Design Guide 3

Improving Commercial Kitchen Ventilation System Performance

Integrating Kitchen Exhaust Systems with Building HVAC

This design guide provides information that may help achieve optimum performance and energy efficiency in commercial kitchen ventilation systems by integrating kitchen exhaust with building HVAC. The information presented is applicable to new construction and, in some instances, retrofit construction. The audience for this guideline is kitchen designers, mechanical engineers, code officials, food service operators, property managers, and maintenance people. The building code analysis is focused on California's Title 24. This guide is intended to augment comprehensive design information published in the Kitchen Ventilation Chapter in the ASHRAE Handbook on HVAC Applications, as well as companion publications under the design series subtitled *Improving Commercial Kitchen Ventilation System Performance.*

The Opportunity: Reduce Utility Costs and Improve Kitchen Comfort

The replacement air required for commercial kitchen ventilation systems is always 100% of the exhaust air—what goes out must come in! A common design practice is to supply at least 80% of replacement air using an independent makeup air unit (MAU) with the remaining 20% supplied by conditioned outside air from heating, ventilating, and air-conditioning (HVAC) roof-top units (RTU) serving the kitchen and/or by transfer air from adjacent spaces. This keeps the kitchen under a negative pressure (relative to the dining room) to prevent cooking odors from migrating into the dining area. In many climates the replacement air from an independent makeup air unit is not conditioned, which may create uncomfortable conditions (too cold and/or too hot) in the kitchen. In other climates, the makeup air is heated, which in many cases results in simultaneous heating (by the MAU) and cooling (by the RTU) of the kitchen during the shoulder seasons. Conventional design practice does not take full advantage of the relatively high rate of occupancy ventilation air that is introduced into the dining room or other areas of the building adjacent to the kitchen.

There is an opportunity to use code-required outdoor air supply to the dining room as replacement air, thus reducing or eliminating the fraction of replacement air from the independent makeup air unit. Since occupancy ventilation air is conditioned in most cases, transferring it to the kitchen as a contribution to the replacement air requirement can improve comfort conditions in the kitchen.

The other design guides in this series explain the principles for selecting and sizing exhaust hoods, as well as the fundamentals of introducing replacement air to avoid degrading exhaust hood capture and containment. This guide explains the advantages and challenges of integrating replacement air with the building HVAC system to maximize the use of occupancy ventilation air as replacement air. The *Ventilation* section describes how outside air requirements are calculated based on occupancy or conditioned floor area. The *Food Service HVAC* section discusses design issues related to selection and control of rooftop HVAC units, including a building energy and control system and maintenance issues. The *Design Considerations* section describes methods to assure proper transfer of air to the kitchen space. This guide includes two design examples, based in part on the kitchen and hood design practices in Design Guides 1 and 2[1], showing applications to quick service and casual dining restaurants.

Ventilation

The first question to be answered is: "***How much occupancy ventilation air is available for use as transfer air?***" State and local building codes prescribe the ventilation rates for occupancy and kitchen exhaust. The California 2008 Building Energy Efficiency Standards for Residential and Nonresidential Buildings are commonly referred to as Title 24 (which is part of the California Code of Regulations). The Energy Efficiency Standards are only Part 6 of Title 24, which contains the entire state-wide building code. Part 6 provides that the design outdoor air ventilation rate shall be the greater of two methods for determining outside air rates. The first method [Section 121 (b) 2 A] is based on the conditioned floor area times a factor in Table 121-A, Minimum Ventilation Rates. Table 121-A of Title 24 Part 6 is reproduced below as Table 1. Restaurants are not specifically listed in the table, but since they can be classified as retail stores, the most appropriate outside air ventilation rate is 0.20 cfm/ft².

Table 1. California Energy Code Minimum Outdoor Air Ventilation Rates

Table 121-A Minimum Ventilation Rates	Type Of Use Cfm Per Square Foot Of Conditioned Floor Area
Auto repair workshops	1.50
Barber shops	0.40
Bars, cocktail lounges, and casinos	0.20
Beauty shops	0.40
Coin-operated dry cleaning	0.30
Commercial dry cleaning	0.45
High-rise residential	Ventilation Rates Specified by the CBC
Hotel guest rooms (less than 500 sq. ft.)	30 cfm/guest room
Hotel guest rooms (500 sq. ft. or greater)	0.15
Retail stores	0.20
All others	0.15

The second method [Section 121 (b) 2 B] requires 15 cfm per person times the expected number of occupants. The expected number of occupants for spaces without fixed seating is the greater of either (1) the number specified by the building designer or (2) one half of the maximum occupant load assumed

[1] http://www.fishnick.com/ventilation/designguides/

for egress purposes by the California Building Code. For spaces with fixed seating the expected number of occupants is determined in accordance with Part 2 of Title 24, which states that design occupancy is the number of seats. For dining rooms, Table 4-1 of the California Building Code (Title 24) sets egress occupancy at a maximum of 15 ft^2 per person.

Title 24 recognizes exceptions to the design minimum outside air flow rate if the space is designed with demand ventilation controls, which typically use carbon dioxide as a proxy for the current occupancy level. If demand ventilation controls are used, during periods of low occupancy the total ventilation flow may be reduced.

Title 24 [Section 121 (c) 3] requires demand ventilation controls on HVAC systems that have air economizers and that serve a space with design occupant density, or a maximum occupant load factor for egress, that is equal to or greater than 25 people per 1000 ft^2. Most restaurants have higher occupancy loads, so demand ventilation[2] must be considered in restaurant design (per Title 24).

There is an exception to this requirement if exhaust from the space is greater than the design ventilation rate specified in Section 121 (b) 2 B minus 0.2 cfm per ft^2 of conditioned area [Section 121 (c) 3, Exception 2]. This is an important exception that permits maximizing the use of occupancy ventilation air as transfer air and avoiding the need for demand ventilation in the dining room.

The application of these requirements to restaurants also depends on the size of the HVAC units (and whether they have an outdoor air economizer) and whether the dining room and kitchen are considered separate zones. These issues are discussed in the *Design Considerations* section.

[2] It is important to recognize that Title 24 is referring to demand ventilation as it would be applied to the dining area. This is not the same as the demand-ventilation controls on the kitchen exhaust hood that is becoming more popular in CKV system design.

Food Service HVAC

The second question to be answered is: "***How will selecting and sizing of heating, ventilating, and air-conditioning (HVAC) equipment affect the availability of transfer air?***" Restaurant HVAC is typically provided by constant-volume, packaged, single-zone air conditioning and heating units (commonly called roof top units (RTUs) since that is where they are usually located). The cooling/heating capacity and the number of RTUs selected for a restaurant depend on the estimated thermal loads, thermal zoning, first costs, and building code requirements.

The cooling and heating capacities of RTUs are selected based on the hottest and coldest days expected during a year. In addition, the cooling capacities of the dining room RTUs are sized assuming that peak (design) occupancy also occurs on peak cooling days. Likewise the kitchen RTUs are sized assuming that a peak business day (corresponding to a peak occupancy day in the dining room), with heavy cooking loads, occurs on peak cooling days.

The number of RTUs used in a design depends on the number of thermal zones, the amount of required occupancy ventilation air, the design latent load, differences in the amount of ductwork required, and the impact of RTU weight on the roof structural design. Thermal zoning divides the restaurant into areas that have similar thermal conditions during the day. For example, a small dining room with east and west windows may have two RTUs, one serving the east side and one serving the west. The latent load capacity of RTUs usually is no larger than 25% of total cooling capacity at design conditions, due to coil sizing and design air flow rates across the coil. This may be a factor that increases the number of units depending on the amount of occupancy outside air and latent load. The amount of above-ceiling space for duct runs and the arrangement of structural members in the roof may also affect the number of RTUs. Larger RTUs usually require larger and longer duct runs, which are often more difficult to install due to tight spaces above the ceiling and hence may be installed with field modifications that restrict airflow. Smaller RTUs will require shorter and smaller duct runs, which may reduce restrictions introduced during construction. Placement of RTUs on the roof also plays a role in the design of the roof structure, and hence the cost of the roof. Trade-offs may be made between the number of and weight of RTUs to minimize the cost of the roof framing.

The first (purchase and install) cost of an RTU depends on the cooling capacity, supply air capacity, and the presence or absence of other features such as air or water economizers and the sophistication of the unit controls.

Over the past several years, building codes have tended to demand better energy efficiency and improved air quality from packaged air conditioning units. Increases in required minimum energy efficiency ratios (EERs), duct sealing, improved programmable thermostats and unit controls, and air or water economizers are examples.

Title 24 [Part 6, Section 144 (e)] requires that each RTU with a design supply air capacity of over 2,500 cfm and a total mechanical cooling capacity over 75,000 Btu/hr (i.e., 6.25 tons) have: (1) an air economizer capable of modulating outside-air and return-air dampers to supply 100 percent of the design supply air quantity as outside-air; or (2) a water economizer capable of providing 100 percent of the expected system cooling load as calculated in accordance with a method approved by the commission[3], at outside air temperatures of 50°F dry-bulb/45°F wet-bulb and below. An air economizer is a damper section attached to the RTU that allows 100% of the design supply air to be outside air while shutting off the return air to the RTU. A water economizer is an RTU section that evaporates water directly (adding moisture to the supply air stream) or indirectly (no moisture added to supply air stream) to cool the supply air stream.

In air economizer mode, the supply fan for a typical small to medium size packaged RTU is not capable of providing sufficient power to supply outside air equal to 100 percent of design supply air without at least a passive barometric vent in the return duct or in the building envelope. If some type of relief vent is not present, economizer operation of the RTUs will likely overpressurize the building and the resulting backpressure will reduce the fan's ability to draw in 100% outside air. In some cases, a powered relief vent will be needed to assure that the economizer can draw in 100% of the design supply air.

When RTUs that serve the kitchen are in economizer mode, they may provide air to diffusers near the exhaust hood at higher than design velocities, which may interfere with proper effluent capture by the hood. This is due to the fact that using barometric relief often by-passes a substantial section of the re-

[3] The California Energy Commission promulgates Title 24, which has prescriptive and performance methods for calculating system cooling loads.

turn duct run, which reduces the static pressure that affects the fan operation. In this situation, the supply fan can provide a higher air flow rate for a given fan speed when the static pressure is reduced. Care should be taken to locate these diffusers as far from the hood as practicable to mitigate disruption to the hood capture and containment when economizer mode is used.[4]

Title 24 [Section 144 (e) 1, Exception 5] provides an exception to the mandatory economizer feature by selecting RTUs with cooling efficiencies (reported as an EER) that meet or exceed the requirements listed in Table 2. This is available for some California climate zones and also depends on the nominal cooling capacity of the unit (note that the EER values shown in the table are effective 1/1/2010). Climate zones can be located through the California Energy Commission website. (http://www.csd.ca.gov/agency/Energy/04%20California%20Climate%20Zone%20Map.pdf

Table 2. Title 24 Table 144-A

	Nominal Capacity (Btu/hr)			
Climate Zone	760,000	240,000 and <760,000	135,000 and <240,000	65,000 and <135,000
1	N/A	N/A	N/A	N/A
2	N/A	N/A	N/A	N/A
3	N/A	N/A	N/A	N/A
4	12.5	12.9	14.1	N/A
5	N/A	N/A	N/A	N/A
6	N/A	N/A	N/A	N/A
7	N/A	N/A	N/A	N/A
8	12.5	12.9	14.1	N/A
9	12.2	12.5	13.7	N/A
10	12.0	12.3	13.5	13.5
11	12.1	12.4	13.6	N/A
12	12.3	12.6	13.8	N/A
13	11.8	12.1	13.3	13.4
14	12.3	12.6	13.8	N/A
15	10.6	11.0	12.0	12.3
16	N/A	N/A	N/A	N/A

[4] Design issues related to introducing replacement (makeup) air into the kitchen and its impact on hood performance are the subject of Design Guide 2, *Improving Commercial Kitchen Ventilation Performance – Optimizing Makeup Air* (previously published by the California Energy Commission under the title *Improving Commercial Kitchen Ventilation Performance*).

Design Considerations

The third question to be answered is: ***"Should transfer air from the dining room to the kitchen be maximized or minimized?"*** Deciding whether to use a traditional design with an independent makeup air unit with minimal transfer air from adjacent spaces or to use an integrated ventilation design that maximizes transfer air, depends on a number of factors: physical layout, occupancy ventilation, code compliance alternatives, controls, and maintenance.

Physical Layout

In the case of a typical quick service restaurant, separation of the dining and kitchen areas is often no more than a serving counter. In effect, the entire restaurant is a single zone from the perspective of air movement within the space. In many casual dining restaurants, the kitchen is separated from the dining room by a wall. In this case the dining room and kitchen are separate zones. This presents an opportunity to either use the maximum amount of transfer air by integrating the design of the HVAC and kitchen ventilation systems, or keep them separate and use demand controlled ventilation in the dining room (and potentially in the kitchen on the exhaust hood and makeup air unit (MAU) as well.

Air that is removed from the kitchen through an exhaust hood must be replaced with an equal volume of outside replacement (makeup). The replacement air may come from an independent makeup air unit that discharges into the kitchen, outside air from the kitchen RTUs, and outside air from the dining room RTUs (as transfer air).

The overall air balance in the restaurant is usually designed so that the air pressure in the entire building is slightly positive relative to the outside when the mechanical system is running. However, the air pressure in kitchens, due to the exhaust hoods, is usually slightly negative relative to the dining room and adjacent spaces, and the outside. This naturally assists in transferring air from the dining room to the kitchen and tends to keep cooking odors in the kitchen. But it may also allow unconditioned air into the kitchen when back doors or drive-by windows are open.

If transfer air is used, there must be sufficient open area between the dining and kitchen zones such that the transfer air velocity is relatively low (e.g., less than 75 feet per minute), or properly sized ducts or other openings must be placed between the two spaces. There may or may not be door openings, open passages, or pass-through openings in the wall between the dining room and the kitchen. The amount of available open area from doors, passages, and pass-through openings may provide sufficient area to keep the transfer air velocity relatively low. Too much air moving through a pass-through opening can cool food quickly, resulting in customer complaints. If the amount of open area is insufficient to allow air transfer at low velocity, transfer ducts must be included in the design. If the transfer air ducts are long or, due to structural space constraints, are too small, an in-duct fan should be used.

Occupancy Ventilation

Dining room RTUs have a minimum outside air setting that is usually based on the design occupancy. During periods of low occupancy, the dining space is over-ventilated, and energy may be wasted by unnecessarily conditioning outside air. To minimize energy waste, Title 24 requires the use of demand ventilation controls (DVC) if certain HVAC and occupancy rules are met as described above.

Demand controlled ventilation in the dining room can be problematic if the outside air provided by the dining room RTUs is used as replacement air for the kitchen exhaust hood. If the kitchen exhaust hoods are constant flow rate systems, a dining room DVC system can starve the kitchen exhaust system of replacement air if transfer air from the dining room is used. On the other hand, if the kitchen exhaust system is controlled by a DVC system, a dining DVC may be acceptable under certain conditions. This typically would be when the menu preparation is not in sync with dining room occupancy and the ventilation of the kitchen and dining area are independent of each other. When peak menu production occurs close to peak occupancy, then it makes sense to utilize transfer air and combine the DVC systems. In this case, as the kitchen exhaust rate is reduced, so is the outdoor air being introduced at the RTUs.

Code Compliance Alternatives

There are essentially two paths for ventilation compliance with respect to dining room spaces under Title 24. Generally the dining room occupancy will be in the range of 50 to 70 people per 1000 ft^2. Title 24 requires demand controlled ventilation (based on CO_2 sensors) for spaces with occupancies equal to or greater than 25 people per 1000 ft^2 (40 ft^2 per person) [Section 121 (c) 3]. So most restaurants with fixed seats in the dining area would require DVC ventilation control, but usually one of the exceptions applies.

There are two exceptions that can be applied to restaurants:

1. If the RTUs are either small in design supply capacity (equal to or less than 2,500 cfm and a total mechanical cooling capacity equal to or less than 75,000 Btu/hr) [Section 144 (e) 1], or high enough in cooling efficiency [Section 144 (e) 1, Exception 5], the mechanical system designer may elect to not use economizers. If economizers are not used, then demand controlled ventilation is not required.
2. Under Section 121 (c) 3, Exception 2, if the volumetric flow rate of required exhaust air is larger than the design occupancy flow rate less 0.2 cfm/ft2 times the dining floor area (ft2), then DVC controls are not required. This would be the case where a large amount of transfer air from the dining room is used for a constant volumetric flow rate exhaust hood system in the kitchen.

Figure 1 lays out alternative paths for complying with Title 24's demand ventilation requirement for high occupancy areas, such as dining rooms. This is the first step in deciding whether to integrate the kitchen ventilation system with the dining room system. If demand controlled ventilation is not required, then analyzing an integrated ventilation approach is straightforward. If demand controlled ventilation is required in the dining room, transferring air to the kitchen for use as exhaust replacement air may lead to air balance control problems. Demand controlled ventilation for the exhaust hoods could be considered, but the designer must carefully evaluate when transfer air will be needed and when it will be available. This is often so complex that computer modeling (or comparing field data from stores with and without DCV systems) would be needed to provide insight into this scenario and the range of options is beyond the scope of this guide. A straightforward solution would be to use DVC controls on an independent makeup air unit and the exhaust hoods.

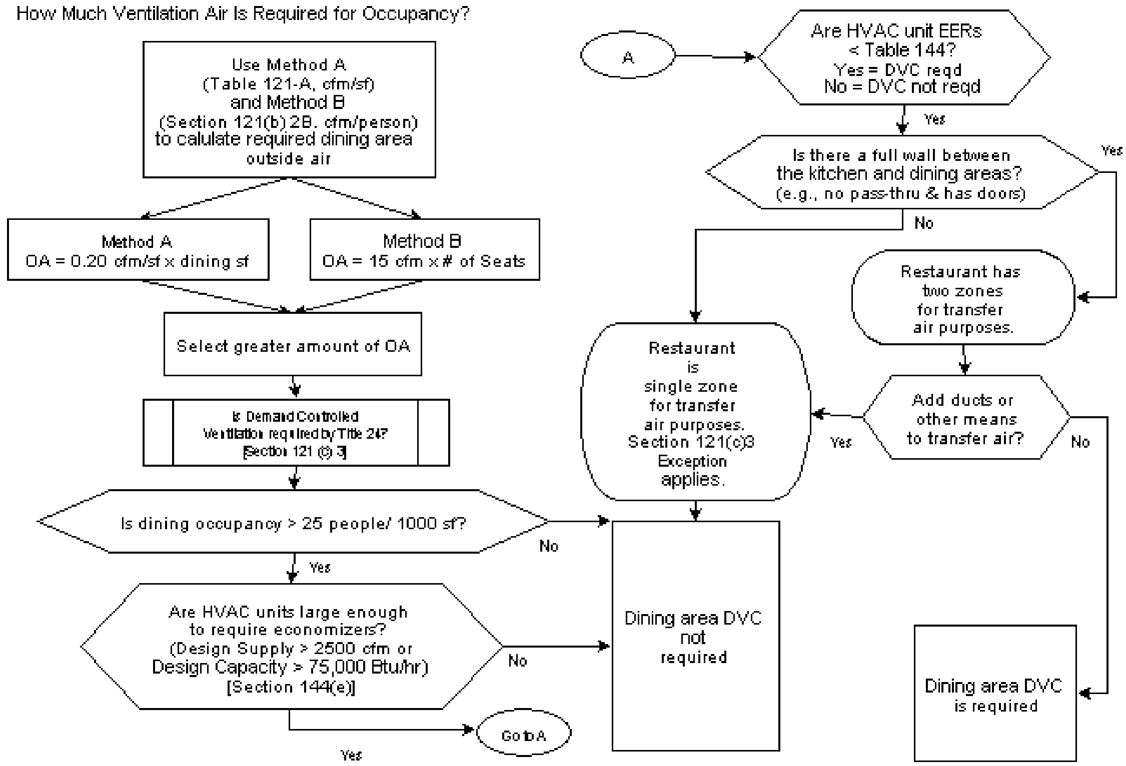

Figure 1. Process to Evaluate Whether Demand Ventilation Control is required under Title 24 for Restaurant Dining Areas

Controls and Maintenance

Other issues that affect the decision to integrate the building and kitchen ventilation systems are controls and maintenance. Keeping the systems separate provides some assurance that the hood will have the proper amount of replacement air. If the replacement air is supplied from multiple rooftop units, there will be less impact on hood performance if one of the RTUs is not running or outdoor air dampers are closed. This is often not noticed by restaurant staff unless customers complain. However, if the independent makeup air unit is not running, the hood may have difficulty maintaining proper capture and containment and the exterior doors will be difficult to open due to the negative pressure in the building. Usually effluent spilling from the hood will get the attention of the kitchen staff and maintenance action will be initiated.

Design Examples

Commercial Kitchen Ventilation (CKV) design examples, based on actual kitchen layouts, illustrate the design process and the potential for optimization. Each example starts with a base case that specifies separate makeup air for the kitchen exhaust hoods and concludes with a "best case" option that may be achieved through a more rigorous design effort.

Design Example A: Quick Service Restaurant (QSR)

Most quick service restaurants (QSRs) do not have a wall between the kitchen and the dining room. This makes it easy to use transfer air and many QSRs today are designed to transfer most of the outdoor air supplied to the dining room air into the kitchen space to be used as makeup air for the exhaust hoods. This also means that CO^2-controlled demand ventilation for the dining area would not have to be considered as an option under Title 24 and the ventilation design is essentially integrated.

The floor plan for this QSR example is 72 ft long and 40 ft wide (total 2880 ft^2). It is open for business 18 hours per day on weekdays and 19 hours per day on weekends.

The dining room is 1488 ft^2 with 75 seats. Part 2 of Title 24 (also known as the California Building Code) states that design occupancy for areas with fixed seating is the number of seats. Additionally, the designer should consider that the order area of a typical QSR has standing customers. For purposes of this example, let's assume that the designer includes 10 standing customers, for a total maximum occupancy of 85. Title 24 Section 121 (b) 2 A calculates occupancy ventilation as 0.2 cfm times the square footage, or 290 cfm. Section 121 (b) 2 B calculates occupancy ventilation based on design occupancy times 15 cfm per person, or 1275 cfm. The required outside air rate is the larger of the two calculations or 1275 cfm.

Since the dining room occupancy density is about 50 people/1000 ft^2 (greater than 25 people/1000 ft^2), demand controlled ventilation for the dining room is required unless one or more of the exceptions apply.

If we assume that the dining room load can be met with 15 refrigeration tons of capacity (nominal 180,000 Btu/hr total), there are two HVAC design options to consider relative to whether economizers are required on the RTUs. If the RTUs are sized smaller than 75,000 Btu/hr (6.25 tons cooling capacity) and less than 2500 cfm supply air, economizers are not required.

The first option would be a single 15 ton (refrigeration capacity) HVAC unit with 6000 cfm supply air. This unit requires an economizer since it exceeds the limits of 2,500 cfm and cooling capacity greater than 75,000 Btu/hr [Title 24, Part 6, Section 144(e)], and therefore DVC control for the dining area also is required.

On the other hand, we could use three 4 ton units (1600 cfm supply air and 48,000 Btu/hr nominal capacity) and one 3 ton unit (1200 cfm supply air and 36,000 Btu/hr nominal capacity) to achieve the same design capacity. But since these units are under the limits set by Section 144(e), no economizers are required. Since no economizers are required, DVC control for the dining area is not required.

In a typical QSR design, the kitchen and dining room could be considered a single zone since there is no wall between the two areas. Per Section 121 (c) 3, Exception 2, if the exhaust requirement for the kitchen hoods exceeds the design occupancy flow rate less 0.2 cfm/ft2 times the dining floor area (ft2), then DVC controls are not required. In the base case design, the minimum transfer amount would be about 980 cfm to avoid using DVC in the dining area.

QSRs are typically not designed with dining room CO^2-based DVC. The base case design will assume that the requirement of Section 121 (c) 3, Exception 2 is met. Either of the suggested HVAC solutions can be used in this case. The optimized design case includes the impact of reducing the kitchen hood exhaust as well as the HVAC solutions.

Base Case Air Balance for QSR

Air is exhausted from the restaurant by the kitchen exhaust hoods and the restroom exhaust fans. The exhausted air must be replaced by outdoor air. The three typical sources for the replacement air are: (1) a dedicated makeup air unit (that may or may not heat and/or cool the replacement air), (2) transfer air from other zones (usually a portion of the outside air supplied by the dining room and kitchen HVAC units), and (3) infiltration (if the air balance is not mechanically maintained).

The Base Case kitchen includes an unlisted wall-canopy hood requiring a 4600 cfm exhaust rate and an independent makeup air unit. Following industry practice, the dedicated makeup air unit is sized at 80% of the exhaust hood flow rate or 3700 cfm, maintaining the kitchen at a slight negative pressure relative to the dining room. The remainder of the replacement air requirement, 1400 cfm, for the kitchen hood, the restroom exhaust fans, and 200 cfm for overall building pressurization, is supplied by the dining room and kitchen rooftop HVAC units.

Table A-1 summarizes the building air balance for QSR Base Case, a design with multiple, small dining RTUs and economizes are not required. In this case, the amount of dining room transfer air is set to 1000 cfm, slightly greater than the 980 cfm required under Section 121 (c) 3, Exception 2. The outside air from the kitchen RTU is set to 400 cfm.

Table A-1 QSR Base Case Air Balance

Replacement and Exhaust Air	Maximum Outside Air for Occupancy (cfm)	Minimum Outside Air for Occupancy (cfm)	Outdoor (Replacement) Air Available for Kitchen Exhaust (cfm)	Exhaust Air (cfm)	Supply Air (SA) (cfm)	Nominal Cooling Capacity (refr. Tons)	Outside Air Fraction [OA/SA] %	SA cfm/ton
4 Dining Room Rooftop Units (RTUs)	1275	1275	1000		6000	15	21%	400
Kitchen Rooftop Unit (RTU)	150	150	400		5000	12.5	8%	400
Makeup Air Unit (sized at 80% of hood exhaust)			3700				100%	
Restroom Exhaust				300				
Kitchen Hood Exhaust				4600				
Total	1425	1425	5100	4900	11000	27.5	15%	
Net Outdoor Air for Building Pressurization			200					

Optimized Air Balance for QSR

To minimize energy use by the HVAC and kitchen ventilation systems, the first step is to minimize the amount of kitchen exhaust air. Design Guide 1 describes the process to minimize the kitchen hood exhaust rate. Using the design methods explained in Design Guide 1, the base case hood design of 4600 cfm can be optimized to require only 2200 cfm (utilizing an engineered backshelf hood design). This reduces the required outside air by 2400 cfm, or 65% of the makeup air unit design rate. Eliminating the makeup air unit is now possible by increasing the amount of transfer air from the dining room and kitchen HVAC units. The main benefits of eliminating the MAU are the first cost savings and possible heating and/or cooling energy savings.

The amount of transfer air is limited by the design of the dining and kitchen HVAC systems. These systems are typically composed of several single-zone, packaged, air conditioners that are roof-mounted. Depending on the climate and the design occupancy, the amount of outside air that each of these units can condition varies from 15 to 25 percent of the total supply air. Assuming the dining HVAC supply air will be 6000 cfm and the assumed maximum outside air fraction (i.e., Outside Air cfm divided by Supply Air cfm, or OA/SA) of 25% the maximum outside air available for use as transfer air would be 1500 cfm, which is sufficient to cover the required occupancy ventilation of 1275 cfm. The transfer in this design is unassisted because of the open design (lack of walls between the dining area and the kitchen area). Assuming the kitchen HVAC system has a supply air rate of 5000 cfm and a 25% OA fraction, an additional 1250 cfm of outside air could be used as replacement air.

The total building exhaust air amount is 2500 cfm (2200 cfm for the kitchen hood and 300 cfm for the restrooms). Assuming that 150 cfm in excess of the total exhaust would be sufficient for building pressurization, a total of 2650 cfm will be needed from the dining and kitchen RTUs.

Based on the maximum outside air from the dining and kitchen HVAC units (1500 cfm from dining and 1250 cfm from kitchen), a total of 2750 cfm is available as replacement air, which exceeds our total outside air requirement of 2650 cfm for the optimized design. This allows us to eliminate the makeup air unit and provides some flexibility in how we set up the air balance.

Table A-2 shows an alternate design solution that does not use dining occupancy DVC control. The outside air from the dining room RTU is set at 1475 cfm, which exceeds the minimum exhaust required under Title 24 Section 121 (c) 3, Exception 2. This would allow selection of either large or small RTUs for the dining room. The outside air for the kitchen RTU is set at 1175 cfm, which is within the assumed latent load capacity of the unit.

The outside air from the RTUs is 975 cfm greater than the base case design (1675 cfm). This is the trade-off for eliminating the 3700 cfm from the MAU, but it will likely improve the temperature comfort conditions in the kitchen and save heating energy costs in some California climate zones.

Table A-2 QSR Optimized Case: Reduced Exhaust (No MUA Unit), Large Dining RTU, and No Dining DVC

Replacement and Exhaust Air	Maximum Outside Air for Occupancy (cfm)	Minimum Outside Air for Occupancy (cfm)	Outdoor (Replacement) Air Transferred for Kitchen Exhaust (cfm)	Exhaust Air (cfm)	Supply Air (cfm)	Nominal Cooling Capacity (refr. Tons)	Outside Air Fraction [OA/SA] %	SA cfm/ton
Dining Room Rooftop Unit (RTU)	1275	1275	1475		6000	15	25%	400
Kitchen Rooftop Unit (RTU)	150	150	1175		5000	12.5	24%	400
Restroom Exhaust				300				
Kitchen Hood Exhaust				2200				
Total	1425	1425	2650	2500	11000	27.5	24%	800
Net Outdoor Air for Building Pressurization			150					

Design Example B: Casual Dining Restaurant

Our casual dining restaurant has a wall between the kitchen and the dining room (as is common for many casual dining and full service restaurants). Maximizing transfer air is more of a challenge and the application of CO^2-based demand ventilation in the dining room may need to be considered according to Title 24.

The restaurant floor plan is 81 ft long and 65 ft wide (total 5265 ft^2). It is open for business 18 hours per day on weekdays and 19 hours per day on weekends. The dining room is 2925 ft^2 with 150 seats. Part 2 of Title 24 (also known as the California Building Code) states that design occupancy for areas with fixed seating is the number of seats. Additionally, the designer should consider that serving staff for a typical casual dining restaurant of this size will be in order of about 10. For purposes of this example, let's assume that the designer includes 10 serving staff, for a total maximum occupancy of 160. Section 121 (b) 2 A calculates occupancy ventilation as 0.2 cfm times the square footage, or 585 cfm. Section 121 (b) 2 B calculates occupancy ventilation based on design occupancy times 15 cfm per person, or 2400 cfm. The required outside air rate is the larger of the two calculations; 2400 cfm in this example.

Title 24 Section 121 (c) 3 B requires demand controlled ventilation for spaces served by HVAC units that have economizers and that serve spaces with design occupancies greater that 40 ft^2 per person. Our example dining room has about 17 ft^2 per person, so demand ventilation would be required unless one of the code exceptions applies. Exception 2 to Section 121 (c) 3 states that demand ventilation is not required if exhaust from the space is greater than the design ventilation rate specified in Section 121 (b) 2 B minus 0.2 cfm per ft^2 of conditioned area. For this exception to apply, at least 1815 cfm (calculated as 2400 cfm less 585 cfm (i.e., 0.2 cfm/ft^2 times 2925 ft^2) must be transferred to the kitchen for use as replacement air. This exception is used in the Optimized Air Balance section below.

Since the design has a wall between the kitchen and dining room, how the air is transferred must be considered. Assuming the wall between the kitchen and dining room has a pass-through opening with 20 ft^2 of area (2-ft by 10-ft), and two doors, the available area would be 20 ft^2 (neglecting the unsealed area around the doors. In using these areas for transfer of replacement air, the main concern is to maintain the air velocity below 50 feet per minute (fpm) so that servings placed in the pass-through area do not cool off quickly. Assuming the velocity is maintained at 50 fpm, the total volumetric flow rate could be as high as 1000 cfm. In this case, the amount of available transfer air is below the threshold for Exception 2, and demand ventilation in the dining room would be required. Adding transfer grilles or ducts sufficient to transfer 815 cfm or more would allow the designer to use Exception 2.

If the wall between the kitchen and the dining room has open passage ways instead of doors with 54 ft^2 of area (equivalent to two doorways 3-ft by 9-ft each) and a pass-through opening with 20 ft^2 of area (2-ft by 10-ft), the total open area would be 74 ft^2. Assuming the velocity is maintained at 50 fpm, the total volumetric flow rate could be as high as 3700 cfm. The impact of these arrangements will be explored in the examples below.

If we assume that the dining room load can be met with 27 refrigeration tons of capacity, there are two HVAC design options to consider relative to whether economizers are required on the RTUs. If the RTUs are sized smaller than 75,000 Btu/hr (6.25 tons cooling capacity) and less than 2500 cfm supply air, economizers are not required and dining room DVC would not be required.

The first option would be to use RTUs that are larger than 6.25 tons cooling capacity. If one 12 ton (refrigeration capacity) and two 7.5 ton HVAC units are used in the dining room, the units will require economizers since it exceeds the limits of 2,500 cfm and cooling capacity greater than 75,000 Btu/hr [Title 24, Part 6, Section 144(e)]. DVC control is required unless Exception 2 to Section 121 (B) 2 B applies.

On the other hand, we could use six 4 ton units (1600 cfm supply air and 48,000 Btu/hr nominal capacity) and one 2 ton unit (800 cfm supply air and 24,000 Btu/hr nominal capacity) to achieve the same design cooling capacity. Since these units are under the limits set by Section 144(e), no economizers would be required. Since no economizers are required, DVC control would not be required.

Our design examples will use the three larger HVAC rooftop units (RTUs) to illustrate the issues involved with DVC and transfer air. Typically, full-service restaurant designers are not specifying RTUs less than 6.25 tons to avoid having to use economizers.

Casual Dining Base Case Air Balance

Air is exhausted from the restaurant by the kitchen exhaust hoods, dry storage exhaust, and restroom exhaust. The exhausted air must be replaced. The three typical sources for the replacement air are: (1) a dedicated makeup air unit (that may or may not heat and/or cool the replacement air), (2) transfer air from other zones (usually a portion of the outside air supplied by the dining room and kitchen HVAC units), and (3) infiltration (if the air balance is not proper).

The amount of replacement air from the HVAC units is limited by the design of the dining and kitchen HVAC systems. These systems are typically composed of several single-zone, packaged, air conditioners that are roof-mounted. Depending on the climate and the design occupancy, the amount of outside air that each of these units can condition varies from 15 to 25 percent of the total supply air. Assuming our dining occupancy is 160 people, we will need 2,400 cfm of outside air according to Title 24, which requires 15 cfm/person.

Table B-1 summarizes the building air balance for the Base Case. The wall-canopy exhaust hoods in the Base Case require 7600 cfm. Following industry practice, the dedicated makeup air unit is sized at 80% of the exhaust hood flow rate or 6100 cfm, maintaining the kitchen at a slight negative pressure relative to the dining room. For this design example, the MAU provides unconditioned makeup air. The dining room and kitchen rooftop HVAC units have economizers and supply the remainder of the replacement air requirement, for the kitchen hood, dry storage, restroom exhaust, and overall building pressurization. To meet part of the kitchen hood exhaust requirement and provide conditioned air for the kitchen workers, the outside air for the kitchen RTU is maximized and set at 1200 cfm.

Since the dining room is a separate zone in the example, and the amount of transfer air totals 800 cfm, which is less than 1815 cfm as calculated in accordance with Exception 2 to Section 121 (c) 3, CO^2-based demand ventilation control must be used in the dining room. The maximum outside air settings for the dining room RTUs would be 2400 cfm in aggregate. If the restaurant operator thought that a typical low occupancy period would have 10% of the maximum number of customers (i.e., 15 out of 150), the minimum outside air flow rate would be 285 cfm (225 cfm for customers, plus a proportionate amount for servers (say 4 servers), or an additional 60 cfm). Using DVC, the outside air rate would be increased by the control system as carbon dioxide levels in the space increased above the ambient level (about 400 ppm). Note that this design does not maximize the benefit of the DVC in this case because transfer air requirement, 800 cfm, is greater than the minimum occupancy outside air (285 cfm).

This design uses open passage ways and the pass-through to transfer air from the dining room to the kitchen. The air velocity through these openings should be no more than about 50 fpm. From Table B-1, 800 cfm transfer air should result in velocities just over 10 fpm.

If the pass-through did not exist, the velocity would be 15 fpm. If the open passage way were reduced to one door way (i.e., to 27 ft^2), the velocity would be 30 fpm. If the passage way had doors, then a transfer duct would be needed.

Table B-1. Casual Dining Base Case Air Balance.

Replacement and Exhaust Air	Maximum Outside Air for Occupancy (cfm)	Minimum Outside Air for Occupancy (cfm) 10% of Max	Outdoor (Replacement) Air Available for Kitchen Exhaust (cfm)	Exhaust Air (cfm)	Supply Air (cfm)	Nominal Cooling Capacity (refr. Tons)	Outside Air Fraction [OA/SA] %	SA cfm/ton
Dining Room Rooftop Unit #1 (RTU-1)	1050	125	400		4800	12	22%	400
Dining Room Rooftop Unit #2 (RTU-2)	675	80	200		3000	7.5	23%	400
Dining Room Rooftop Unit #3 (RTU-3)	675	80	200		3000	7.5	23%	400
Kitchen Rooftop Unit (RTU-4)	300	300	1200		4800	12	25%	400
Makeup Air Unit (sized at 80% of hood exhaust)			6100				100%	
Restroom Exhaust				300				
Dry Storage Exhaust				100				
Kitchen Hood Exhaust				7600				
Total	2700	585	8100	8000	15600	39	23%	
Net Outdoor Air for Building Pressurization			100					

Optimized Air Balance for Casual Dining Example

To minimize energy use by the HVAC and kitchen ventilation systems, the first step is to minimize the amount of kitchen exhaust air. Design Guide 1[5] describes the process to minimize the kitchen hood exhaust rate. Using the methods from Design Guide 1, the exhaust requirement is reduced from 7600 cfm (base case) to 4800 cfm (optimized wall-canopy hoods). In Design Guide 1, this level of exhaust reduction was achieved by replacing conventional wall canopy hoods with engineered UL listed hoods in combination with a slightly rearranged cookline. Tables B-2 shows optimized design air balance alternatives based on reduced kitchen exhaust, Case I.

The outside air required for the dining room occupancy ventilation is 2400 cfm. All of this outside air could be used as transfer air from the dining area to the kitchen for use as exhaust replacement (makeup) air.

However, in a typical restaurant mechanical design, only a portion the outside air from the dining and kitchen HVAC units is used as replacement air, as shown in Table B-2. The exhaust hood fans and the makeup air unit fan can be downsized using this optimized design alternative. Note that the amount of air transferred from the dining room (2100 cfm) is greater than the 1815 cfm threshold for demand ventilation control. The air balance in Table B-2 would not require independent demand ventilation control in the dining room.

This design uses the open passage ways and the pass-through to transfer air from the dining room to the kitchen. The air velocity through these openings should be no more than about 50 fpm. The 2100 cfm of transfer air from the dining room should result in velocities less than 30 fpm.

If the pass-through did not exist, the velocity would be less than 40 fpm. If the open passage ways were reduced to one door way (i.e., to 27 ft^2), the velocity would be under 80 fpm. This would require a transfer duct or grille, as would a design with doors.

[5] http://www.fishnick.com/ventilation/designguides/CKV_Design_Guide_1_031504.pdf

Table B-2. Case I Optimized Design Air Balance with Engineered Hoods and MAU Size Reduced.

Replacement and Exhaust Air	Maximum Outside Air for Occupancy (cfm)	Minimum Outside Air for Occupancy (cfm) 10% of Max	Outdoor (Replacement) Air Available for Kitchen Exhaust (cfm)	Exhaust Air (cfm)	Supply Air (cfm)	Nominal Cooling Capacity (refr. Tons)	Outside Air Fraction [OA/SA] %	SA cfm/ton
Dining Room Rooftop Unit #1 (RTU-1)	1050	110	900		4800	12	19%	400
Dining Room Rooftop Unit #2 (RTU-2)	675	70	600		3000	7.5	20%	
Dining Room Rooftop Unit #3 (RTU-3)	675	70	600		3000	7.5	20%	
Kitchen Rooftop Unit (RTU-4)	300	300	600		4800	12	13%	400
Makeup Air Unit (sized at 33% of hood exhaust)			2600					
Restroom Exhaust				300				
Dry Storage Exhaust				100				
Kitchen Hood Exhaust				4800				
Total	2700	550	5300	5200	15600	39	17%	
Net Outdoor Air for Building Pressurization			100					

Table B-3 shows an alternative air balance (Case II). Using the methods from Design Guide 1, the exhaust requirement is reduced from 7600 cfm (base case) to 3300 cfm (optimized design). In Design Guide 1, this level of exhaust reduction was achieved by replacing conventional wall canopy hoods with engineered backshelf hoods in combination with a smaller UL listed, wall canopy hood (It is recognized that the significant exhaust rate reduction requires the use of a backshelf or proximity style hood which may or may not be acceptable to the restaurant operations team). By reducing the exhaust air and maximizing the amount of transfer air from the dining room based higher outside air fractions the independent makeup air unit is eliminated. The final step to eliminating the MAU and achieving air balance comes from bringing in an additional 1200 cfm of outside air through the kitchen RTU. This assumes that the kitchen HVAC system has a supply air rate of 4800 cfm and with a 25% OA fraction, an additional 1200 cfm of outside air could supplied and used as replacement air.

This design uses the open passage way and the pass-through to transfer air from the dining room to the kitchen. The air velocity through these openings should be no more than about 50 fpm. The 2600 cfm of transfer air from the dining room should result in velocities of 35 fpm. If the pass-through did not exist, the velocity would be less than 50 fpm. If the open passageways were reduced to one door way (i.e., to 27 ft^2), the velocity would be about 95 fpm. This would require a transfer duct or grille, as would a design with doors.

Table B-3. Case II Optimized Design Air Balance with Maximum Transfer Air and Eliminated MAU.

Replacement and Exhaust Air	Maximum Outside Air for Occupancy (cfm)	Minimum Outside Air for Occupancy (cfm) 10% of Max	Outdoor (Replacement) Air Available for Kitchen Exhaust (cfm)	Exhaust Air (cfm)	Supply Air (cfm)	Nominal Cooling Capacity (refr. Tons)	Outside Air Fraction [OA/SA] %	SA cfm/ton
Dining Room Rooftop Unit #1 (RTU-1)	1050	n/a	1200		4800	12	25%	400
Dining Room Rooftop Unit #2 (RTU-2)	675	n/a	700		3000	7.5	23%	
Dining Room Rooftop Unit #3 (RTU-3)	675	n/a	700		3000	7.5	23%	
Kitchen Rooftop Unit (RTU-7)	300	n/a	1200		4800	12	25%	400
Restroom Exhaust				300				
Dry Storage Exhaust				100				
Kitchen Hood Exhaust				3300				
Total	2700		3800	3700	15600	39	24%	
Net Outdoor Air for Building Pressurization			100					

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The examples in this design guide show representative cases for illustration of design concepts only. Application of the concepts to particular designs may result in savings that are lower or higher than those depicted in the examples. Close coordination with local code officials, hood and fan manufacturers, and construction contractors is highly recommended.