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The Effect of Supply Air Systems on Kitchen Thermal Environment

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ABSTRACT

A commercial kitchen is a complicated environment where multiple components of a ventilation system including hood exhaust, conditioned air supply, and makeup air systems work together but not always in unison. That is why many kitchens are hot. A hot and uncomfortable kitchen contributes to productivity loss, employee turnover, and eventually profit loss for the restaurant operator. Using thermal displacement ventilation in kitchen environment allows for a reduction in space temperature without increasing the air-conditioning system capacity. Application of two systems (traditional mixing ventilation system and thermal displacement ventilation system) is compared in a typical kitchen environment using computational fluid dynamics (CFD) modeling. Often kitchen exhaust hoods are provided with untempered makeup air. It is not uncommon to hear the claim that this makeup air is exhausted through the hood without having any effect on kitchen space temperature. The validity of this claim is analyzed in this paper for two makeup air configurations using a combination of measured data and results from CFD models. Kitchen space temperature increase is calculated as a result of supplying unconditioned makeup air during the summer.

INTRODUCTION

It has been well documented that temperature affects productivity. If the temperature in the space increases by 10°F (5.5°C) above the comfort level, the productivity may drop as much as 30% (Wyon 1996). Take, for example, a well-designed and comfortable kitchen staffed with seven employees. If the temperature in this kitchen increased by 10°F (5.5°C) the manager would have to hire three more people to do the same job.

Besides the productivity losses, high temperature in a kitchen also contributes to a very high turnover rate. In the restaurant industry, on average, four persons per year will be hired and trained for the same job position. It is no wonder that the National Restaurant Association identified the single most critical issue facing the restaurant industry to be hiring and retaining a professional kitchen staff (National Restaurant Association 2001).

A modern commercial kitchen is often characterised by high heat loads and air change rates. All cooking appliances release heat into the kitchen space in the form of convection and radiation. Kitchen hoods are designed to localize and

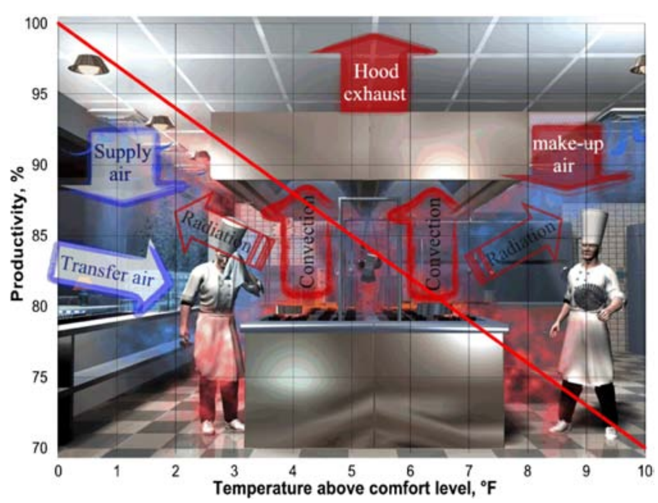

Figure 1 *Temperature and productivity in a kitchen.*

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capture convective heat and cooking effluents rising from hot cooking surfaces. Radiation heat from cooking appliances under the hoods, as well as heat from other sources in the kitchen (lights, people, heat transfer through the building envelope, heat from other equipment installed apart from the hoods), are transferred to the kitchen and have to be accounted for as load to the space. This load is used to properly size the air-conditioning system and achieve the desired temperature in the space. If this load is underestimated, the air-conditioning system will be undersized, and it will lack the cooling capacity to reach design temperature in the space—the kitchen will be hot.

All the air exhausted from the kitchen through the hoods has to be replaced with outside air. There are three sources of replacement air in an air-conditioned kitchen:

1. outside air portion of supply air delivered by the air-conditioning system
2. transfer air from adjacent dining room
3. a dedicated kitchen makeup air unit

The first two sources deliver conditioned air (cooled in summer) and the third usually brings untempered (not cooled in summer) air into the kitchen.

The amount of outside air that has to be cooled is the primary factor affecting cooling capacity of the air-conditioning system for a dominant part of the US with summer temperatures above 78°F (26°C). The higher the cooling capacity requirement of the air-handling unit (AHU), the higher its operating costs and energy consumption. That is why it is a common practice to bring most of the replacement air in the kitchen through a dedicated makeup air unit that doesn't condition the air—a solution that is inexpensive and not always effective. It is not uncommon to see a hood manufacturer claim that this makeup air can be delivered close to the hood in such a way that this air is being exhausted through the hood and has no affect on kitchen space air temperature.

A recent commercial kitchen ventilation study (CEC 2002) tested the impact of various makeup air systems on hood capture and containment. A few systems, including rear discharge (back drop plenum) and perforated perimeter supply (PPS), were identified as least affecting hood performance. Schrock (2002), studying thermal comfort in kitchen, discovered that outside air, hot and humid in summer, supplied through the back drop plenum ends up in the kitchen, thus being the main reason for a hot kitchen. VanStraten and Brown (2003) came to the same conclusion solving the makeup air problems for a family chain restaurant.

Field practice and experiments have shown that to date there isn't a method of bringing unconditioned makeup air into a kitchen such that it doesn't have an effect on the temperature and humidity in the space. It is not a question of whether or not the space temperature will increase, but rather how much the air temperature in the kitchen would increase when unconditioned makeup air is supplied into a kitchen on a hot summer day.

PERFORMANCE ANALYSIS OF THE PPS SYSTEM

As discussed earlier, one makeup air system that is being recommended because it was found to have minimal impact on the hood performance is the PPS system that introduces air from a horizontal plenum (installed near the ceiling) in a downward direction (CEC 2002). It was found that PPS has little or no impact on the hood performance at 150 fpm (0.762 m/s). However, one key parameter that has a large impact on how the PPS system performs in conjunction with the exhaust hood is the supply temperature of the air being introduced. For the CEC study, the air was introduced in a thermally neutral condition of 75°F (23.9°C).

To test the impact of the PPS system on the hood performance and kitchen comfort under different supply temperatures, four CFD scenarios were modeled using a commercial CFD code:

1. An exhaust-only style hood.
2. A PPS system supplying air at 75°F (23.9°C), which is similar to the configuration tested by Fisher.
3. A PPS system supplying air at 50°F (10.0°C).
4. A PPS system supplying air at 100°F (37.8°C).

For all of the PPS configurations, the supply air was introduced at a velocity of 150 fpm (0.762 m/s) to match what Fisher tested in his experiments (CEC 2002). The appliance modeled in the CFD scenarios had a surface temperature of 600°F (315.6°C) and the exhaust airflow was set to 248 cfm/ft (384 l/(s·m)) of hood length. The makeup air (when used) was equivalent to 80% of the exhaust airflow.

The three-dimensional CFD model for these simulations contained approximately 520,000 elements. The turbulence model used for the simulation was RNG k-ε, radiation was turned off, and a first order upwind advection scheme was utilized. The simulation was run in steady-state mode until the convergence criteria of a residual of 1E-4 and a global imbalance of 0.3% were met.

Results were generated for each of the scenarios showing the effect of the makeup air in the space and the comfort level in the kitchen. The exhaust-only system (see Figure 2) shows all of the convective heat from the appliance being captured by the hood, and the kitchen space air temperature is 75°F (23.9°C). For this scenario, no untempered makeup air was introduced to the space.

The results with the front PPS supplying 75°F (23.9°C) air [see Figure 3] are similar in terms of space temperature to the exhaust-only case. However, it is evident that the plume inside the hood is being induced slightly by the air from the PPS system as it passes the front of the hood.

The results with the front PPS supplying 50°F (10.0°C) air (as shown in Figure 4) are more dramatic in terms of the space temperature in the kitchen. For this scenario, there would be cold air bowling downward directly onto the chef, which could cause discomfort. Cold supply air, as it accelerates toward the floor, entrains the plume out from the hood.

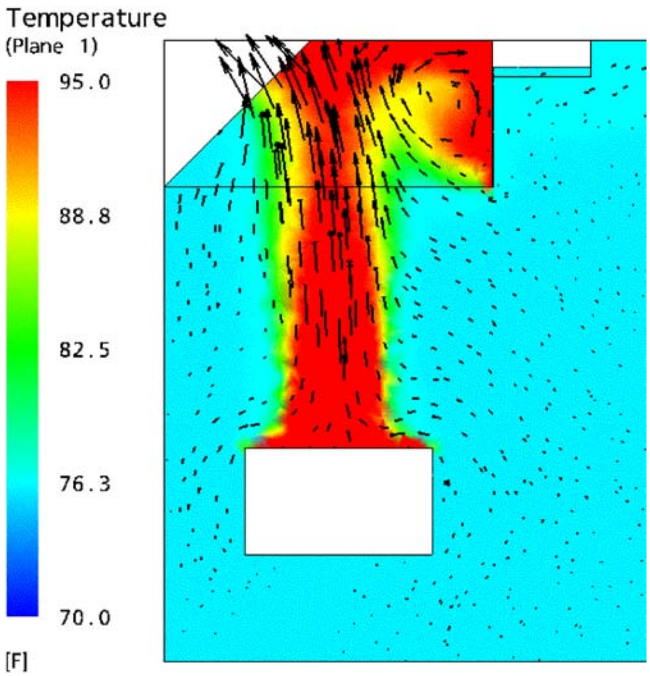

Figure 2 *CFD simulation, case 1—exhaust-only hood.*

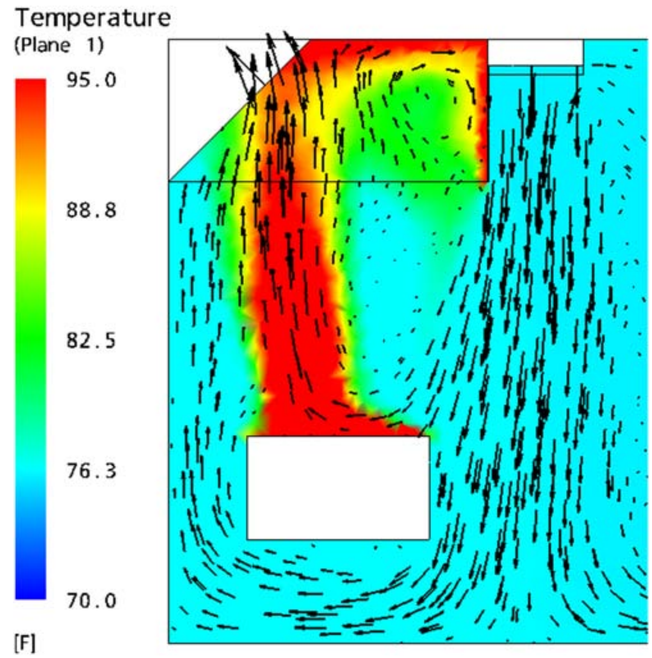

Figure 3 *CFD simulation, case 2—Front Supply at 75°F (23.9°C).*

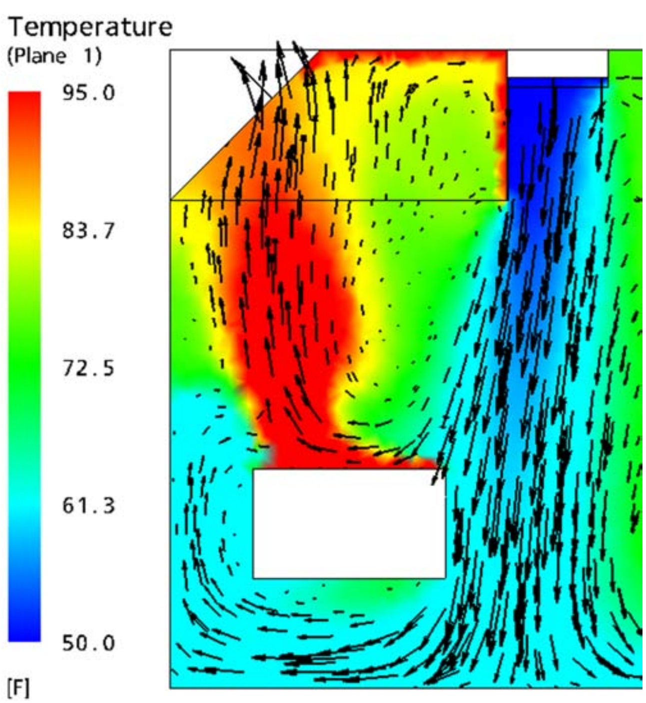

Figure 4 *CFD simulation, case 3—Front supply at 50°F (10.0°C).*

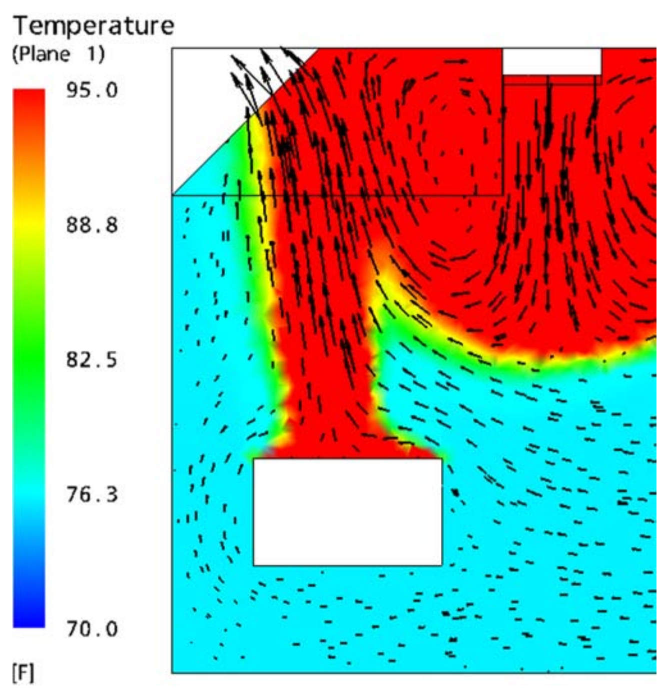

Figure 5 *CFD simulation, case 4—front supply at 100°F (37.8°C).*

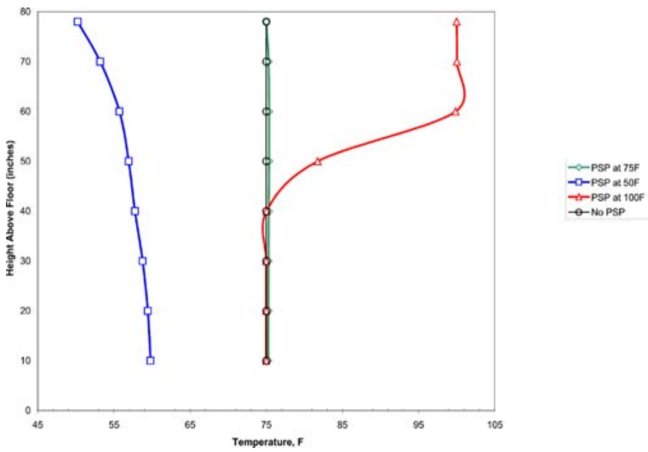

Figure 6 *Space Temperature vs. room height for various PPS configurations.*

Part of this cold air also impinges on the hot cooking surface and compromises the performance of the cooking appliance.

During the summer a PPS supply 100°F (37.8°C) air would have an extremely negative impact on the comfort in the kitchen (see Figure 5). For this scenario, the upper body of the chef would be enveloped by hot, and potentially humid, outdoor air.

A comparison of the space temperature as a function of height above the floor for the various PPS configurations is presented in Figure 6. This shows that the largest difference in the temperature that the chef is exposed to is in the upper body between the heights of 60 and 70 inches (1.5 to 1.8 m) from the floor. The sampling location for the space temperatures was 1 foot (0.305 m) from the front edge of the griddle to represent where a chef would stand.

CALCULATING THE EFFECT OF UNCONDITIONED MAKEUP AIR DELIVERY AND HOOD SPILLAGE ON KITCHEN SPACE TEMPERATURE

An energy balance equation (see Equation 1) can be used to calculate the supply airflow and supply air temperature needed to achieve a desired kitchen design air temperature. This equation assumes that hood is capturing and the untempered summer makeup air is captured by the hood without entering kitchen space.

$$m_s \cdot c_p(t_r - t_s) + m_{tr} \cdot c_p(t_r - t_{tr}) = Q_{hg} \tag{1}$$

where

m_s = supply airflow, lb/h [kg/s]

c_p = specific heat of air, Btu/(lb·°F) [J/(kg·K)]

t_r = design kitchen space air temperature, °F [°C]

t_s = supply air temperature, °F [°C]

m_{tr} = transfer airflow, lb/h [kg/s]

t_{tr} = transfer air temperature, °F [°C]

Q_{hg} = total design heat gain to the space, Btu/h [W]

If the hoods are not sized properly they will spill effluents and convective heat Q_{sp} into the kitchen; also, in summer, the untempered outside air, delivered through the conventional makeup systems, will add heat to the kitchen Q_{ma}. With the added heat gain to the space from Q_{sp} and Q_{ma} the source of cooling in the kitchen remains unchanged—it is the same amount of transfer and supply air delivered to the space at the same temperature. The new kitchen space air temperature t_r^1 as a result of additional heat gain to the space can be calculated from the following set of equations:

$$m_s \cdot c_p(t_r^1 - t_s) + m_{tr} \cdot c_p(t_r^1 - t_{tr}) = Q_{hg} + Q_{sp} + Q_{ma} \tag{2}$$

$$Q_{sp} = M_{sp} \cdot c_p(t_{sp} - t_r^1) \tag{3}$$

$$Q_{ma} = M_{ma} \cdot c_p(t_{oa} - t_r^1) \tag{4}$$

where

Q_{sp} = heat gain to the space due to the hood spillage—convective heat escaping from the hood, Btu/h [W]

Q_{ma} = heat gain to the space from untempered makeup air—hot in summer outside air entering the kitchen space, Btu/h [W]

m_{ma} = makeup airflow, lb/h [kg/s]

m_{sp} = amount of hot air spilling from under the hood, lb/h [kg/s]

t_r^1 = kitchen space air temperature as a result of additional heat gains to the space, °F [°C]

t_{sp} = temperature of convective airflow escaping from the hood, °F [°C]

t_{oa} = temperature of air, supplied through the makeup system, equal to or above[1] the outside air temperature in summer. °F [°C]

Solving the system of equations 1 to 4 allows calculating the new kitchen space air temperature as a result of additional heat gains to the space from hood spillage and untempered makeup air.

$$t_r^1 = \frac{(X_s + X_{tr}) \cdot t_r + X_{sp}t_{sp} + X_{ma}t_{ma}}{X_s + X_{tr} + X_{sp} + X_{ma}} \tag{5}$$

where

X_s, X_{tr}, X_{sp}, X_{ma} = corresponding ratios of m_s, m_{tr}, m_{sp}, m_{ma}, to the total hood exhaust airflow m_{hood}.

Equation 5 assumes fully mixed conditions —mixing air distribution is used and air temperature t_r^1 is uniform throughout the space.

1. Usually the air intake for a makeup system is located on the roof where temperature is several degrees above ambient.

Example 1

A restaurant kitchen air-conditioning system is designed to maintain a space air temperature t_r of 76°F (24.4°C). The design is based on an assumption that hoods are not spilling and the untempered makeup air has no effect on kitchen heat gain. Hoods are delivered with the makeup air system sized for 80% of the total hood exhaust, X_{ma} = 0.8. The makeup air is untempered in summer and, contrary to common perception, enters the kitchen space before being exhausted through the hood. Transfer airflow amounts to 10% of hood exhaust, X_{tr} = 0.1. A standard rooftop packaged air conditioner is used with 25% of outside air, with the total supply airflow of 40% of hood exhaust (10% outside and 30% return air), Xs = 0.4. Let's calculate the resulting kitchen air temperature t_r^1 when the outside air temperature is 96°F (35.5°C). In that case temperature of air, delivered through the makeup air system, will be at least t_{ma} = 98°F (36.7°C). It is not uncommon to see even higher t_{ma} since the air intake of the makeup system is on the roof. Using Equation 5,

$$t_r^1 = \frac{(0.4 + 0.1) \cdot 76 + 0.8 \cdot 98}{0.1 + 0.4 + 0.8} = 89.5°\mathrm{F}$$

$$t_r^1 = \frac{(0.4 + 0.1) \cdot 24.4 + 0.8 \cdot 36.7}{0.1 + 0.4 + 0.8} = 32.0°\mathrm{C}$$

that is, 13.5°F (7.6°C) above the design temperature.

Example 2

Let's use example 1 and assume that hoods are not capturing and spilling 20% of convective heat back into kitchen X_{sp} = 0.2 at t_{sp} = 100°F (37.8°C). The resulting kitchen air temperature will be

$$t_r^1 = \frac{(0.4 + 0.1) \cdot 76 + 0.2 \cdot 100 + 0.8 \cdot 98}{0.1 + 0.4 + 0.2 + 0.8} = 90.9°\mathrm{F}$$

$$t_r^1 = \frac{(0.4 + 0.1) \cdot 24.4 + 0.2 \cdot 37.8 + 0.8 \cdot 36.7}{0.1 + 0.4 + 0.2 + 0.8} = 32.7°\mathrm{C}$$

that is, 14.9°F (8.3°C) above the design temperature. The effect of spillage alone, considering there is no untempered makeup air, will result in 8°F (4.4°C) temperature rise in the kitchen.

Productivity Gain Due to Improved Thermal Comfort

High temperature in the kitchen causes thermal discomfort of employees, leading to productivity loss. As demonstrated in the examples above, it is not uncommon to see temperatures in the kitchen 10°F (5.5°C) and more above the comfort level. Such a high temperature may result in productivity loss of 30% (Wyon 1996). According to *2003 Restaurant Industry Operations Report*, an average full service restaurant spends 33% of their sales on salaries, wages, and benefits and has a before tax profit of only 4% of sales. Let's assume that kitchen staff accounts for 50% of salaries, wages, and benefits. Reducing temperature in a kitchen to a comfortable level would result in 30% increase in productivity of kitchen personnel—fewer employees would be required to do the same job. That will reduce labor costs to 28% of sales, and income before taxes will more than double to end up at 9% of sales.

Figure 7 *Mixing ventilation, supplying air through ceiling diffusers at the ceiling level.*

DISPLACEMENT VENTILATION—THE MORE EFFICIENT WAY TO COOL A KITCHEN

Most conventional kitchens use mixing ventilation to cool the space (Figure 7). In mixing ventilation systems, cool conditioned air is typically supplied through ceiling diffusers at a high discharge velocity. This high velocity is required to create a high momentum air jet for efficient mixing of supply air with room air. This air distribution system may not necessarily be the best fit for a commercial kitchen for two reasons:

1. High discharge velocity from mixing diffusers creates unwanted air movement and cross-drafts in the kitchen that make it difficult to capture and contain the plume from cooking appliances with the hoods.
2. All the heat rising from multiple heat sources in the kitchen is mixed within the space.

An alternative air distribution system—thermal displacement ventilation (TDV)—is a better fit for the kitchen environment. In TDV systems cool air is supplied at low velocity directly into the occupied zone (Figure 8). Rather than mix the heat and contaminants in the space, as the mixing system does, TDV stratifies and displaces them out of the occupied zone into the upper part of the space. As a result the air velocity in the space is low, with no undesired cross-drafts, which makes it easier for the hoods to capture and contain plumes from

Figure 8 *Thermal displacement ventilation, supplying air through displacement diffusers installed at the walls.*

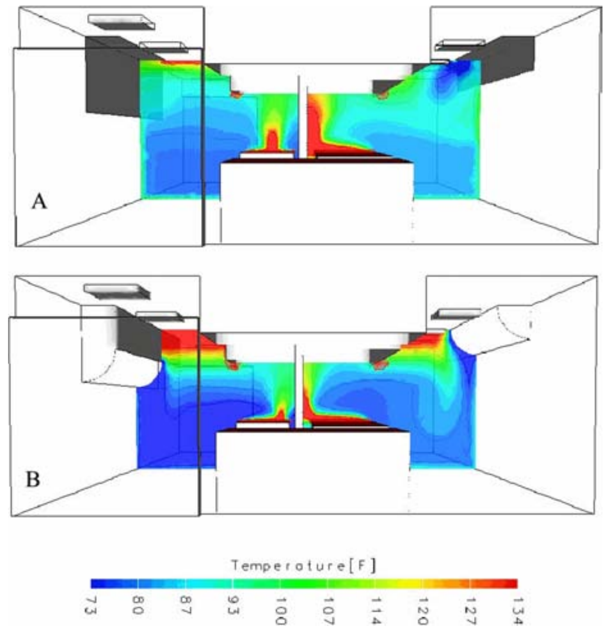

Figure 9 *Temperature in a commercial kitchen with (A) mixing ventilation and (B) displacement ventilation.*

cooking appliances. In fact, the hood exhaust airflow can be reduced by 12.5 % (VDI 2052 1999) when a mixing air distribution system is replaced with TDV.

CFD SIMULATION RESULTS

A commercial CFD model was generated on a typical layout for a popular family dining establishment in the United States. Two different configurations were run for this establishment: (1) a base case using mixing ventilation supplied from two-way ceiling diffusers and (2) the same supply air configuration using displacement diffusers located in the upper part of the room.

It should be noted that it is poor design practice to place a ceiling diffuser that blows supply air at a hood in close proximity to the hood. At a minimum it is recommended that diffusers blow away from the hood. However, this is still a common design practice in the restaurant industry. One reason is that some designers and engineers have the misconception that they can use the air from the ceiling diffusers to “cool” the chef, not realizing the negative impact this has on hood performance.

The primary reason for placing the diffusers in the upper part of the room is that kitchens generally lack sufficient floorspace to place diffusers at the floor, which is the optimal location for introducing air. Two effects that are seen when using diffusers mounted above the floor are that some warm air can be entrained back into the space, causing an increase in the actual supply temperature, and the velocity of the air can increase as it falls down from the diffusers to the floor. However, after the supply air reaches the floor it behaves like a traditional TDV system.

The total amount of supply air was 4,124 cfm (1.94 m^3/s) at 57°F (13.9°C) with corresponding air exhausted through three hoods. The appliances in the kitchen have a total power input of 207,500 Btu/h (60.8 kW). Each of these CFD simulations contained over 900,000 elements and used the k-ε turbulence model. The simulation was run in steady-state mode until the convergence criteria of a residual of 1E-4 and a global imbalance of 0.3% were met.

The results of the two CFD cases show a striking difference in the temperature in the occupied zone of the kitchen. Figure 9 shows a temperature slice in the kitchen in front of a gas under-fired broiler for the mixing and displacement system. It can be observed that the space temperature, especially in the lower occupied part of kitchen, is much cooler with the displacement ventilation system than with the mixing ventilation system.

A more accurate comparison of the temperature difference between the displacement and mixing ventilation systems at various heights in the room is shown in Figure 10. It can be observed that at a head height of 70 inches (1.8 m) the displacement system has a temperature of 85°F (29.4°C) and the mixing ventilation system has a temperature of 98°F (36.7°C). This result shows that displacement ventilation can have a positive impact on comfort in the kitchen space.

Besides occupant comfort, an additional benefit of using the displacement ventilation system is that it can improve the efficiency of the kitchen exhaust hoods due to a reduction in turbulence in the kitchen environment. Figure 11 presents a view of the airstreams from the diffusers with the mixing and displacement ventilation systems.

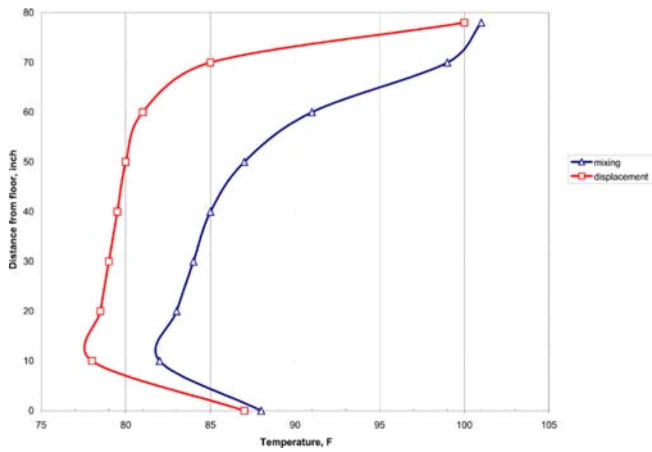

Figure 10 *A comparison of space temperature for mixing vs. displacement ventilation.*

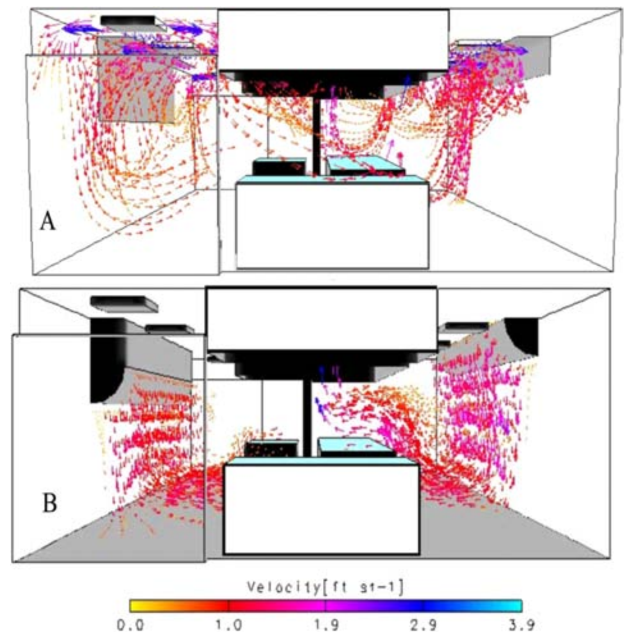

Figure 11 *Streamlines from a (A) mixing and (B) displacement ventilation systems.*

Several things can be observed by examining airstreams from mixing diffusers. First, it is apparent that the air exiting the diffusers travels across the ceiling, impinges on the front of the hood surface, and drops vertically downward, which will tend to disrupt the hood performance as verified by laboratory testing (CEC 2002). Secondly, it is evident that drafts are present in the kitchen as a result of the mixing ventilation system, which will further degrade the exhaust system performance, causing more convective heat to enter the kitchen space and cause discomfort for the employees.

After an evaluation of the streamlines from displacement diffusers, it is evident that the air from the diffusers is not impacting the front edges of the hoods. In practice this means that the exhaust systems will operate more efficiently and be more stable and employee comfort will improve.

CONCLUSIONS

CFD study of PPS plenum demonstrated that its 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 air supplied through the PPS.

A theoretical equation was developed to account for temperature rise in a kitchen when hoods are not capturing or untempered hot outside air is supplied into a kitchen space.

Delivering untempered makeup air into a kitchen on a hot summer day may result in temperature rise in a kitchen 10°F (5.5°C) and higher above the comfort level. Such a high temperature may result in 30% productivity loss of kitchen personnel.

TDV cooled the kitchen space more effectively than the mixing ventilation system. The CFD simulation results demonstrate up to 10°F (5.5°C) lower temperature in the kitchen when the mixing air distribution system was replaced with a TDV system having the same cooling capacity.

ACKNOWLEDGMENTS

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