be 0.04ºC.

In conclusion, testing in both a controlled environment and in a live data center confirmed The Industrial Perforators Association's data that there is no meaningful airflow improvement to be achieved beyond 63% open perforated server cabinet doors.