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# Dedicated Outdoor Air System for Commercial Kitchen Ventilation

By **Stephen L. Brown**, Member ASHRAE

This case study presents data collected as a follow-up to the June 2003 ASHRAE Journal article, “Keeping Cooks in the Kitchen—Solving the Makeup Air Dilemma,”<sup>1</sup> which described the design process for a commercial kitchen ventilation (CKV) system that would be installed in a casual family dining restaurant. The restaurant owners had challenged their engineer to design a CKV system that would solve their kitchen comfort problems. The system described in the article was the engineer’s response to the customer’s challenge.

The customer’s previous CKV system included a kitchen hood system with heated makeup air (MUA) and a separate kitchen HVAC system. The hood system included canopy-style hoods with back-wall<sup>2</sup> MUA plenums. The kitchen HVAC system included one packaged roof top

unit (RTU) designed for kitchen heating/cooling and introduction of a small quantity of outdoor air (OA). The RTU operated in continuous fan mode with active return air. To replace the previous system, the engineer designed the system described in the article, including a pack-

aged dedicated outdoor air (DOA) unit, which included direct gas-fired heating and DX cooling. The OA supplied by the DOA unit is used to replace the air exhausted from the kitchen through the hood and to heat or cool OA as required to maintain space comfort.

The revamped CKV system included a new hood design consisting of exhaust-only, listed backshelf-style hoods<sup>3</sup> along with listed canopy-style hoods. Appliances were relocated to optimize the use of the backshelf hoods. The hood redesign lowered total exhaust airflow, which was consistent with the findings of a design guide by Southern California Edison.<sup>4</sup>

The DOA system installation met the engineer’s performance criteria. This customer has continued to work with the

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#### About the Author

**Stephen L. Brown** is president of LCSsystems in Dublin, Ohio. He is chair of ASHRAE TC 5.10, Kitchen Ventilation.

This customer felt its current hood system performed well, but had been receiving complaints about moisture in its restaurants, as well as the costs associated with repairing structural and image damage caused by that moisture. The customer decided to consider the new DOA design as one component of its moisture management program.

engineer and to include the DOA design in all new restaurants built since 2003.

Utility bills from the customer's restaurant with the MUA system were compared to those from its restaurant with the DOA system and indicated that the DOA facility had noticeably lower gas consumption. The utility bills were not adequate to identify the source of the savings. The customer requested energy monitoring to provide more definitive answers. It was determined, however, that too many components had been changed in the design process of the two systems (hoods, appliance location, some appliances, etc.) to be able to quantify the savings. It was decided a more similar base building design would be required to quantify and compare the two CKV designs.

The opportunity to compare similar designs arose in late 2003 when a different multiunit casual dining customer was contacted about the DOA design. This customer had already optimized its current hood system, which included listed backshelf and canopy-style hoods that had significantly reduced exhaust airflow rates required by the previous all-canopy hood system. Its current CKV system included a direct, gas-fired, heat-only MUA unit that was ducted to perimeter-style<sup>2</sup> MUA plenums, which were mounted at the ceiling line.

This customer felt its current hood system performed well, but had been receiving complaints about moisture in its restaurants, as well as the costs associated with repairing structural and image damage caused by that moisture. The customer decided to consider the new DOA design as one component of its moisture management program. The system design work began in 2003.

### Case Study System Comparison

Two building sites were compared. For the purpose of this article, the systems will be designated as the control site and the test site. The control site includes the owners' existing CKV design with MUA system and separate kitchen heating and cooling RTU. The test site features the new CKV design, which includes one DOA unit providing replacement air for the kitchen hood and control of kitchen comfort. The two sites have

identical building footprints, seating capacities and seating arrangements, window placement and quantity, as well as overall construction packages (insulation and other building materials). In addition, all exhaust hoods, design exhaust airflow rates and cooking appliances are identical. The dining rooms and kitchen, at both locations, are separated by full-height walls, which include swinging doors and two relatively small (10 ft<sup>2</sup> [0.9 m<sup>2</sup>] each) pass-through openings. System descriptions follow.

The control site system diagram is shown in *Figure 1*. This represents the hood and HVAC systems the owner had used for the previous five years. The kitchen RTU is designed to provide space comfort (heating and cooling) and introduce a small quantity of conditioned OA. The MUA unit is designed to provide OA to replace the air exhausted through the hood. Makeup air is heated to prevent overcooling of food or staff but is not cooled. The restaurant is in a suburban location.

The control site kitchen RTU is a conventional, packaged rooftop unit with indirect gas-fired drum and tube heat and two-stage DX cooling. No hot gas reheat (HGRH) is used. Nominal tonnage is 15 (53 kW) with a 10 ton (35.2 kW) first-stage compressor. The RTU calls for heating or cooling are made by a conventional wall-mounted, non-adjustable temperature sensor located in the kitchen prep area.

The kitchen heating setpoint is 68°F (20°C), and the cooling setpoint is 72°F (22°C). The room sensor cannot be adjusted by the staff. The kitchen MUA unit is a heat-only, direct gas-fired unit with the calls for heat provided by a thermostat that monitors OA temperatures. The MUA call for heat is set at 55°F (13°C). No lockout exists to prevent simultaneous heating/cooling because previous attempts with lockout determined that the lockout would permit introduction of MUA that was cold enough to cool food, or negatively impact hood performance or staff comfort. Prior to this project, the owner had completed considerable testing at other sites, which determined that the MUA introduced had to be at least 55°F (13°C) to avoid cooling food, or negatively impacting hood performance or staff comfort.

The test site system diagram is shown in *Figure 2*. This represents the system, which includes the DOA unit and indicates

the elimination of the MUA unit, MUA duct system and hood MUA plenums that are included at the control site. Not shown, but also eliminated, were gas piping, electrical, and curb and roofing for the MUA unit. The functions of the conventional kitchen RTU and MUA units are incorporated into the single DOA unit. The restaurant is located on the outskirts of a rural community.

The test site DOA unit is a packaged rooftop design with direct gas-fired heat and two-stage cooling. Similar to the control site kitchen RTU, there is no HGRH, although the first-stage compressor includes mechanical capacity control. Nominal tonnage is 17.5 (61.5 kW) with a 10 ton (35 kW) first stage compressor. Calls for dehumidification, heating, cooling or ventilation modes are made by the DOA unit's OA sensors. Sensible heat (SH) and relative humidity (RH) OA sensors are located in the inlet hood of the unit. The primary control function is dehumidification, with call for stage one or stage two compressors, based on individually adjustable enthalpy setpoints. The system controller calculates sensible enthalpy. Call for ventilation is based on the adjustable enthalpy range. In the ventilation mode, no dehumidification, cooling or heating occurs. Calls for heating or cooling are based on OA temperature. OA setpoints are 55°F (13°C) heating and 78°F (26°C) cooling.

A non-adjustable temperature sensor is located in the kitchen prep area. This sensor acts as an override to prevent space overshoot of heating or cooling setpoints. Simultaneous heating and cooling is eliminated by the DOA unit's control system. The space heating setpoint is 68°F (20°C) and the cooling setpoint is 72°F (22°C).

The control site dining room heating and cooling system does not include HGRH. As a component of the owner's moisture management program, HGRH was added to the dining room units at the test site. Dining room tonnage and heating capacities were identical at both sites. This article does not attempt to evaluate the impact of dining room HGRH. The control site includes a gas dishwasher booster heater. Conversely, the test site uses an electric booster heater.

The sites are located 37 miles (59.5 km) apart at similar latitude. As the data collection process proceeded, the data demonstrated that, although the sites were in the same region, temperature and humidity readings were often considerably different. These differences did not alter the collection

methods, but did result in a more complex method of data analysis.

### Case Study Desired Deliverables

At the beginning of this project, the following deliverables were set:

- Justify cost-effectiveness to owner;
- Determine impact of reduced airflows on kitchen comfort;
- Compare ability to control kitchen comfort—temperature and humidity; and
- Compare energy consumption of the two kitchen systems.

**Justify Cost-Effectiveness to Owner.** The owner designed a restaurant using the conventional MUA design and another restaurant with the DOA design for the test site. The bidders were required to bid both, not knowing which design would be built on this site. Dining room HGRH was not included in the original bid. In the DOA design, the equipment costs

for the CKV and HVAC package increased, but installation costs decreased. The net result was cost-neutral. Based on the bidding results, the owner elected to proceed with construction of the DOA design at the test site.

### Determine Impact of Reduced Airflows on Kitchen Comfort.

*Table 1* compares dining room and kitchen design airflows and air changes per hour (ACH) at the two sites. It was important to verify that the drastic reduction in total kitchen airflow at the test site would be adequate to maintain kitchen comfort. One feature of the DOA design is the integration of the kitchen hood into the kitchen HVAC system. In the DOA design, the heat gain is ventilated from the space through the kitchen hood. The control site uses recirculation and the kitchen RTU to control the space gain. The additional kitchen airflow, shown

in *Table 1*, for the control site is the air that is recirculated by the kitchen RTU for space heating and cooling. The DOA design eliminates recirculated air, which in this study reduced the total kitchen airflow by 46%. This airflow reduction would obviously reduce fan motor energy, but would it provide the needed capacity to control space comfort?

**Compare Ability to Control Kitchen Comfort—Temperature and Humidity.** The customer was satisfied with the performance of its current MUA design. The focus at this time was moisture control. The data to be collected was designed to consider total comfort and would include temperature and humidity readings.

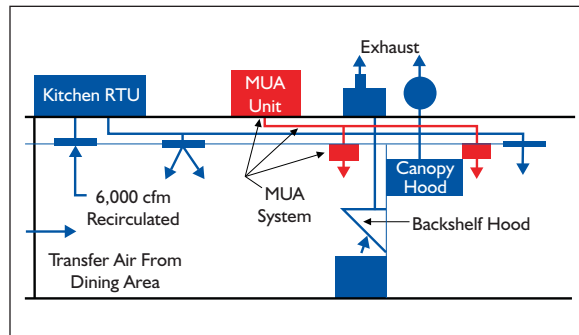


Figure 1: Control site kitchen HVAC.

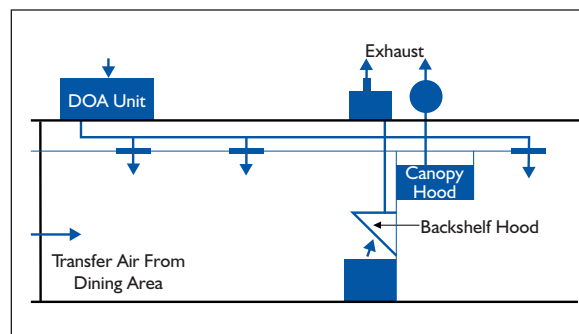


Figure 2: Test site kitchen HVAC.

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**Compare Energy Consumption of the Two Kitchen Systems.** An initial assumption was that the single-unit DOA design would reduce gas consumption. To confirm gas consumption, separate gas meters were installed to submeter dining room and kitchen HVAC gas consumption. Two gas meters were installed at each site: one to record gas consumption of the two dining room RTUs, and the second to record the combined gas consumption of the MUA unit and the kitchen RTU at the control site, while the test site meter recorded the DOA unit gas consumption.

Initial power performance was not a clear assumption. To have a total picture of HVAC power, current transformers (CT) were placed on each phase conductor of all rooftop heating and cooling equipment. Additional CTs were placed on one of the phase conductors of each HVAC exhaust fan motor. This configuration provided a complete picture of HVAC power consumption.

Separate air balance checks were conducted at each site to ensure that airflows were correct and properly recorded. *Table 2* shows the final balance numbers. Total building exhaust includes restroom, kitchen hoods and dish hood exhaust airflows. The intent was to balance transfer airflows as closely as possible. Matching transfer airflows isolated the kitchen and dining areas so that the MUA unit and RTU on one gas meter could be compared directly to the single DOA unit gas meter. This data was used to report and compare kitchen HVAC gas consumption. It was decided this was the most acceptable method since the scope of this case study was not sufficient to monitor all of the variables (customer counts, table turns, door openings, etc.) that could impact dining room heating or cooling performance. Isolating the dining and kitchen energy was important in cooling months, as HGRH was added to the test site dining room units, but not to the control site dining room units.

### Data Collection

An independent third party (P3) automatically collected survey data from each site at the end of every day via the site's dedicated fax lines. Dual sensors that recorded temperature and humidity were located in the dining rooms, kitchens and rooftop units. At each site there were: (2) dining room sensors, (4) kitchen sensors and (1) outdoor air sensor. Outdoor air sensors were located in the DOA unit at the test site and in the MUA unit at the control site. In addition, each night the P3 nightly downloaded the CT pulses, dining room and kitchen gas meter pulses, and provided a separate power analysis of each main conductor feeding the building. The P3 participated in installation, calibration and service of its system and the collection modules. The collected data is stored on the P3 computer system. Online access to the P3 computer data is accomplished through a secure routine. The base data can be accessed and downloaded but cannot be modified.

After considerable data collection, it was determined that the P3 humidity sensors had an unacceptable saturation level. It was later learned that the collection system does include a remote reset capability, but additional checks with handheld calibrated

Zone	Area (sq. ft.)	Ceiling (Height ft.)	Volume (cu. ft.)	Airflow (cfm)	AC (per hour)
<b>CONTROL SITE</b>					
Dining Room	3,400	10	34,000	8,000	14
Kitchen	2,000	9	18,000	11,205	<b>37</b>
<b>TEST SITE</b>					
Dining Room	3,400	10	34,000	8,000	14
Kitchen	2,000	9	18,000	5,852	<b>20</b>

*Table 1: Total kitchen airflow—air changes per hour. The control site kitchen RTU handles 6,000 cfm, of which 5,700 cfm is recirculated and 300 cfm is OA.*

Airflow	CONTROL SITE	TEST SITE
Dining Room OA	1,945	1,975
Kitchen OA	4,584	4,701
Total Bldg. Exhaust	-6,689	-6,702
Building Balance	-120	+64
Transfer Air	1,105	1,061

*Table 2: Final air balance—transfer air set within 4%. Total building exhaust includes restrooms and dishwasher hood.*

equipment showed that the P3 humidity sensors were less accurate at higher humidity levels. Therefore, in early summer 2006, the author added six upgraded data loggers to each site. The additional loggers required manual on-site downloading every two to four weeks. However, when checked against calibrated handheld instruments, the humidity readings from the loggers proved to be more accurate than the P3 sensors. Prior to addition of the loggers, it was discovered that each store did not use the same routine for alternating between right-side-only, left-side-only and full grill line operations. Therefore, two of the additional loggers were mounted at each site to record at grill line left and right locations. The grill logger readings were averaged to eliminate the variable of cooking or no cooking under the single grill line P3 sensor. A logger also was added on the blower side of the evaporator to provide coil versus OA performance data.

This report uses the online data for electricity and gas consumption. This report also uses the online data for all temperatures recorded in heating and ventilation modes. The data logger readings were used in dehumidification and cooling modes for calculation of enthalpy, as well as for humidity reporting. The P3 sensor temperature readings were compared to the logger temperature readings in all modes to further verify accuracy.

The initial heating data comparison looked only at the same day and hours of the day. The initial data indicated that, within an OA temperature range of 35°F to 60°F (1.7°C–16°C), the test site consumed more gas than the control site. However, closer examination of the data indicated a non-system controller was turning the DOA unit on at night when space conditions indicated no heating was required. This was repaired. Further investigation found that the OA temperatures at the two sites

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varied by as much as 12°F (6.7°C). The rural test site location was consistently colder than the suburban control site. Both heating systems discontinued all or some heating when the OA temperature reached 55°F (13°C). It was discovered that OA temperatures above 55°F (13°C) at the control site occurred at the same time the test site was recording OA temperatures in the upper 40s. Therefore, initial gas readings were comparing a location with heating off to a location with heating on; similar temperature disparities were found during colder heating months. During summer months, particularly when dehumidification was required, the wet-bulb temperature at the test site was consistently higher than at the control site. It was surprising to see such discrepancies between two sites that, at first glance, appeared to be located in such similar weather zones.

To resolve these discrepancies, the data was sorted into temperature bins, and gas consumption and power were charted for each bin. For dehumidification and cooling, enthalpy was calculated and similar enthalpy setpoints were compared to power consumption at those conditions. This greatly increased the time required to analyze the data, but provided a much clearer comparison of the two operations.

## Results

### Summer Cooling

Figure 3 charts control and test site OA enthalpy points that are above the ASHRAE summer cooling 89°F (32°C) DB and 73°F (23°C) MWB 1% design used for the two sites. Figure 3 charts the 89°F/73°F (32°C/23°C) design which equals an enthalpy of 36.58 Btu/lb (85.09 kJ/kg) and compares that data point to the control site OA at 92°F/76°F (33°C/24°C) = 39.41 Btu/lb (91.67 kJ/kg) and kitchen at 86°F/73°F (30°C/23°C) = 36.61 Btu/lb (85.15 kJ/kg) to the test site OA at 84°F /76°F (29°C/24°C) = 39.49 Btu/lb (91.85 kJ/kg) and kitchen at 76°F /64°F (24°C/18°C) = 29.22 Btu/lb (67.97 kJ/kg). This data presents a reduction in kitchen enthalpy of 10.27 Btu/lb (23.89 kJ/kg) at the test site, versus 2.8 Btu/lb (6.51 kJ/kg) at the control site. This data demonstrates that at virtually equal OA enthalpy, the test site kitchen remained within the summer comfort zone, while the control site kitchen was well outside the zone. These conditions were chosen because they were above the design conditions for these sites and provided a severe test of the ability of the DOA design to maintain kitchen comfort at reduced total kitchen airflow. The results were similar when comparing larger blocks of data.

These results also are similar to those presented in an ASHRAE paper,<sup>5</sup> which modeled the impact of supply air systems on kitchen temperature. The paper stated that, “CFD study of perimeter perforated supply (PPS) plenums demonstrated that performance depends on supply air temperature. Under all tested conditions, most of the air supplied through the PPS ended up in the kitchen space before being exhausted through the hood. Kitchen space air temperature will depend on the temperature of the air supplied through the hood.” Figure 4

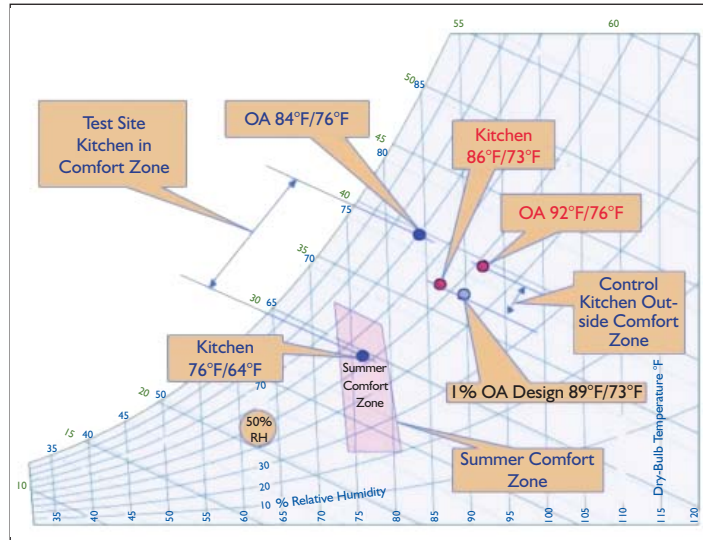


Figure 3: Enthalpy comparison.

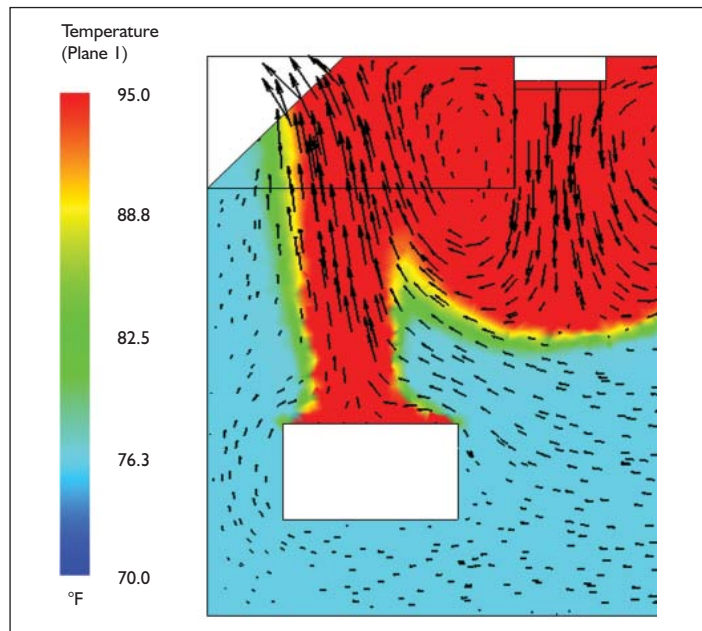


Figure 4: Example of CFD simulation.<sup>5</sup>

gives an example of the results from the paper. This dedicated outdoor air case study did not attempt to quantify the percentage of MUA that entered the general kitchen area at the control site before entering the hoods.

Tables 3 and 4 compare averages of temperature data recorded between July 31, 2006 and Aug. 6, 2006. The results indicate that, at nearly identical OA enthalpy, the average test site kitchen temperature was 8°F (14.4°C) cooler than the control kitchen. Chart (AVERAGE) = average of readings taken within each bin. The average test site kitchen enthalpy was 15% lower than the control kitchen enthalpy.

Tables 3 and 4 also chart the power consumption (kW) for the period. Although the test kitchen was more comfortable in all bins, the DOA unit power consumption was 15% to 26%

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lower than the test site combined RTU and MUA unit power consumption. The lower DOA consumption is due, in part, to the unit's refrigeration capacity controller, which monitors compressor suction pressure and temperatures and reduces compressor capacity to match the controller's adjustable setting. Compressor power consumption is reduced as capacity is reduced. The reduction is not linear. Test site inspections found some level of capacity reduction taking place even with OA at 94°F (34°C).

### Part-Load Cooling

Tables 5 and 6 provide examples of kitchen comfort that occurred in part-load OA conditions that required significant dehumidification. As the data indicates, the test site was capable of controlling kitchen relative humidity below 57%, without hot gas reheat, even when outdoor air relative humidity reached 98.5%. The test site kitchen relative humidity was consistently lower than the control site. The test site kitchen was more comfortable in all bin categories and the power consumption was also lower in all categories but one. In that category the OA enthalpy is higher than the unit controller's minimum setting for dehumidification, so the compressor is operating and the refrigeration capacity controller is reacting to the latent load. In the next lower bin, the OA enthalpy is below the minimum control setting, which means the unit is no longer operating in a dehumidification mode and the compressor is off.

### Winter Heating

The December 2006 data reported in this article is consistent with data collected during the winters of 2004 and 2005. Kitchen temperatures were consistent at both sites. Table 7 provides a comparison of gas usage for 280 operating hours in December 2006. The test site DOA unit gas consumption was lower than the control site MUA plus RTU gas consumption at all OA temperatures. The test site operated 34% of total hours, during hours that were at or below 35°F (1.7°C) compared to 17% at the control site. Neither the impact of hours at colder OA temperatures, nor the impact that dining room activity had on dining room gas consumption were within the scope of this study. As the chart indicates, the DOA unit gas savings increased at lower OA temperatures.

### Conclusions

**Justify the Cost-Effectiveness to Owner.** The owner's bidding process determined the DOA design to be a good first-cost option. The bid for the building with the DOA design was \$500 lower than the owner's standard MUA system bid for this site. Not including dining room HGRH, the HVAC equipment costs for the DOA design were higher. MUA equipment and installation costs were eliminated and these savings offset the increased DOA equipment costs. The cost comparison was supported by subsequent projects.

### Determine Impact of Reduced Airflows on Kitchen

BIN temps. (F)	(HRS)	Kitchen HVAC Power		OA (average)				Kitchen (average)			
		(kW)	(KWh)	DB (F)	WB (F)	H <sup>1</sup> (%)	RH (%)	DB (F)	WB (F)	H <sup>1</sup> (%)	RH (%)
90-95	19.3	408	21	92.5	75.5	38.9	46.2	88.7	72.1	35.9	46.5
85-90	23.3	467	20	87.4	71.8	35.7	47.9	83.6	68.2	32.6	47.2
80-85	11.3	216	19	78.9	69.5	33.7	63.9	80.9	66.9	31.5	51.8
75-80	13.5	252	19	77.8	70.2	34.2	69.4	78.7	66.8	31.4	55.0
70-75	3.5	66	19	73.1	66.4	31.1	70.9	78.8	65.4	30.2	50.3
65-70	0.8	14	19	67.6	62.3	28.0	74.8	75.9	64.1	29.3	52.9
<b>Average</b>				<b>79.6</b>	<b>69.3</b>	<b>33.6</b>	<b>62.2</b>	<b>81.1</b>	<b>67.3</b>	<b>31.8</b>	<b>50.6</b>

Table 3: Summer cooling control site—July 31 to Aug. 6, 2006. (Average) = average of readings taken in each bin. H<sup>1</sup> = Enthalpy (Btu/lb).

BIN temps. (F)	(HRS)	Kitchen HVAC Power		OA (average)				Kitchen (average)			
		(kW)	(KWh)	DB (F)	WB (F)	H <sup>1</sup> (%)	RH (%)	DB (F)	WB (F)	H <sup>1</sup> (%)	RH (%)
90-95	11.0	190	17	91.6	74.8	38.3	46.5	77.0	63.8	29.1	49.5
85-90	23.0	391	17	87.2	72.1	35.9	49.1	75.1	61.9	27.8	48.3
80-85	21.0	343	16	82.4	70.9	34.8	57.6	74.6	61.1	27.2	47.0
75-80	15.0	231	15	77.5	70.7	34.6	72.4	72.7	60.4	26.7	49.8
70-75	11.0	168	15	73.1	68.1	32.4	77.7	70.9	59.6	26.1	52.5
65-70	5.5	77	14	68.0	65.3	30.2	86.6	70.0	59.3	25.9	53.9
<b>Average</b>				<b>80.0</b>	<b>70.3</b>	<b>34.4</b>	<b>65.0</b>	<b>73.4</b>	<b>61.0</b>	<b>27.1</b>	<b>50.2</b>

Table 4: Summer cooling test site—July 31 to Aug. 6, 2006. H<sup>1</sup> = Enthalpy (Btu/lb).

BIN temps. (F)	(HRS)	Kitchen HVAC Power		OA (average)				Kitchen (average)			
		(kW)	(KWh)	DB (F)	WB (F)	H <sup>1</sup> (%)	RH (%)	DB (F)	WB (F)	H <sup>1</sup> (%)	RH (%)
80-85	4.5	86	<b>19.1</b>	82.2	66.1	30.8	42.8	73.0	63.8	29.0	60.8
75-80	12.0	212	<b>17.7</b>	77.1	65.2	30.1	53.2	73.0	63.8	29.0	60.8
70-75	23.0	334	<b>14.5</b>	72.4	62.9	28.6	59.4	70.8	62.5	28.1	63.4
65-70	36.8	367	<b>10.0</b>	67.1	63.3	28.8	81.5	69.5	63.6	28.9	72.8
60-65	12.0	84	<b>7.0</b>	62.7	59.4	26.0	82.9	69.5	62.2	27.9	66.9
55-60	1.5	9	<b>5.8</b>	58.8	57.7	24.9	93.6	68.4	62.0	27.7	69.9
<b>Average</b>				<b>70.1</b>	<b>62.5</b>	<b>28.2</b>	<b>68.9</b>	<b>70.7</b>	<b>63.0</b>	<b>28.4</b>	<b>65.8</b>

Table 5: Part-load cooling control site—Sept. 10–16, 2006. H<sup>1</sup> = Enthalpy (Btu/lb).

BIN temps. (F)	(HRS)	Kitchen HVAC Power		OA (average)				Kitchen (average)			
		(kW)	(KWh)	DB (F)	WB (F)	H <sup>1</sup> (%)	RH (%)	DB (F)	WB (F)	H <sup>1</sup> (%)	RH (%)
80-85	2.3	28	<b>12.3</b>	82.2	67.1	31.8	45.8	75.5	61.7	27.6	45.6
75-80	10.0	109	<b>10.8</b>	76.9	64.7	29.7	51.9	73.6	60.7	26.9	47.6
70-75	17.3	182	<b>10.6</b>	77.9	64.3	29.4	47.8	73.8	61.1	27.2	48.3
65-70	30.3	291	<b>9.6</b>	67.0	64.5	29.6	87.6	73.1	61.6	27.5	52.5
60-65	24.0	186	<b>7.7</b>	62.6	60.8	27.0	90.5	70.8	60.1	26.5	53.7
55-60	6.0	14	<b>2.3</b>	57.8	57.5	24.8	98.5	71.2	61.1	27.2	56.6
<b>Average</b>				<b>70.7</b>	<b>63.1</b>	<b>28.7</b>	<b>70.4</b>	<b>73.0</b>	<b>61.1</b>	<b>27.1</b>	<b>50.7</b>

Table 6: Part-load cooling test site—Sept. 10–16, 2006. H<sup>1</sup> = Enthalpy (Btu/lb).

**Comfort.** The study's comfort results demonstrate that the reduced kitchen airflow at the test site was more than ad-

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equate to maintain kitchen and operator comfort. At the lower airflow rates, the test site kitchen was more comfortable in dehumidification and cooling modes, equal in heating modes and was capable of operating in a ventilation mode (no heating and no cooling) in a wider band of OA conditions. It is the author's conclusion that integration of the exhaust hoods into the kitchen HVAC system contributed to the ability to maintain comfort at lower airflows. The test site hood ventilates the space gain that is returned to the kitchen RTU at the control site. The study results indicate ventilation to be more effective than recirculation, especially in dehumidification, cooling and ventilation modes.

**Compare ability to control kitchen comfort—temperature and humidity.** Perhaps the conclusion that is easiest to draw is the negative impact conventional makeup air had on kitchen comfort when outdoor air conditions required dehumidification or cooling. The test site DOA unit removed moisture from the OA before it entered the kitchen. The MUA unit at the control site did not. The test site kitchen relative humidity could be controlled below 60% in both high moisture, part-load conditions, and in conditions that were at or above OA design. This was not the case at the control site, where the MUA system added moisture, and the kitchen temperature and humidity data trend lines tended to parallel OA conditions. Although no complaints were received at the test site, the part-load kitchen temperatures were on the border—or slightly cooler than—the comfort zone. Part-load complaints did originate from the control site regarding cave-like (damp and cool) conditions, kitchen floors that would not dry and condensation on HVAC diffusers. The preferred embodiment removes moisture before it enters the space. Heating was less of an issue, as both systems maintained thermostat settings. However, the staff and management at the control site did comment on feeling the “cold air” around the hoods. Although the MUA they referred to was heated to 55°F (12.8°C), it was sufficiently cooler than the space to illicit the “cold” comments.

**Compare energy consumption of the two kitchen systems.** The conclusion that is easiest for the author to draw is that moving less air saved energy. The energy data supports this conclusion. The comfort data enforces that the energy saved moving less air did not come at the cost of sacrificing comfort, but rather comfort was enhanced. The overall comparison for the months of July and August showed the control site consumed, on average, 37% more power than the test site. This higher average was due, in part, to periods where the test site DOA was capable of maintaining space comfort while operating in the ventilation mode. The ability of the DOA system to identify those OA and space conditions that can maintain kitchen comfort using ventilation only, offers the greatest potential for energy savings. It also is the author's conclusion that integration of the exhaust hood into the kitchen HVAC system contributed to energy savings. The DOA design also took maximum advantage of the kitchen balance point during the heating season. The

BIN temps. (F)	Operating Hours		Cu. Ft. Totals		Cu. Ft./Hr.	
	Control	Test	Control	Test	Control	Test
60-65	9.5	2.3	0	0	0	0
55-60	30.0	7.5	337	0	11	0
50-55	43.0	31.8	1,745	346	41	11
45-50	47.0	41.5	3,181	1,478	68	36
40-45	45.8	55.3	4,517	4,005	99	72
35-40	56.0	46.5	6,867	4,612	123	99
30-35	34.5	55.8	5,522	6,905	160	124
25-30	12.8	29.0	2,121	4,021	166	139
20-25	1.3	7.0	333	1,214	266	173
15-20	0.0	3.3	0	618	0	190
<b>Total</b>	<b>280</b>	<b>280</b>	<b>24,621</b>	<b>23,196</b>		

Average Kitchen Temps Control Site = 73.0

Average Kitchen Temps Test Site = 73.5

**Table 7: Kitchen heating gas usage—December 2006 (partial). Gas consumption per hour in individual temperature BIN.**

DOA unit direct gas-fired heating system was effective at maintaining a constant replacement air discharge temperature that was warm enough to avoid staff feeling cool, yet was cool enough to allow the space heat gain to work with the replacement air and maintain the desired 68°F (20°C) space thermostat temperature.

Finally, robust DOA unit controls are required. The DOA design integrates the kitchen hood and HVAC into a single system. Therefore, a single control system and control strategy, along with complete cooperation between hood and HVAC vendors, are essential for successful implementation. The owner of the test sites has and is proceeding with more DOA design installations.

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This article is dedicated to the memory of Mr. Keith Sweitzer.●