数据中心高效制冷：紧靠热源的行级制冷解决方案

第 137 号白皮书

版本 1

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> 摘要

随着近年来信息科技不断地发展，数据中心中热负荷密度呈不断增长的趋势。随着功率密度的增加，为高功率负载制冷的难度也显著地加剧。近年来，传统数据中心制冷系统设计事实上已经无法有效地移除密集负载所产生的热负荷（每机柜 20kW 或者更高）。这种情况已经推动了数据中心制冷系统架构的变革和创新。一种创新制冷架构的出现，专为解决高密度制冷难题而设计，能够显著地提升数据中心效率。本文讨论行级制冷优于其它两种常见的传统制冷系统架构，及其在能效方面的巨大收益。

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精华导读

简介

本篇白皮书最初发表于美国采暖制冷与空调学会（ASHRAE）期刊，2008 年 10 月。本白皮书将收录发表于 ASHRAE 期刊的英文原件，在英文原件之前提供的中文精华导读以方便中文读者阅读。

从数据中心诞生开始，其制冷架构一直广泛采用房间级机房空调或机房空气处理装置与高架地板下送风的组合。这种方式是指将机房空调置于机房的周边，冷风经过高架地板下的风道，再穿过地板砖上的穿孔或栅格进入机房。在功率密度较低（每机柜 1-5 kW）的情况下，这种方式可以为 IT 设备提供充足的制冷，尽管这时在机房范围内冷风和热风发生混合。另一种相似的机房制冷方案是中央空气处理装置，这种系统使用容量更大的中央制冷装置，同样采用高架地板或吊顶风管送风。这两种制冷架构都属于房间级制冷系统架构的范畴。然而随着单台机柜功率密度的不断升高，传统房间级制冷系统正面临来自高热负荷密度，制冷系统能效和总拥有成本等方面的挑战。其本身的结构决定其无法有效地满足未来需求的发展。

本文介绍紧靠热源的行级制冷解决方案能够有效地应对传统制冷系统架构所面临的来自高热负荷密度，制冷系统能效和总拥有成本等方面的挑战。文中还比较行级制冷相对于传统房间级制冷与创新的优势所在，并通过对制冷系统建模分析对能效和成本方面的收益进行量化。

传统制冷系统面临的挑战

当机柜密度超过每台 5 kW 且功率密度仍在不断增长时，采用房间级制冷系统配送冷风和排出废热所面临的挑战就越发明显。

1. 传统的房间级制冷架构最受人诟病的弱点就是送风距离。制冷装置和产生热负荷的设备之间的长距离使得用于冷却 IT 设备的冷风和 IT 设备排除的废热很难不发生混合。这就会导致“过热点”的出现和送回风系统设计过于复杂。
2. 冷却 IT 设备所需要的风量也随着功率密度的升高而增大。房间级制冷架构送风和回风经过距离很长的风道（高架地板或吊顶），其用于保持压力和抵消阻力所消耗的风机功耗是非常巨大的。
3. IT 设备所需的净冷量是由其所产生的热负荷决定的，但是由于冷风和废热发生混合而损失的冷量，为了补偿这部分损失的冷量又会增加压缩机和风机的负载。造成设备容量的过度规划和能耗进一步升高，带来设备投资和能源成本的增加。

什么是紧靠热源的行级制冷解决方案?

为了应对房间级制冷系统配送冷风和排出废热方面所面临的挑战，紧靠热源的行级制冷架构设计方案应运而生。为了解决制冷设备与产生热负荷的热源距离过远的问题，行级制冷架构将空调单元置于机柜行列之内。作为冷热通道布局设计的一部分，冷风由机柜前部进入，冷却 IT 设备之后成为废热排入机柜后部的热通道。废热随后被吸入行级空调进行冷却，然后再由风机送至机柜前部的冷通道（见图 **1**）。

紧靠热源的行级空调与冷热通道的机柜布局方式非常契合，通过独有的控制逻辑能够最大限度地保持机房内环境温湿度的稳定性，而且其出众的灵活性使其容量与 IT 负载接近完美的匹配，减少过度规划造成浪费。此外，融入机柜行的摆放方式可以将冷风和废热混合减至最低，提高冷量的使用率，即实际用于冷却 IT 设备的冷量与总冷量之比。

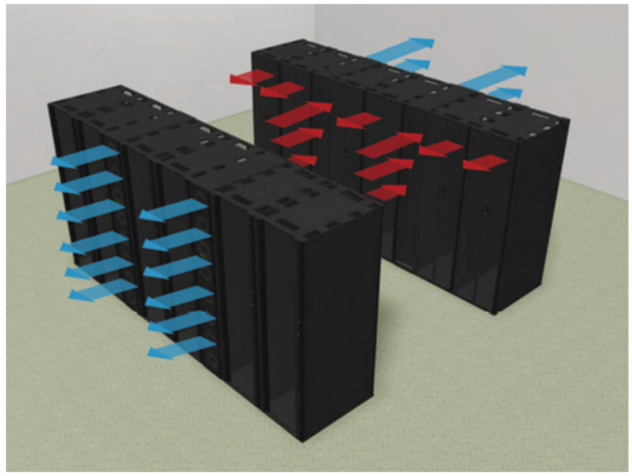

图 1

行级制冷架构，从机柜后部热通道吸入热风，冷却后将冷风送至前部冷通道

紧靠热源的行级制冷解决方案的优势

与传统的房间级空调相比，紧靠热源的行级制冷解决方案主要有以下几个优势：

缩短空调送风和回风路径 —将机房空调安装在机柜行列间将会缩短送风和回风的路径使制冷设备仅靠热源。这种方案采用更合理的方式收集废热和配送冷风，供回风过程中由于路径缩短导致风阻减少和保持压力的要求降低，可以大幅降低风机的功耗。这种与生俱来系统架构优势是房间级制冷系统所不具备的，可以带来能效方面的收益。而且在配合变频调速风机使用时，房间级空调为了保持高架地板下的静压和冷风配送的均匀性，风机多数时间维持在定速运行。使用变频调速风机节能的初衷无法实现。而行级空调靠风量来维持静压和均匀性的要求低，变频调速风机节能的效果更加明显。

减少冷风与热风混合 — 由于送风路径的减少，冷热风混合的几率也就变小，这主要会有以下两个方面的节约能源和成本效果：

- 首先，当废热被更好、更快的捕获，冷热风混合就会减少，这样就可以设定较高的机房空调送风温度。送风温度与机柜前的进风温度更为接近，因此避免为了补偿由于冷热风混和所损失的冷量而设定较低的机房空调送风温度。因此压缩机需要提供的冷量也会降低，达到节能的效果。
- 其次，冷热风混合减少，所需要配送的冷风风量也会减少。由于风机的风量与转速的一次方成正比，风压与转速的二次方成正比，而功率与转速的三次方成正比，所以变频调速风机根据风量降低转速，可以带来成倍的能耗节约。
- 最后，每台机房空调所能提供的冷量也会变大。当废热被更好、更快的捕获，冷热风混合就会减少，回风的温度提高。较高的回风温度可以实现空气与制冷剂或冷冻水盘管之间的温度梯度更大，换热效率因此升高，所以整个系统移除机房内热量的效率就会提高，总制冷功耗就会下降，达到节能的效果。

高显热比 —紧靠热源的行级制冷解决方案具有很高的显热比。这是因为当送风温度提高，可以避免空调内的冷却盘管结露导致的被动除湿。减少被动除湿和补偿加湿将使行级空调的冷量更多地用于冷却 IT 设备的显冷量。制冷系统的显热比越高，系统就越节能。

更匹配真实负载 — 由于行级制冷解决方案紧靠热源，其与负载或者说与热负荷的匹配就更紧密和精确。制冷系统通常按最差工况进行设计，行级制冷系统过度规划选型的现象要远远低于房间级制冷系统。

紧靠热源的行级制冷解决方案分析

在比较用于关键制冷的房间级解决方案与行级解决方案时，使用制冷系统消耗的功率与 IT 设备所产生的总热负荷之比（**公式 1**）作为衡量两种房间级制冷解决方案和紧靠热源的行级制冷解决方案的效率优劣关键的系数。能效系数的值越低就说明制冷系统的能效越优。

$$\text{Efficiency}_{\text{Metric}} = \frac{\text{Cooling Power }[\text{kWh}]}{\text{IT Power }[\text{kWh}]} \qquad \text{（公式 1）}$$

作为以上能效系数分子的制冷系统功率可以由整个制冷系统回路的所有组件的功耗进行累加得出。这些组件包括空调风机，冷冻水水泵，冷水机，冷却水水泵和冷却塔等。如今，变频调速装置在制冷系统风机和水泵应用非常广泛，但是本分析致力于解释房间级制冷系统与行级制冷系统最根本的区别，既送风路径缩短，冷热风混合减少，高显热比以及适度规划带来的能效节约，所以在建模时风机和水泵为定速，则风量和冷冻水流量为定值，而冷水机采用变频驱动装置的螺杆式压缩机。其效率在室外温度不同的工况下，当冷却水温度较低时压缩机效率较高。（效率值参考开利公司的冷水机组）。

表 1

不同工况下冷水机效率系数 λ_J 的值

冷却水温度范围	J	运行时长（每年 8760 小时）	λ_J
29.5-32.2	1	194	0.18
26.7-29.4	2	916	0.16
23.9-26.6	3	1353	0.14
21.2-23.8	4	894	0.12
18.4-21.1	5	1041	0.11
15.6-18.3	6	932	0.09
12.8-15.5	7	1234	0.08
12.7 及以下	8	2196	0.07

注：运行时长计算基于测试所在地美国圣路易斯当地的气象数据

风机和水泵轴功率计算公式如**公式 2** 所示：

$$\text{Power}_{\text{Shaft}} = (\dot{V} * \rho * g * H)/(\eta * 1000)[\text{kW}] \qquad \text{（公式 2）}$$

机房空调送回风温度与送风温度关系如**公式 3** 所示：

$$\text{LAT} = \text{EAT} - \dot{Q}_{\text{NetSensible}}/(\dot{V}_{\text{Air}} * \rho_{\text{Air}} * Cp_{\text{Air}})[^\circ\text{C},(^\circ\text{F})] \qquad \text{（公式 3）}$$

冷水机平均功率为不同室外温度工况下各功率之和的平均值，如**公式 4** 所示：

$$\text{Power}_{\text{Chiller}} = \sum_{J=1}^{8} (\dot{Q}_{\text{Demand}} * \beta_J * \lambda_J)/8760[\text{kW}] \qquad \text{（公式 4）}$$

运算公式所涉及到的参数如下：

Cp_{Air} – 空气比热, 1.022 [kJ/kg • °C]

Cp_{Water} – 水比热, 4.188 [kJ/kg • °C]

ρ_{Air} – 空气密度, 1.173 [kg/m^3]

ρ_{Water} – 水密度, 999.7 [kg/m^3]

η – 风机或水泵的效率
η_{Pump} – 水泵效率取值为 0.65
$\dot{V}$ – 体积流量[m^3/s]
H – 压头损失 [m] （1 m = 98066.5 Pa）
g – 重力加速度系数 9.81 [m/s^2]
LAT – 送风温度 [°C, (°F)]
EAT – 回风温度 [°C, (°F)]
LFT – 冷冻水回水温度 [°C, (°F)]
DB – 干球温度 [°C, (°F)]
WB – 湿球温度 [°C, (°F)]
kWh – 能耗, 千瓦时
$\dot{Q}_{NetSensible}$ – 净显冷量 [kJ•sec]
$\dot{Q}_{Demand}$ – 冷水机总冷量 [kW]
λ_J –冷水机效率系数, 功耗/冷量 [kW/kW]
β_J – 各工况下运行小时数，基于冷却水温度 [小时]

假设数据中心中 IT 设备、照明以及风机等的总热负荷为 750 kW，两种房间级制冷架构和行级制冷架构都采用变频调速驱动的螺杆式冷水机，冷冻水的供水温度为 7°C，电费为每千瓦时 10 美分（$ 0.10）。三种制冷架构所采用的设备规格参数如**表 2** 所示：

表 2

三种制冷架构设备规格参数

行级制冷装置（IRAH）规格参数	机房空气处理装置（CRAH）规格参数	中央空气处理装置（CAHU）规格参数

• 2,700 CFM @ 自由送风
• 显冷量：25.2 kW @干球温度 35°C，湿球温度 20°C
• 显热比：1.0
• 冷冻水流量：1.13 L/s @ 8 m 水压头
• 回风温度：(19.7°C)（**公式 3**）
• 风机功率：1 kW （来自制造商公开资料）
• 冷冻水泵功率：0.34 kW（**公式 2**）
	• 风量：17,100cfm @ 75 Pa 地板下静压
	• 显冷量：113 kW @ 干球温度 24°C 和相对湿度 45%，即湿球温度 16°C
	• 显热比：0.95
	• 冷冻水流量：6 L/s @ 6 m 水压头
	• 每台风机轴功率(3x) $Power_{Shaft}$： 3.2 kW（**公式 2**）
	• 前倾式双进双吸离心风机（DIDW）：38cm x 38cm
	  - 吹风面积（BA） = 0.25 m × 0.47m = 0.12 m^2
	  - 出口面积（OA） = 0.40m ×0.47 m = 0.19 m^2
	  - BA/OA = 0.38m^2 / 0.62m^2 = 0.61
	  - 静效率：0.59
	  - 出口流速：14 m/s
	  - 出口系统压力： 149 Pa
	  - 地板下静压：75 Pa
	  - 过滤网损失：187 Pa
	  - 盘管损失： 162 Pa (有结露)
	  - 蜗壳损失：125 Pa
	• 电机功率：11.0 kW（0.92 效率 * 1.05 驱动损耗）
	• 静显冷量： 102 kW
	• 送风温度：13°C（**公式 3**）
	• 冷冻水泵功率：1.4 kW（**公式 2**）
		• 风量：34,000cfm @ 249 Pa 外部静压
		• 显冷量：220 kW @干球温度 24°C 和相对湿度 45%，即湿球温度 16°C
		• 显热比：0.95
		• 冷冻水流量：10 L/s @ 6 m 水压头
		• 每台风机轴功率 (2x) $Power_{Shaft}$: 9.1 kW（**公式 2**）
		  - 76cm 后倾式双吸离心风机(BIDW)
		  - 出口流速：9 m/s
		  - 静效率：0.66
		  - 回风风管损失：75 Pa
		  - 过滤网损失：187 Pa
		  - 盘管损失：149 Pa (有结露)
		  - 蜗壳损失：87 Pa
		  - 送风风管损失：249 Pa
		• 电机功率：20.8 kW （0.92 效率 * 1.05 驱动损耗）
		• 静显冷量：200 kW
		• 送风温度：14°C （**公式 3**）
		• 冷冻水泵功率：2.8 kW （**公式 2**）

根据以上规格参数可以得出三种制冷架构：

- 热负荷为 750 kW 的数据中心需要 30 台行级空气处理装置，风机的总功率为 30.6 kW，冷冻水泵功率为 10.2 kW，冷水机的总制冷量为 791 kW，显热比为 1。
- 热负荷为 750 kW 的数据中心需要 8 台行机房空气处理装置，风机的总功率为 88 kW，冷冻水泵功率为 11.0 kW，冷水机的总制冷量为 893 kW，显热比为 0.95。
- 热负荷为 750 kW 的数据中心需要 4 台行中央空气处理装置，风机的总功率为 83.2 kW，冷冻水泵功率为 11.1 kW，冷水机的总制冷量为 888 kW，显热比为 0.95。

表 3

三种制冷架构能耗和运行成本比较

制冷系统组件	行级制冷装置	机房空气处理装置	中央空气处理装置
机房空调数量	30 台	8 台	4 台
风机总功率	30.6 kW	88 kW	83.2 kW
冷冻水泵功率	10.2 kW	11.0 kW	11.1 kW
冷水机平均功率（公式 4）	83.9 kW	94.7 kW	94.2 kW
冷却水泵功率	18.5 kW	18.5 kW	18.5 kW
冷却塔功率	16.2 kW	18.3 kW	18.2 kW
制冷系统总功率	159.30 kW	230.5 kW	225.1 kW
制冷系统能效系数（公式 1）	0.21	0.31	0.30
制冷系统年度运行成本	$ 139,572	$ 201,878	$ 197,211

小结

紧靠热源的行级制冷解决方案和两种房间级制冷解决方案的用电成本分别是 $139,572、$201,878 和 $ 197,211。行级制冷架构与其它两种房间级制冷架构相比可以节约 2/3 的风机功耗，并且在制冷循环中的其它各环节取得不错的节能效果。从整体来看，节约 30%制冷系统总功耗。

紧靠热源的行级制冷系统可以节约运行成本，但是现实中并不存在一种药可以“包治百病”的情况。现实中，建成的数据中心的数量远远高于新建的数据中心的数量，采用将本文提到的三种解决方案“混合”部署的架构是更可能的发展方向。在现有数据中心中，随着数据中心的运行和功率密度的不断升高，针对不同功率密度负载的“混合”架构更加实际。了解更多关于“混合”部署的知识，请参阅第 134 号白皮书《在低密度数据中心中部署高密度区域》。但是对于新建的数据中心来说，应该优先考虑高效、可预测的行级制冷解决方案。

资源链接
第 134 号白皮书
在低密度数据中心中部署高密度区域

Energy-Efficient Data Centers

A Close-Coupled Row Solution

By **John Bean,** Member ASHRAE; and **Kevin Dunlap**

The predominant architecture for cooling data centers since the inception of the mainframe has been raised floor air delivery from perimeter computer room air handlers (CRAH). In this approach, CRAHs are placed around the perimeter of the room, and they distribute cold air through a raised floor with perforated floor tiles or vents to direct the air into the room (*Figure 1*). At lower densities (1 to 5 kW/rack) adequate cooling is provided to sensitive IT equipment, despite the mixing of air throughout the room.

A similar air delivery system that has been used to cool data centers is central air-handling units (CAHU) (*Figure 2*). These systems use much larger, more centralized cooling units with similar air delivery to perimeter CRAH cooling of either raised floor or custom overhead ductwork.

As rack power grew beyond 5 kW, air delivery and heat removal challenges with use of CRAH and CAHU systems became evident. The major obstacle in these architectures is the length scale of air delivery. Distance between the cooling units and the heat load make it difficult to properly remove the heat generated from IT equipment without mixing with supply air. This separation results in hot spots and a complicated design approach to air distribution.

To add to this problem, the airflow demands of the IT equipment also increases with power density. Since CRAH and CAHU systems use a plenum for supply air delivery (and warm air return in certain designs), a significant amount of fan horsepower is required to pressurize the plenum and overcome resistances in the air-distribution system. Additionally, to overcome the effects of mixing the net volume of air circulated is significantly greater than actual air volume required

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by IT equipment further compounding energy consumed by fans.

To address the air delivery and heat removal challenges of CRAH and CAHU systems, row-based cooling systems have begun to appear in many data center designs (*Figure 3*). To address the separation of cooling units and heat loads, row-based designs place the air-conditioning units in the row of rack enclosures. Incorporating a hot/cold aisle design, heat is removed from the hot aisle as it is dispelled from the IT equipment. The hot air is then cooled and discharged to the cold aisle.

While row-based designs addressed the issue of proper heat removal and cold air supply, they also brought with them inherent energy-efficiency advantages. The first of these was a reduction in fan power requirement to move the air. Close coupling to the heat load allows for a much shorter air delivery and heat removal path. This represents a shift in the mindset of data center air distribution from cold air supply to heat removal. Removal of heat from the hot aisle before it has a chance to mix with surrounding air in the room makes the remaining areas in the room a large volume of supply air. With this in mind, the length scale for air delivery in row-based systems is only a few feet (varies with number of racks and air-conditioning units).

In most CAHU and CRAH implementations it is necessary to maintain a fixed fan speed to deliver the necessary pressure for uniform airflow through delivery vents. In close-coupled designs, such as row-based, the static pressure requirement is significantly less, with only the cooling unit resistance to overcome. Without the requirement for constant pressure, row-based designs allow for variable air volume to scale back fan speed with heat load demand. This feature boosts the energy efficiency through part-load operation with increasing gains at lower loads as shown in *Figure 4*.

Eliminating mixing of hot and cold airstreams produces another energy benefit resulting from much warmer return air temperatures to the cooling units. Some advantages to warmer air return temperatures are:

- An increase in cooling capacity per unit that reduces the overall cooling footprint. The warmer return air temperatures provide a higher temperature differential to the cooling coil over rooftop and perimeter systems, and, therefore, more heat removal.
- More effective capture of hot air enables a much warmer supply temperature (no need to overcool the air to compensate for mixing).
- Limited or no condensate removal, reducing makeup humidification requirement.

Several row-based configurations are available in the market, which use varying placement of the cooling unit in the row and different methods of heat rejection. While these approaches to row-based cooling can be compared for the best energy efficiency, the real energy gain is with the row-based architecture over distributed air delivery systems like perimeter CRAH and CAHU systems. The following comparison of these different architectures illustrates the energy-efficiency advantage of the row-based architecture.

Data Center Cooling Architecture Efficiency Comparison

Let's compare three cooling architectures for the cooling of a mission critical information technology space. The key metric for this comparison shall be power consumed by the cooling infrastructure versus power dissipated by information technology equipment. This comparison attempts to understand and account for all power consumed across the entire length scale of thermal transport (IT rack exhaust to outdoor ambient).

$$\text{Efficiency}_{\text{Metric}} = \frac{\text{Cooling Power}\left[\text{kWh}\right]}{\text{IT Power}\left[\text{kWh}\right]} \tag{1}$$

The general format of the metric equation, from above, yields the ratio of cooling power to IT power. Proper understanding of this metric reveals that the lower the value, the more energy efficient the cooling architecture.

Symbols and Constants Used

Cp_{Air} = Specific Heat Air, 1.022 kJ/kg·°C

Cp_{Water} = Specific Heat Water, 4.188 kJ/kg·°C

ρ_{Air} = Density Air, 1.173 kg/m^3

ρ_{Water} = Density Water, 999.7 kg/m^3

η = Fan or Pump Efficiency

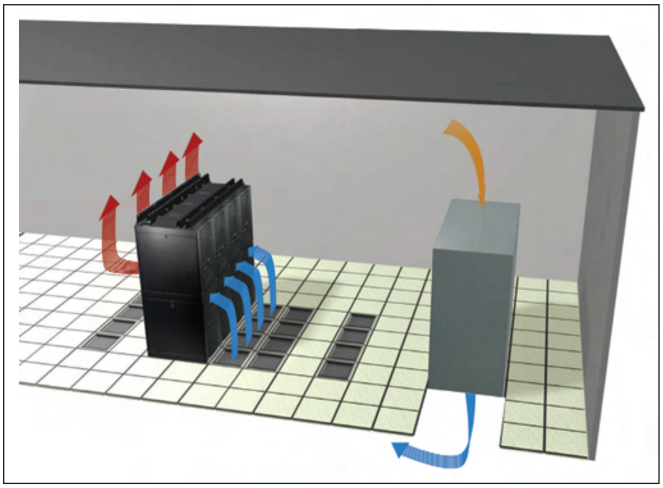
η_{Pump} = Pump Efficiency, 0.65

$\dot{V}$ = Volumetric Flow Rate (m^3)

H = Head Loss (m)

g = Gravitational Acceleration 9.81 m/^{2}

$\dot{Q}_{NetSensible}$ = Net Air Handler Cooling Power (kJ·sec)



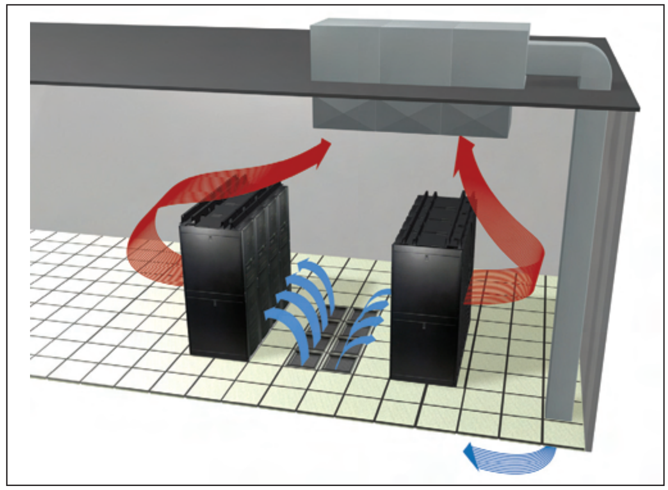


Figure 1 (left): Perimeter computer room air handlers (CRAH). Figure 2 (right): Rooftop units.

$\dot{Q}_{Demand}$ = Architecture Specific Total Chiller Load (kWh)
λ_J = Power Ratio Chiller, Consumed/Load (kWh/kWh)
β_J = Bin Data Condenser Water Temperature (hr)

Equations Used

$\text{Power}_{\text{Shaft}} = \left(\dot{V}_{Air} \cdot \rho \cdot g \cdot H\right) / \left(\eta \cdot 1000\right)[\text{kW}]$ applies to fans or pumps (2)

$$\text{LAT} = \text{EAT} - \dot{Q}_{NetSensible} / \left(\dot{V}_{Air} \cdot \rho \cdot Cp_{Air}\right) {}^{\circ}\text{C}\left({}^{\circ}\text{F}\right) \quad (3)$$

$$\text{Power}_{\text{Chiller}} = \sum_{J=1}^{8} \left(\dot{Q}_{Demand} \cdot \beta \cdot \lambda_J\right) / 8{,}760 \ [\text{kW}] \quad (4)$$

General Considerations for Comparison

The architectures considered here include CAHUs, perimeter floor mounted CRAHs, and in-row air handlers (IRAHs). This study focuses only on sensible cooling requirements for the IT equipment and excludes considerations regarding space humidity control (dehumidification and or humidification).

The reader should be cautioned that architectures using CAHU and CRAH equipment have lower sensible heat ratios than the IRAH, and likely require additional energy consumption to maintain space humidity requirements.

Ultimately, the primary metric driver becomes the characteristic efficiencies of the three air delivery and distribution methods used by the specific architectures.

The theoretical data center used for this evaluation has an actual heat release by IT equipment and lighting set at 0.75 MW. The chilled water cooling source for the IT loads, lighting, and air-handlers is supported by a vapor compression chiller, using screw compressor technology outfitted with an inverter drive. This chiller supplies a constant 45°F (7°C) chilled water supply for all three architectures considered. The heat of rejection of said chiller is removed by cooling tower water.

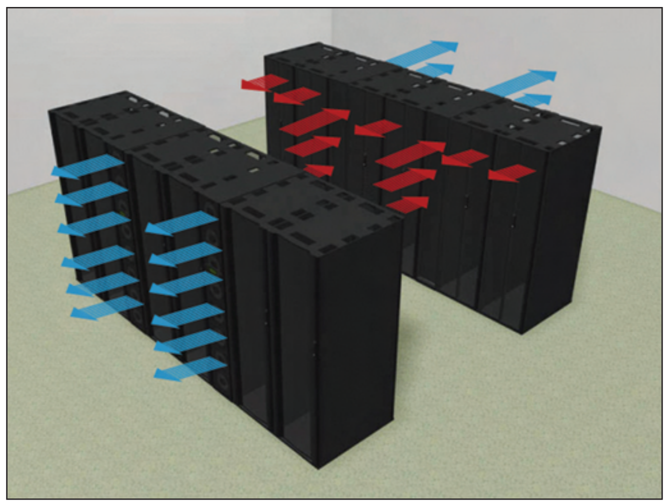


Figure 3: Row-based cooling. Row-based cooling architecture, as depicted, uses a free air discharge without ducting or any containment of hot or cold airstreams.

The water temperature from the cooling tower can track ambient environmental wet-bulb temperature down to a minimum tower leaving fluid temperature of 55°F (13°C). The leaving tower temperature for given wet-bulb bin temperatures is determined from cooling tower performance curve for 100% design flow with 10°F (–12°C) temperature range line.[1] The combination of inverter compressor drive and lower condenser water temperatures allows for significant chiller efficiency gains during periods of low chiller lift.

For the purpose of this comparison, the bin wet-bulb hours for St. Louis shall be considered. The resulting condenser water leaving fluid temperature bin hours are depicted in *Table 1*.

While many condenser water systems may vary the condenser water flow as a function of chiller load, this study shall maintain full design condenser water flow to further enhance performance/efficiency of the chiller. The duty cycle of the

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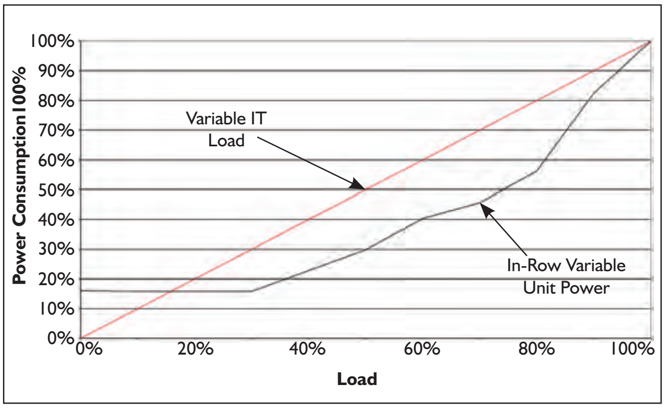

Figure 4: Variable speed fan electrical consumption. Note: Energy savings from variable airflow may not be recognized by all row-based cooling systems. This feature is specific to systems incorporating a variable speed control algorithm.

°F	J	Hours β_J
85.1 – 90.0	1	194
80.1 – 85.0	2	916
75.1 – 80.0	3	1,353
70.1 – 75.0	4	894
65.1 – 70.0	5	1,041
60.1 – 65.0	6	932
55.1 – 60.0	7	1,234
55.0	8	2,196
Total		8,760

Table 1: Condenser water bin hours.

cooling tower fans shall be adjusted accordingly to the demand factor placed upon the chiller by the various cooling architectures considered.

Table 2 establishes the performance of the central chiller in terms of kWh consumed versus kWh load. The particular chiller selected has exceptionally high part load and low lift efficiencies. This was a deliberate choice to avoid possible exaggeration of downstream efficiency gains between the various cooling architectures. Less efficient selections downstream of the air handler (chiller, pump, and cooling tower) will magnify overall power consumed by increased fan loads. Chiller power ratios (λ) with condenser water temperature greater than 85°F (29°C) have been extrapolated. Errors introduced by this method are minimal, as operating hours beyond this temperature account for only 3% of total hours.

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The chilled water circulating loop shall have a base loss of 40 ft (12 m) of head allowed for facility piping and chiller, and shall be summed with the air-handler losses for the specific cooling architectures. The chilled water flow rates shall be set at the value required for the specific cooling architecture.

In-Row Air-Handler (IRAH)

As previously mentioned, an alternate method and emerging cooling architecture for IT loads is to intersperse air-handling units within rows containing racks housing the IT loads. These air handlers are designed for this application with special control algorithms to maximize the stability of the thermal environment. Typically, these air handlers are small, allowing nearly ideal capacity resolution versus IT loads. Additionally, placement within the IT

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°F	J	λ_J
85.1 – 90.0	1	0.18
80.1 – 85.0	2	0.16
75.1 – 80.0	3	0.14
70.1 – 75.0	4	0.12
65.1 – 70.0	5	0.11
60.1 – 65.0	6	0.09
55.1 – 60.0	7	0.08
55	8	0.07

Table 2: Chiller performance versus water temperature.

Cooling Component	IRAH	CRAH	CAHU	Units
AHU Fan Power	30.6	88.0	83.2	kW
Chilled Water Pump Power	10.2	11.0	11.1	kW
Mean Chiller Power (from *Equation 4*)	83.9	94.7	94.2	kW
Condenser Pump Power	18.5	18.5	18.5	kW
Cooling Tower Power	16.2	18.3	18.2	kW
Mean Total Cooling Power	159.30	230.50	225.10	kW
$Efficiency_{Metric}$ (from *Equation 1*)	0.21	0.31	0.30	
Annual Cooling Operating Cost	139,572	201,878	197,211	$ USD

Table 3: Cooling infrastructure power consumption.

rows minimizes the mixing of air and allows a much greater percentage of air delivered from the air handler to have first-pass opportunity through the IT loads.

IRAH Unit Specifications[2]

- 2,900 cfm (1369 L/s) at free discharge
- Sensible Cooling: 25.2 kWh at 95°F DB and 67.7°F WB (35°C DB and 20°C WB)

- Sensible Heat Ratio: 1.0
- Chilled Water Flow: 17.9 gpm at 25 ft of head (1.13 L/s) at 8 m of head)
- Leaving Air Temperature: 67.4°F (19.7°C) (from Equation 3)
- Fan Power: 1 kWh (published manufacturer data, all losses included)
- CW Pump Power: 0.34 kW (from Equation 2)

The theoretical data center load of 0.75 MW would require 30 IRAH units, adding an additional 30.6 kW fan power plus 10.2 kW chilled water pump power above and beyond the 750 kW IT and lighting load. This scenario yields a total chiller load of 791 kW.

Computer Room Air Handler (CRAH)

Presently, common practice for reducing first-time capital expense of perimeter cooling solutions leverages the largest practical and commercially available cooling equipment. An unfortunate consequence of using large capacity boxes is a reduction in capacity resolution.

An additional consideration for CRAH units is the fan placement within the air-distribution path. These fans are placed in the bottom of the CRAH unit with little or no outlet transition into the raised floor plenum. The consequence is a fan outlet system effect[3] that adds a significant static effect on fans. This effect is a function of blast area, outlet area, velocity, and transition length. The outlet system effect is often overlooked and frequently may result in products as installed delivering less than anticipated airflow quantities.

The theoretical CRAH depicted below has a net sensible cooling capacity of 102 kWh versus the 750 kWh for the combined IT and lighting load. In this case, full capacity without redundancy would require 7.3 CRAH units per the below specification. The IT load being considered requires eight CRAHs with an immediate over-provisioning factor of 1.09 times the base load.

CRAH Unit Specifications

- 17,100 cfm at 0.3 in. w.c. (8070 L/s at 75 Pa) floor pressure
- Sensible Cooling: 113 kWh at 75°F (24°C) DB, 45% RH, 61°F (16°C) WB
- Sensible Heat Ratio: 0.95
- Chilled Water Flow: 81 gpm at 18 ft of head (5 L/s at 6 m of head)
- Fans (3x) $Power_{Shaft}$: 3.2 kWh each (from Equation 2)
- Forward Curve 15 in. × 15 in. (38 cm × 38 cm) double inlet, double-width (DIDW)
  – Blast Area = 0.81 ft × 1.55 ft = 1.26 ft^2 (0.25 m × 0.47 m = 0.12 m^2)
  – Outlet Area = 1.32 ft × 1.55 ft = 2.05 ft^2 (0.40 m × 0.47 m = 0.19 m^2)
  – BA/OA = 1.26 ft^2/2.05 ft^2 = 0.61 (0.38 m^2/0.62 m^2 = 0.61)
  – Static Efficiency: 0.59
  – Outlet Velocity: 2,800 fpm (14 m/s)
  – Outlet System Effect: 0.6 in. w.c. (149 Pa)
  – Floor Pressure: 0.3 in. w.c. (75 Pa)
  – Filter Loss: 0.75 in. w.c. (187 Pa)
  – Coil Loss: 0.65 in. w.c. (162 Pa) (wet)
  – Cabinet Loss: 0.5 in. w.c. (125 Pa)
- Motor Power: 11.0 kW (0.92 motor efficiency × 1.05 drive loss)
- Net Sensible Cooling: 102 kW
- Leaving Air Temp: 56°F (13°C) (from Equation 3)
- Chilled Water Pump Power: 1.4 kW (from Equation 2)

The above eight CRAH units combined would add an additional 88 kW fan power plus 11.2 kW chilled water pump power above and beyond the 750 kW IT and lighting load. This scenario yields a total chiller load of 893 kW at a sensible heat ratio of 0.95.

CAHU

Most applications using central air handlers will have custom air handler units designed and built for the specific project. The wide variation of design practices and component selection make it difficult to express performance data in absolute terms. The values used herein are for purpose of comparison and are believed to reasonably represent nominal values. However, some variation should be anticipated.

The reader may notice that a significant contribution to the CRAH fan losses from above, outlet system effect, is missing in the below CAHU example. This is possible due to physical geometry of custom air handlers allowing better practice regarding fan placement and operation. Unfortunately, in many cases the gains from reducing and or eliminating blower outlet system effects are frequently offset by increased pressure losses in delivery system: ducting, elbows, and diffusers.

The theoretical data center load of 0.75 MW will require a quantity of four CAHUs per the below specification without any redundancy. With the CAHU being custom built equipment the amount of over provisioning can be carefully controlled allowing only for the desired factor of safety.

CAHU Specifications

- 34,000 cfm at 1.0 in. w.c. (16 046 L/s at 249 Pa) external static pressure
- Sensible Cooling: 220 kWh at 75°F (24°C) DB, 45% RH, 61°F (16°C) WB
- Sensible Heat Ratio: 0.95
- Chilled Water Flow: 158 gpm at 20 ft of head (10 L/s at 6 m of head)
- Fans (2x) $Power_{Shaft}$: 9.1 kW each (from Equation 2)
  – 76 cm backward-inclined, double-width (BIDW)
  – Outlet Velocity: 1,825 fpm (9 m/s)
  – Static Efficiency: 0.66
  – Return Air Duct: 0.3 in. w.c. (75 Pa)
  – Filter Loss: 0.75 in. w.c. (187 Pa)
  – Coil Loss: 0.60 in. w.c. (149 Pa) (wet)

– Cabinet Loss: 0.35 in. w.c. (87 Pa)
– Supply Duct Loss: 1 in. w.c. (249 Pa)
- Motor Power: 20.8 kW (0.92 motor efficiency × 1.05 drive loss)
- Net Sensible Cooling: 56.8 tons (200 kW)
- Leaving Air Temperature: 56.3°F (14°C) (from Equation 3)
- Chilled Water Pump Power: 2.8 kW (from Equation 2)

The above four CAHUs combined would add an additional 83.2 kW fan power plus 11.2 kW chilled water pump power above and beyond the 750 kW IT and lighting load. This scenario yields a total chiller load of 888 kW at a sensible heat ratio of 0.95.

Conclusion

The annual electrical cost of three cooling architectures: IRAH, CRAH, and CAHU are given respectively $139,572, $201,878, and $197,211 (at $0.10/kWh). Of course, the magnitude of savings would vary due to chiller plant efficiency, utility cost, and base IT and lighting loads. The row-based cooling architecture versus the other two choices affords a two-thirds reduction in fan power consumed by cooling equipment, with additional savings compounded throughout the entire downstream cooling infrastructure.

Although row-based cooling has a sizable advantage in operational cost savings, it will not be a silver bullet for all applications. Certainly, there are far more existing data centers than new ones being built, and it is likely that a mix of cooling architectures, including all of the above mentioned, will be deployed within the same data center. As data centers evolve and densities increase, a hybrid approach to cooling various density heat loads is the likely result. However, new data center space (whether expansion or entirely new) should always consider row-based cooling for the best energy efficiency and predictability whenever possible.

Acknowledgments

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