BN-97-17-3

Minimum-Energy Kitchen Ventilation for Quick Service Restaurants

Vernon A. Smith, J.D., P.E.
Associate Member ASHRAE

Donald J. Frey, P.E.

C. Victor Nicoulin

ABSTRACT

Commercial cooking equipment exhaust systems have a significant impact on the total energy consumption of food-service facilities. It is estimated that commercial cooking exhaust ventilation capacity in food-service facilities across the United States totals 3 billion cfm (1.4 billion L/s) with an associated annual energy cost approaching $3 billion, based on an average of $1/cfm ($0.47 per L/s) per year. Significant energy and cost savings can be achieved by reducing ventilation rates. There are different optimum constant ventilation rates for gas-electric and all-electric kitchens that differ by climatic zone and that result in the development of minimum energy ventilation (MEV) strategies. This paper documents a preliminary investigation using computer simulations on the effects of different levels of constant ventilation rates for wall-mounted canopy and backshelf exhaust hoods in quick service gas-electric and all-electric restaurants.

INTRODUCTION

It has been reported (Claar et al. 1985) that the heating, ventilating, and air-conditioning (HVAC) load represents 30% of the total energy consumed in restaurants and that up to 75% of this load can be directly attributed to the operation of the kitchen exhaust ventilation system (Fisher 1986). The kitchen exhaust ventilation system is often the largest energy-consuming component in a commercial food-service facility. It is estimated that commercial cooking exhaust ventilation capacity in food-service facilities across the United States totals 3 billion cfm (1.4 billion L/s) with an associated annual energy cost approaching $3 billion, based on an average of $1/cfm ($0.47 per L/s) per year (Claar et al. 1995).

Existing building codes and design standards prescribe kitchen ventilation rates greater than are needed to ensure complete capture and containment of cooking emissions, smoke, and heat. Twenty-five years ago, commercial kitchen ventilation (CKV) design practices were based on practical experience and no known engineering research. In the last decade, field, laboratory, and simulation research on capture and containment and general kitchen ventilation efficiency has developed a significant body of knowledge regarding kitchen ventilation. Minimum capture and containment exhaust rates for commercial kitchen ventilation are under investigation at two laboratories in the United States. It is evident from laboratory tests that current ventilation exhaust rates required by building codes are significantly in excess of actual requirements for capture and containment (EPRI 1996a-1996h).

It is important to recognize that a large percentage of the CKV systems being installed today are designed and operated below the code-ventilation rates. Many of the commercially available exhaust hoods have been tested using Underwriters Laboratories (UL) Standard 710 (UL 1996) at airflow rates significantly below code (e.g., 300 cfm vs. up to 450 cfm per linear foot of hood [465-697 L/s per m of hood] and are permitted by the "authority having jurisdiction" as "engineered" systems. The National Fire Protection Association's Standard 96 (NFPA 1996), which is a national code, simply states that "exhaust air volumes for hoods shall be of sufficient level to provide for the capture and removal of grease laden vapors."

Significant energy and cost savings can be achieved by reducing ventilation rates. Laboratory tests have also demonstrated that radiant heat gain to the kitchen space is greater from hooded gas appliances than from hooded electric appliances (EPRI 1996a-1996h). The incremental heat gain to the kitchen from hooded gas appliances could create an increased cooling load during periods of hot ambient temperatures and a decreased heating load during periods of cold ambient temperatures. This raises the possibility that there are optimum constant ventilation rates that will differ by climatic zone and fuel type and that result in the development of minimum energy ventilation (MEV) strategies. To define MEV strate-

Vernon A. Smith is a senior engineer/project manager, **Donald J. Frey** is executive vice president, and **C. Victor Nicoulin** is a staff engineer for the Architectural Energy Corporation in Boulder, Colo.

gies requires quantifying the impact of climatic conditions on the energy consumption of restaurant ventilation systems.

Objectives

The primary purpose of this work was to investigate three issues that relate to the energy performance of commercial kitchen ventilation systems:

- the effects of increasing or decreasing exhaust hood air flow rates on kitchen HVAC system energy consumption and energy costs,
- the potential energy and cost savings from the use of high flow rate economizers to reduce cooling requirements in commercial kitchens, and
- the impact on kitchen HVAC energy consumption and energy costs from cooking appliance radiant heat gain reductions that occur as exhaust flow rates increase.

A secondary purpose for performing this work is to attempt to identify, in the context of each of the three issues above, kitchen ventilation design and operating strategies that minimize kitchen HVAC system energy consumption. These strategies may be termed "minimum energy ventilation" (MEV) strategies.

COMPUTER MODELS AND SIMULATIONS

The investigation of restaurant energy performance is greatly facilitated by the use of computer models and computer simulations. A simulation-based approach allows for proper accounting of the direct and indirect effects on building energy performance produced by changes in building design or operating characteristics. This is of particular importance in analyzing the energy performance of commercial kitchen facilities because of the interactions between cooking equipment and the kitchen ventilation system. Given the variety of cooking and ventilating equipment available and in use, it is important to understand the influence of this equipment on kitchen energy consumption and energy costs.

Computer Models

The project team selected a quick service restaurant as the type of restaurant building to model for this work. Nationally, quick service restaurants are the largest single and fastest growing food-service market sector. All building structures in the reference models meet regional building codes and standards in place as of 1993. Occupancy ventilation rates meet ASHRAE Standard 62-1989 requirements. Heating, ventilating, and air-conditioning (HVAC) systems in the reference models are packaged, direct expansion (DX) cooling systems that use either gas furnace or electric resistance heating according to building fuel type. The full load AFUE for the gas furnace was 0.75. Kitchen HVAC systems do not include economizers.

Table 1 shows maximum input ratings and daily consumption amounts for the major cooking appliances, including a six-foot griddle, three atmospheric fryers, one pressure fryer, and a six-burner range/convection oven. The cookline arrangement is similar to that found in architectural drawings from several major quick service chains. All major cooking appliances are hooded, using wall-mounted canopy hoods or backshelf hoods. Make-up air for the kitchen exhaust hoods is supplied through the dining room and kitchen HVAC units in all buildings.

A total of four baseline computer models were created: (1) a gas/electric restaurant with canopy hoods, (2) a gas-electric restaurant with backshelf hoods, (3) an all-electric restaurant with canopy hoods, and (4) an all-electric restaurant with backshelf hoods.

Computer Simulations

Hour-by-hour simulation software based on ASHRAE energy calculation methods was chosen for this work because of the modeling flexibility offered by the program (UW 1990). It readily allows accurate modeling of internal heat gains and building ventilation systems typically found in food-service establishments.

The computer simulations were conducted using the four baseline computer models and typical meteorological year (TMY) weather data for five cities: Akron, Atlanta, Los Angeles, Oklahoma City, and Phoenix. A variety of climates in which to examine building energy performance was recognized as an important factor in identifying kitchen ventilation strategies that minimize energy consumption by the kitchen HVAC system and exhaust hoods. Table 2 summarizes several important annual and design criteria figures for each location.

In order to identify the ventilation strategies that result in the minimum energy consumption, the project team performed three groups of computer simulations. Each group was designed to measure the kitchen HVAC system's response to variations in exhaust hood flow rate, economizer airflow rates, and kitchen heat gains from cooking appliances. This was accomplished by varying the values of specific computer model input parameters. The three groups of computer simulations are described in greater detail in the paragraphs below.

Kitchens with Varying Exhaust Flow Rates. The first group of simulations investigates the effects of varying exhaust hood airflow rates on kitchen HVAC system energy consumption. These simulations are conducted for two reasons. The first is to quantify the energy consumption and cost implications associated with using the kitchen ventilation system to remove heat from the kitchen space, particularly at higher exhaust flow rates. The second is to identify the exhaust flow rate that results in the minimum amount of energy used by the kitchen HVAC system and to assess the role that climate plays in the determination of the minimum energy flow rate.

To accomplish this, four exhaust flow rates ranging from 7200 cfm to 3150 cfm (3398 L/s to 1487 L/s) were selected for the canopy hood models. Three exhaust flow rates, ranging from 4800 cfm to 2400 cfm (2266 L/s to 1133 L/s), were selected for the backshelf hood models. The chosen flow rates correspond to values established by recognized or emerging

TABLE 1
Quick Service Restaurant Model Cookline Equipment Summary

Qty	Appliance	Gas Input Rating	Daily Energy Use, therms (MJ)		Electric Input Rating	Daily Energy Use, kWh (MJ)	
		kBtu/h (kW)	Weekday	Weekend	kW	Weekday	Weekend
1	Griddle	162 (47)	6.7 (707)	7.7 (812)	36	85 (306)	92 (331)
3	Atmospheric Fryers	110 (32)	7.3 (770)	8.9 (939)	17	90 (324)	110 (396)
1	Pressure Fryer	85 (25)	1.1 (116)	1.3 (137)	11.25	18 (65)	21 (76)
1	Six-burner Range Top	120 (35)	1.6 (169)	2.1 (222)	12	19 (68)	28 (101)
1	Range Convection Oven	30 (9)	0.2 (21)	0.2 (21)	6.5	4 (14)	4 (14)
	Totals:	727 (213)	16.9 (1,783)	20.2 (2,131)	116.75	216 (778)	255 (918)

TABLE 2
Climatic Data for Selected Locations

Climate Parameter	Akron	Atlanta	Los Angeles	Oklahoma City	Phoenix
Heating Degree-Days — Base 65°F (18°C)	6160 (3422)	2991 (1662)	1819 (1011)	3666 (2037)	1550 (750)
Cooling Degree-Days — Base 65°F (18°C)	625 (347)	1667 (926)	615 (342)	1930 (1072)	4162 (2312)
97.5% Heating Design — Temperature, °F (°C)	6 (-14)	22 (-6)	43 (6)	13 (-11)	34 (1)
2.5% Cooling Design — Temperature, °F (°C)	86 (30)	92 (33)	80 (27)	97 (36)	107 (42)
Mean Coincident Wet-Bulb — Temperature, °F (°C)	71 (22)	74 (23)	68 (20)	74 (23)	71 (22)
Mean Coincident Humidity Ratio	0.0129	0.0140	0.0120	0.0128	0.0080

standards that govern kitchen exhaust system design. The lowest flow rates are still sufficient to provide full capture and containment of cooking effluent from the assumed cooklines. Table 3 summarizes the complete selection of exhaust flow rates selected for this group of simulations.

The canopy hood model assumes a total of 18 linear feet (5486 mm) of exhaust hood, while the backshelf hood model assumes a total of 16 linear feet (4877 mm) of hood. The difference in hood lengths is due to the overhang requirements imposed on canopy hood designs.

Canopy hood flow rates described in Table 3 as "Above Code" correspond to an exhaust flow rate of 400 cfm per linear foot (cfm/lf) (619 L/s per m) of exhaust hood. "Code" flow rates are those specified by the mechanical codes in force for each of the five cities: the *Standard Mechanical Code* for Atlanta, the *National Mechanical Code* for Akron and Oklahoma City, and the *Uniform Mechanical Code* for Los Angeles and Phoenix. Code exhaust flow rates per linear foot range from 344 cfm/lf to 242 cfm/lf (533 - 375 L/s per m). "UL-Listed" flow rates are obtained from design procedures recommended by a leading manufacturer of UL-listed exhaust hoods and are based on an average exhaust rate of 250 cfm/lf (387 L/s per m). The "Custom" exhaust flow rates are determined from laboratory test results and correspond to an exhaust flow rate of 175 cfm/lf (271 L/s per m).

Backshelf hood flow rates described as "Code" flow rates are also as specified by the applicable mechanical code and correspond to an exhaust rate of 300 cfm/lf (465 L/s per m). "UL-Listed" exhaust rates are obtained using manufacturer's design procedures. The resulting average exhaust rate per linear foot of hood is 250 cfm (387 L/s per m). "Custom" exhaust rates for backshelf hoods are based on a laboratory-tested flow rate of 150 cfm/lf (232 L/s per m).

Kitchens with Enhanced Economizer Cooling. The second group of computer simulations examines "super ventilation" (SV) economizer systems and their ability to remove heat from the kitchen space. These simulations were conducted to more thoroughly explore the role that HVAC system economizers may play in reducing the mechanical cooling requirements of the kitchen space. The simulation results will help to quantify the potential for reducing energy consumption for kitchen space cooling and, therefore, reducing restaurant energy costs.

The project team defined the term "super ventilation" as "ventilation provided by the kitchen HVAC system in excess of that which the same system would provide under typical operating conditions." The excess ventilation capability is realized through the use of powered relief fans that are installed in the return half of the kitchen HVAC system. Relief fan operation is controlled to occur only during economizer cycles. Thus, the SV system is envisioned as a conventional HVAC system with an economizer assisted by a dedicated powered exhaust fan.

TABLE 3
Exhaust Flow Rate Summary for Varying Exhaust Flow Simulations

CANOPY HOODS					
		Exhaust Rate Category, cfm (L/s)			
Location	Fuel Type	Above Code	Code	UL-Listed	Custom
Akron	Gas/Electric	7200 (3398)	6200 (2926)	4500 (2124)	3150 (1487)
	All-Electric	7200 (3398)	6200 (2926)	4500 (2124)	3150 (1487)
Atlanta	Gas/Electric	7200 (3398)	6200 (2926)	4500 (2124)	3150 (1487)
	All-Electric	7200 (3398)	6200 (2926)	4500 (2124)	3150 (1487)
Los Angeles	Gas/Electric	7200 (3398)	5450 (2572)	4500 (2124)	3150 (1487)
	All-Electric	7200 (3398)	4360 (2058)	4500 (2124)	3150 (1487)
Oklahoma City	Gas/Electric	7200 (3398)	6200 (2926)	4500 (2124)	3150 (1487)
	All-Electric	7200 (3398)	6200 (2926)	4500 (2124)	3150 (1487)
Phoenix	Gas/Electric	7200 (3398)	5450 (2572)	4500 (2124)	3150 (1487)
	All-Electric	7200 (3398)	4360 (2058)	4500 (2124)	3150 (1487)
BACKSHELF HOODS					
		Exhaust Rate Category, cfm (L/s)			
Location	Fuel Type	Above Code	Code	UL-Listed	Custom
Akron	Gas/Electric	n/a	4800 (2266)	4000 (1888)	2400 (1133)
	All-Electric	n/a	4800 (2266)	4000 (1888)	2400 (1133)
Atlanta	Gas/Electric	n/a	4800 (2266)	4000 (1888)	2400 (1133)
	All-Electric	n/a	4800 (2266)	4000 (1888)	2400 (1133)
Los Angeles	Gas/Electric	n/a	4800 (2266)	4000 (1888)	2400 (1133)
	All-Electric	n/a	4800 (2266)	4000 (1888)	2400 (1133)
Oklahoma City	Gas/Electric	n/a	4800 (2266)	4000 (1888)	2400 (1133)
	All-Electric	n/a	4800 (2266)	4000 (1888)	2400 (1133)
Phoenix	Gas/Electric	n/a	4800 (2266)	4000 (1888)	2400 (1133)
	All-Electric	n/a	4800 (2266)	4000 (1888)	2400 (1133)

Prior research indicated the need for powered relief at economizer outdoor airflow rates of approximately 60% (of supply air volume) and greater, apparently due to a buildup of back pressure in the return half of the kitchen HVAC system. For each of the four baseline computer models, the second group of simulations assumes four levels of kitchen HVAC system economizer operation:

- a baseline case without an economizer;
- a system that allows a maximum of 60% outdoor air in the economizer mode;
- a system that allows a maximum of 80% outdoor air in the economizer mode and uses a small 0.5 hp (0.4 kW) relief fan; and
- a system that allows a maximum of 100% outdoor air in the economizer mode and uses a slightly larger 1.0 hp (0.7 kW) relief fan.

Effects of Decreasing Radiant Gain with Increasing Exhaust Flow. The third group of computer simulations investigates the impact of radiant gain upon kitchen heating and cooling requirements at different exhaust flow rates. Recent laboratory tests measuring the heat gain to space from commercial cooking equipment suggest that the rate of heat gain from an appliance is dependent upon both the type of exhaust hood and the exhaust airflow rate through the hood (Gordon et al 1994; Smith et al. 1995). Preliminary test results indicate that as the exhaust flow rate increases, the rate of appliance heat gain to the surrounding space decreases. For a given increase in exhaust flow rate, the reduction in heat gain to space varies according to hood and appliance type. This group of simulations is conducted in order to determine the impact of decreases in radiant gain to space at high exhaust flow rates on restaurant energy use, energy demand, and energy costs.

The project team identified a set of five computer simulations for the third group. The Oklahoma City location was selected as the weather site for the simulations for two reasons. First, the mean annual ambient temperature for Oklahoma City is nearly identical to that for Atlanta and Los Angeles. Second, Oklahoma City has a significant number of both heating and cooling degree-days, which allows the influence of changes in exhaust flow rate and radiant gain to space to be observed on both heating system and cooling system energy and economic performance.

A gas/electric restaurant model is used for these simulations so that performance impacts on the cooling system can be differentiated from those on the heating system. Wall-mounted canopy hoods are assumed in the model. Exhaust flow rates corresponding to "Above Code," "Code," "UL," and "Custom" rates (as described for the first simulation group) are modeled; however, instead of assuming a fixed rate of cookline heat gain for each exhaust rate, the rate of heat gain for the major appliances is allowed to decrease as exhaust flow rates increase, as shown in Table 4. The fifth simulation assumes that all of the radiant heat given off by the cookline is removed by the exhaust hoods, resulting in zero cookline heat gain to the kitchen space. This is an extreme case that establishes the theoretical limit on benefits that can be derived if all of the heat gain from appliances is removed by the exhaust hoods.

VENTILATION INVESTIGATIONS AND RESULTS

The results from the computer simulations are discussed below. First, results from the computer simulations performed to examine the effects of changes in exhaust flow rate upon restaurant energy use and energy costs are presented. This discussion is followed by presentation of the simulation results that examine the potential for economizer cooling and its impact on energy and costs. The results from the computer simulations that investigate the influence of radiant gain to space in conjunction with varying exhaust flow rate are presented last.

Results from Simulation of Varying Exhaust Rates

Figure 1 shows the annual energy for all buildings. The upper plots show the whole building energy use for the gas/electric buildings, and the lower plots show the same for the all-electric buildings.

In all locations except Los Angeles, increasing the hood exhaust flow rate above the baseline flow rate results in increased energy use. Likewise, decreasing the exhaust flow rate below the baseline flow rate results in decreased energy use. This is independent of both type of exhaust-hood employed and the fuel mix. Energy consumption in the buildings located in Akron, Atlanta, and Oklahoma City exhibits greater sensitivity to changes in exhaust flow rate than energy consumption by the buildings in other locations. These three locations have the highest number of heating degree-days, and the increased sensitivity is due to the contribution of energy consumed for space heating. The exhaust flow rates that yield the minimum annual whole building energy consumption in all locations except Los Angeles correspond to the lowest flow rates, 3150 cfm (3150 L/s) for canopy hoods and 2400 cfm (1133 L/s) for backshelf hoods.

The buildings located in Los Angeles experience the minimum annual whole building energy consumption at flow rates that correspond to those assigned to the baseline buildings, namely, at UL hood flow rates (4500 cfm [2124 L/s] for canopy hoods and 4000 cfm [1888 L/s] for backshelf hoods). Increasing or decreasing exhaust flow rate from the baseline flow rate results in increased annual energy use.

The mechanism for the behavior of the Los Angeles buildings is climate-driven. In the gas/electric buildings, gas

TABLE 4
Gas Cooking Appliance Heat Gains at Selected Exhaust Flow Rates

Appliance Type	Input Rating kBtu/h (kW)	Exhaust Flow Rate scfm/lf (L/s per m)	Heat Gain to Space Btu/h (W)
Griddle, 6 ft	162 (47)	400 (619)	8,799 (2,578)
		344 (533)	9,136 (2,677)
		250 (387)	9,701 (2,842)
		175 (271)	10,151 (2,974)
Atmospheric Fryers (3)	110 each	400 (619)	5,833 (1,709)
	(32 kW each)		
		344 (533)	6,262 (1,835)
		250 (387)	6,984 (2,046)
		175 (271)	7,559 (2,215)

energy consumption increases as exhaust flow rate increases, regardless of hood type; however, electrical energy consumption behaves much differently. In the case of buildings with canopy hoods, electrical energy use decreases as exhaust flow

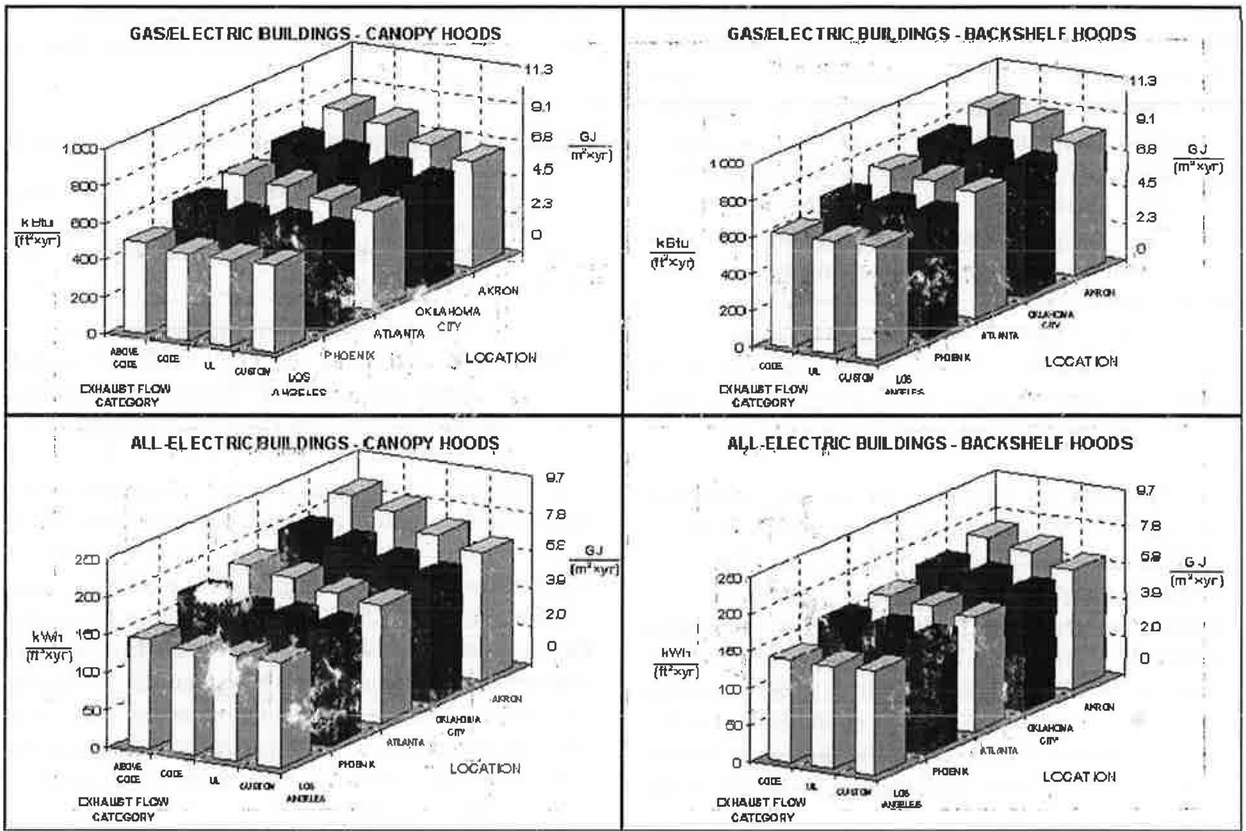

Figure 1 *Annual energy consumption results from simulation of varying exhaust rates.*

rate increases from the "Custom" rate to the "Code" rate. It then increases as the exhaust flow increases to the "Above Code" rate. In the case of buildings with backshelf hoods, electrical energy use decreases as exhaust flow rate increases from "Custom" rates to "Code" rates. In both hood cases, this behavior occurs because of the economizing effect that make-up air admission has over the mid-range exhaust flow rates. The combined effect of differing fuel consumption trends with variations in exhaust flow rate produces a minimum energy exhaust rate that corresponds to neither the minimum gas nor the minimum electricalenergy consumption flow rate.

Similar climate-related behavior is observed in the all-electric buildings in Los Angeles. In buildings with canopy hoods, electrical energy consumption varies with exhaust flow rate as it does in the gas/electric buildings. The same is true when comparing the variation in electrical energy use between the gas/electric and all-electric buildings with backshelf hoods. In the allelectric buildings, the changes in energy use resulting from changes in exhaust flow rate are not as dramatic as the changes in the gas/electric buildings.

Figure 2 shows the effects of varying exhaust flow rate on whole building annual peak electrical demand for all buildings. The changes in annual peak demand for the gas/electric buildings reflect the impact of variation in exhaust rates on cooling performance. The gas/electric buildings set their annual peak demand in the summer, regardless of exhaust rate or location. In contrast, the changes in annual peak demand arising from variation in exhaust rates are generally more dramatic in the all-electric buildings than the demand changes in the gas/electric buildings. The principal reason for this difference is that a majority of the all-electric buildings set their annual peak demand during the winter. This is true for all locations except Los Angeles and Phoenix, the two climates with the smallest heating requirements.

It is important to note that there are two locations where an all-electric building experiences a shift in the season during which their peak demand is set. Both buildings with canopy and backshelf hoods located in Atlanta set their annual peak demand during the winter at the higher exhaust flow rates. At the lowest flow rates for each hood type, the month in which the annual peak demand is set shifts from a winter month to a summer month. A similar event occurs in the all-electric building with canopy hoods located in Los Angeles, except the change of season occurs at the highest exhaust flow rate. In this case, at lower flow rates, the building sets its annual peak demand during the summer, and at the highest flow rate, the peak is set during the winter.

Figure 3 shows the annual whole building energy costs for all buildings. The upper plots show the yearly energy costs for the gas/electric buildings, and the lower plots show the yearly energy costs for the all-electric buildings.

From the four plots in Figure 3 it is evident that for each of the four baseline building types in all of the locations, the exhaust rate that results in minimum annual energy costs

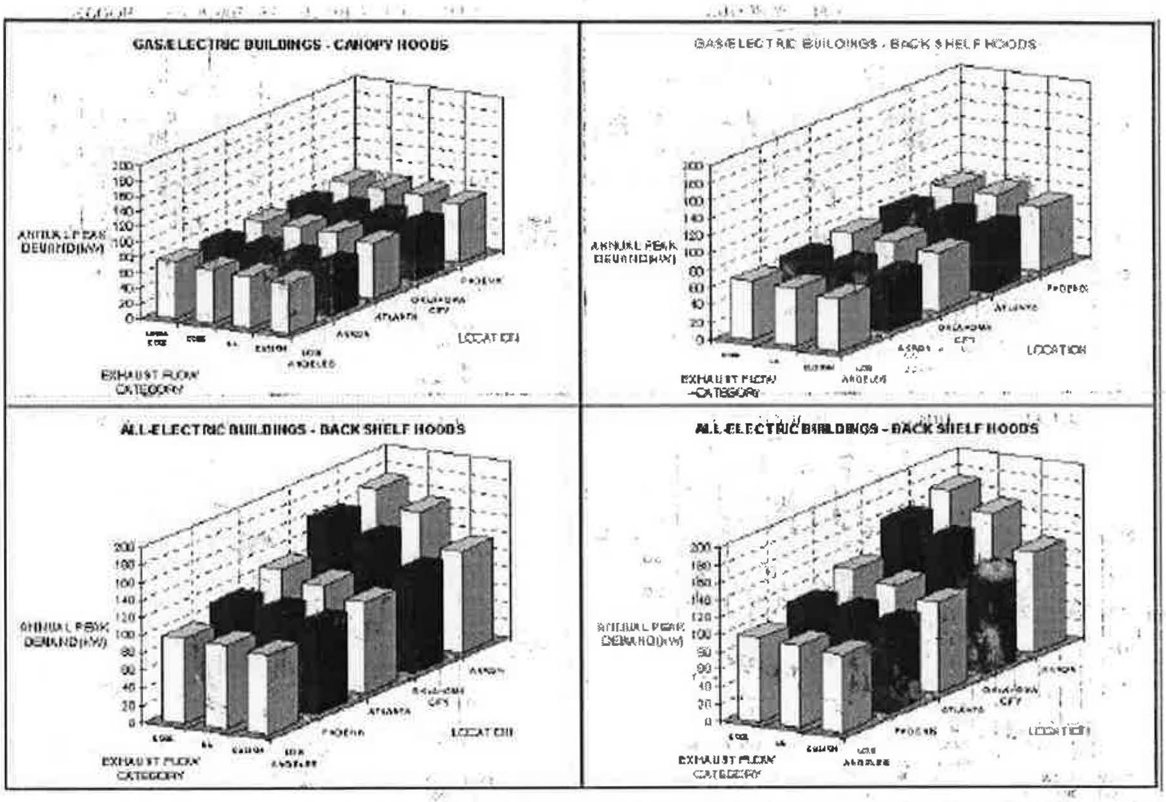

Figure 2 *Annual peak electrical demand results from simulation of varying exhaust rates.*

corresponds to the lowest of the exhaust rates simulated. A comparison of Figures 1 and 3 shows that in all locations except Los Angeles, the exhaust flow rate that results in the minimum energy consumption also results in the minimum energy cost. The differences in utility rate structure from one location to another are responsible for the reordering of the locations in Figure 3. The sensitivity shown in the plots of annual energy consumption for the gas/electric buildings is not preserved in the plots showing annual energy costs because electric energy and demand costs dominate the buildings' total annual energy cost in all locations.

With respect to annual energy costs, the buildings located in Los Angeles behave the same as the buildings in the other locations. In Los Angeles, the minimum energy cost is achieved at the lowest flow rates; however, the exhaust rates that yielded the minimum energy costs are not the exhaust rates that yielded the minimum energy consumption.

Results from Simulation of Economizer Cooling Strategies

The results from the economizer simulations were analyzed in two ways. First, kitchen HVAC cooling energy was plotted against ambient temperature and regression lines were fitted to represent cooling energy as a function of ambient temperature. Second, results were analyzed and compared by building type and location to identify the minimum energy kitchen HVAC system configuration and the energy and cost savings potential for each of the locations selected for this study.

The potential benefit of using economizers was investigated by comparing the model kitchen HVAC system performance with and without economizer. Figures 4 and 5 show typical plots of the regression of electric energy consumption against outside air temperature for the base case (UL-listed hoods and no HVAC economizer) and the super ventilation case corresponding to 100% economizer with powered relief.

The regression lines are based on a full year of simulated data (8,760 hours). The general shape of the regression lines shows that the results from the computer simulations agree with the results from previous work with regard to the effective temperature range for the economizer (Horton et al. 1993). The economizer case shows that use of mechanical cooling with compressors can be avoided when the ambient air temperature is between about 52°F and 62°F (11°C - 17°C) in gas-electric quick service restaurants and about 53°F to 63°F (12°C - 17°C) in all-electric quick service restaurants. A crossover point occurs at approximately 80°F (27°C).

Table 5 shows annual electrical consumption for each baseline model at each location (rounded to the nearest 100). The maximum annual electrical consumption savings shown in Table 6 are relatively small fractions of the respective annual consumption figures. Economizer benefit was greater in the gas-electric buildings than in the all-electric buildings. For a given fuel type, the restaurants with backshelf hoods experienced greater energy savings from the inclusion of

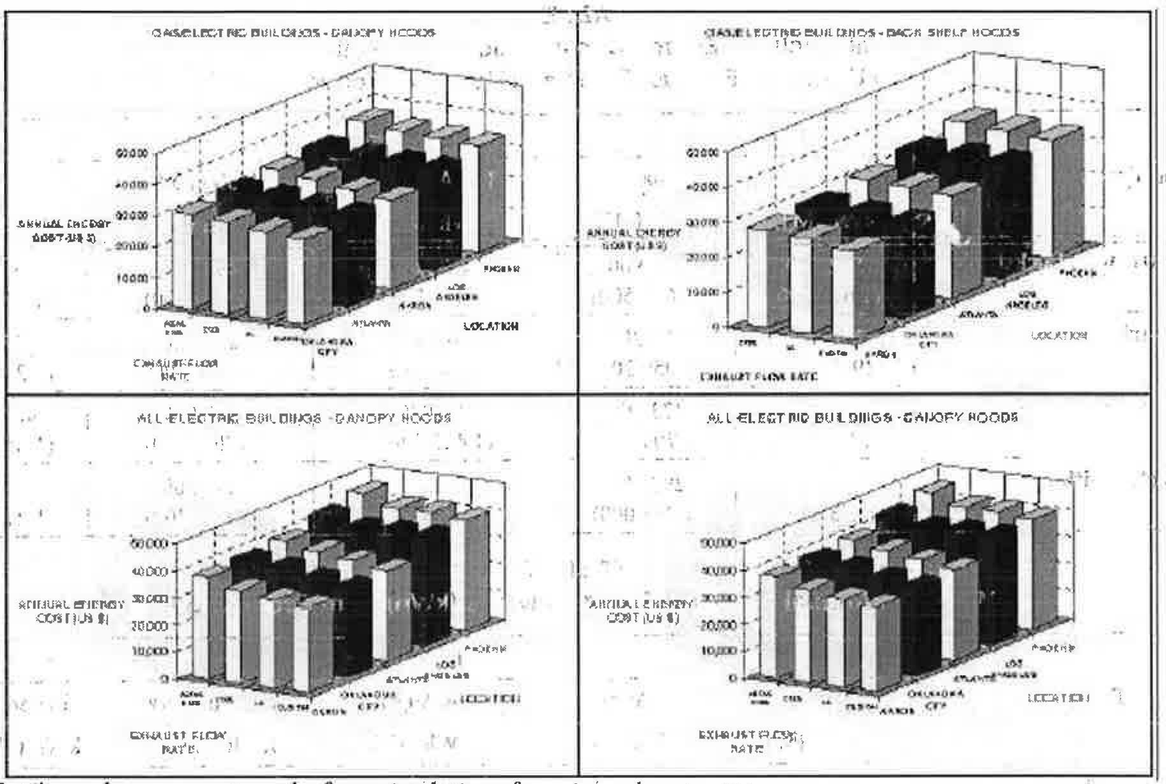

Figure 3 *Annual energy cost results from simulation of varying exhaust rates.*

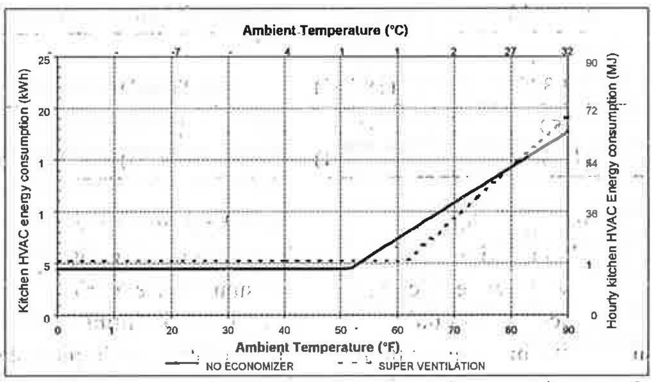

Figure 4 *Kitchen HVAC hourly cooling energy regressions for all-electric restaurant with backshelf hoods in Akron, Ohio.*

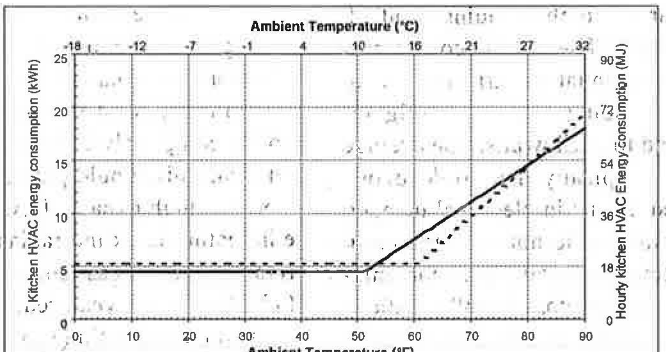

Figure 5 *Kitchen HVAC hourly cooling energy regressions for gas-electric restaurant with backshelf hoods in Akron, Ohio.*

TABLE 5
Annual Electrical Energy Consumption for Baseline Models with UL-Listed Exhaust Rates and No HVAC Economizers

Building Type	Location				
	Akron kWh (MJ)	Atlanta kWh (MJ)	Los Angeles kWh (MJ)	Oklahoma City kWh (MJ)	Phoenix kWh (MJ)
All-Electric with Backshelf Hoods	512 500 (1 673 500)	464 900 (1 673 500)	397 400 (1 430 700)	490 600 (1 766 100)	455 800 (1 641 000)
Gas-Electric with Backshelf Hoods	270 100 (972 500)	294 200 (1 059 200)	276 200 (994 200)	296 700 (1 068 300)	335 000 (1 206 000)
All-Electric with Canopy Hoods	529 600 (1 906 400)	473 500 (1 704 400)	396 000 (1 425 400)	502 000 (1 807 000)	460 300 (1 657 000)
Gas-Electric with Canopy Hoods	270 500 (973 900)	295 500 (1 064 000)	273 800 (985 800)	298 500 (1 074 400)	337 400 (1 214 600)

TABLE 6
Maximum Annual Electrical Energy Savings (kWh) from Economizers

Building Type	Location				
	Akron kWh (MJ)	Atlanta kWh (MJ)	Los Angeles kWh (MJ)	Oklahoma City kWh (MJ)	Phoenix kWh (MJ)
All-Electric with Backshelf Hoods	3876 (13 954)	5461 (19 660)	9743 (35 075)	4052 (14 587)	2329 (8384)
Gas-Electric with Backshelf Hoods	5245 (18 882)	7103 (25 571)	12 249 (44 096)	5636 (20 290)	5516 (19 858)
All-Electric with Canopy Hoods	3147 (11 329)	5101 (18 364)	6637 (23 893)	3724 (13 406)	2236 (8050)
Gas-Electric with Canopy Hoods	3793 (13 655)	5754 (20 714)	7676 (27 634)	4292 (15 451)	3944 (14 198)

economizers than did the buildings with canopy hoods. Table 6 shows the maximum annual kilowatt-hour savings achieved in each location by building type. The savings were calculated by subtracting the lowest consumption achieved in buildings with economizers from the similar baseline buildings without economizers. Los Angeles was the only location that exhibited substantial energy savings using economizers.

In general, the use of economizers provided only modest electrical energy savings for nearly all of the buildings studied, regardless of fuel mix or hood type. The use of economizers had very little impact on peak annual electrical demand. Demand reductions for all four building and hood configuration types were between zero and four kilowatts. The average reduction for a given location was typically on the order of one kilowatt or less. Substantial reductions in electrical demand were not expected since economizers do not operate during periods of heating or peak cooling. Consequently, economizers provided only small reductions in cost for all locations except Los Angeles.

Results from Simulation of Decreasing Radiant Gains

The results from the third group of simulations are shown in Figure 6. This set of four plots shows the changes in gas consumption, electrical energy consumption, electric demand, and whole building energy use predicted by the computer simulations. Each plot shows that as exhaust flow rates increase, so does energy consumption and electrical demand. Reductions in radiant heat gain appear to have little effect on building energy consumption at increased exhaust flow rates.

The single point shown on each plot of Figure 6 as the "zero gains" point presents an interesting special case. Reducing cookline radiant gains to zero and setting the exhaust flow rate to the highest level tests the case in which the exhaust hoods completely remove the cookline heat gain from the kitchen. This should produce the maximum possible savings from the high exhaust flow rate. As can be seen in Figure 6, eliminating cookline radiant gains at the high exhaust flow rate increased gas consumption by roughly 400 therms (42,200 MJ) per year, reduced electrical energy consumption by about 4,000 kWh (14,400 MJ) per year, and had no effect on electrical demand. The net result in terms of annual energy consumption was an increase of nearly 10,000 Btu/(y·ft^2) (114 MJ/y·m^2)).

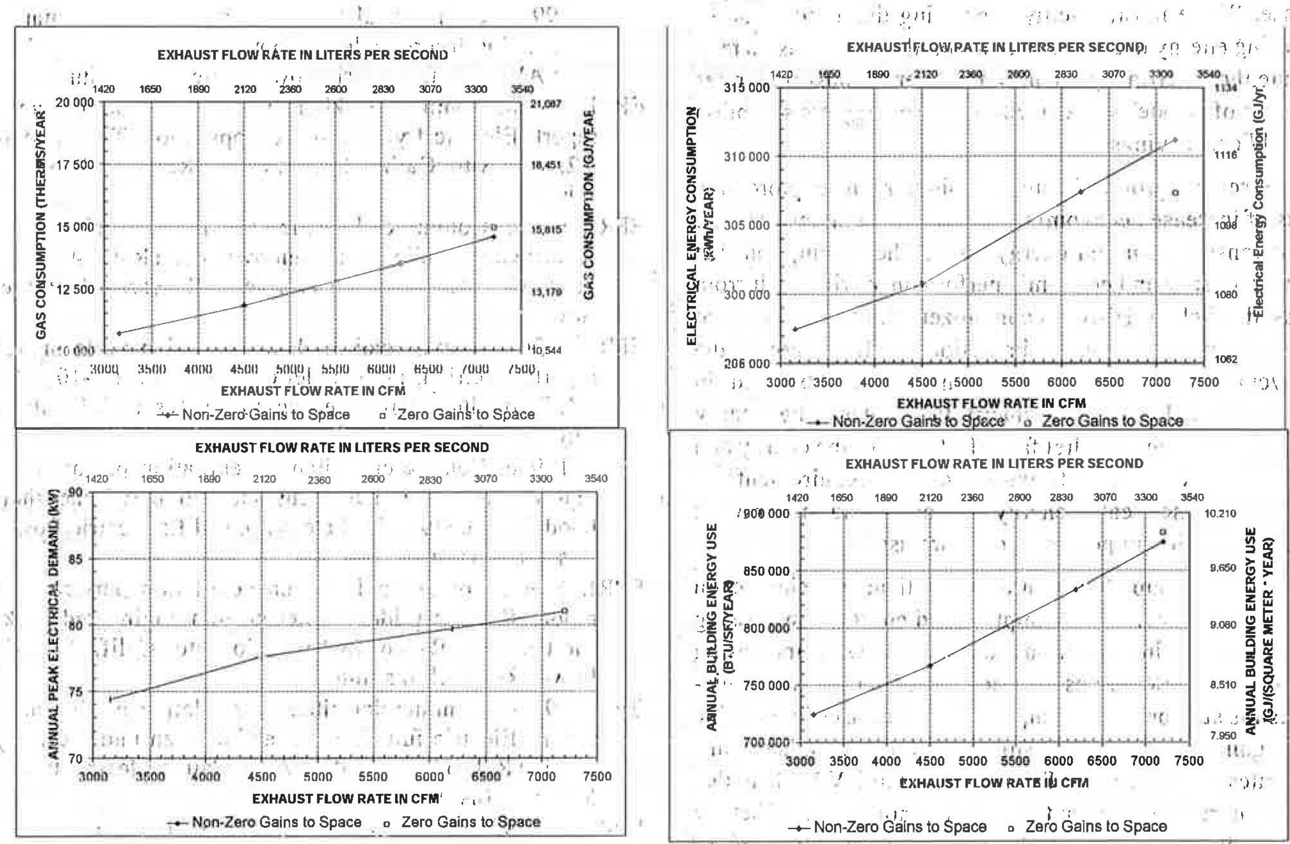

Figure 6 *Results from simulation of decreasing radiant gains.*

Figure 7 shows the changes in whole building energy costs computed from the simulation results. The plot shows that energy costs increase as exhaust flow rates increase, even when the hoods remove more of the cookline heat gain at higher airflow rates. As the "zero gains point" on the plot shows, even the complete removal of cookline heat does not offset the increased energy costs resulting from the high exhaust rate.

CONCLUSIONS

The first group of computer simulations investigated the effects of varying exhaust flow rates on kitchen HVAC energy consumption and costs. The simulation results presented above support the assumption that decreases in kitchen exhaust flow rates will result in improved energy and economic performance in quick service restaurants. Exhaust flow rates at "Code" values and above result in higher energy consumption and energy costs by virtue of the larger make-up air requirement and the attendant increases in ventilation load. This is true in all but the most moderate of climates.

In very moderate climates such as Los Angeles, make-up air admission requirements tend to reduce energy consumption and costs over the "UL-Listed" and "Code" exhaust flow rates. At the extremes—"Above Code" and "Custom-engineered" exhaust flow rates—energy consumption tends to

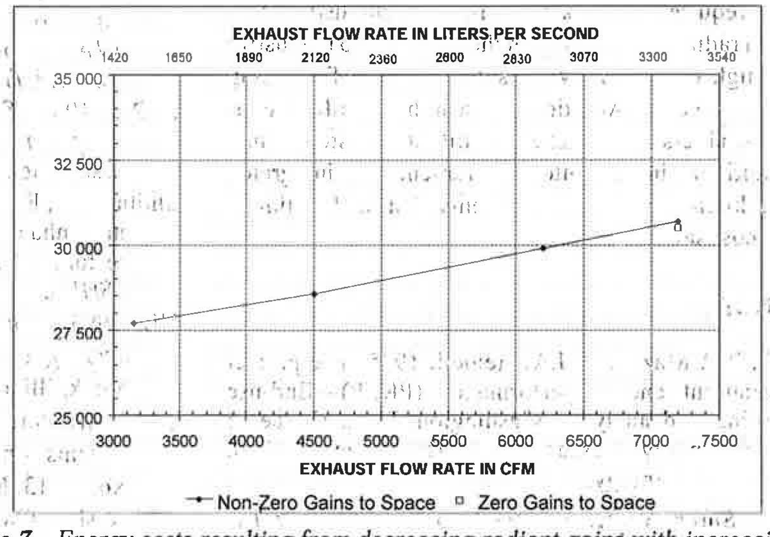

Figure 7 *Energy costs resulting from decreasing radiant gains with increasing flow rate*

increase. While not explicitly supporting the assumption of decreasing energy use and cost, the simulation results do tend to refute the assertion that constant higher exhaust flow rates (in excess of "Code" values) will reduce energy consumption and yield cost savings.

The second group of computer simulations explored the effects of increased economizer flow rates on kitchen HVAC energy consumption and energy costs. The assumption that improved energy and economic performance will result from increased kitchen HVAC economizer flow rates is also supported by the computer simulation results, although the improvements are less dramatic than those observed in conjunction with reduced exhaust flow rates. The energy required to operate the relief fan offsets part of the energy and cost savings in climates with extreme cooling requirements. In such climates, the greatest energy and cost savings occur when economizers without powered relief are used.

The third group of simulations investigated the impact on kitchen HVAC energy consumption and energy costs arising from radiant gain reductions that occur with increasing kitchen exhaust flow rates. As shown previously, the simulation results support the assumption that decreases in cookline radiant gain to space are not sufficient to make higher ventilation rates cost-effective. The reductions in HVAC load that stem from reduced radiant gains are more than offset by increases in ventilation loads due to larger make-up air requirements. Even if the exhaust hoods could remove all of the cookline heat gain at higher flow rates, when compared to lower flow rates the net result is still an increase in energy consumption and costs.

Overall, the computer simulation results indicate that decreases, not increases, in kitchen exhaust flow rates will produce energy and economic savings in most quick service restaurants. Furthermore, the simulation results show that high flow rate HVAC economizers provide significant energy and economic savings only in climates with minimal heating and cooling requirements. Last, the results demonstrate that reductions in radiant heat gain from the cookline do not justify the use of high exhaust flow rates as a means of reducing HVAC cooling costs. Additional research regarding two-speed or variable-speed kitchen ventilation systems may provide additional insight into requirements for integrated HVAC and kitchen ventilation systems that will optimize energy and cost savings.

REFERENCES

Claar, C.N., R.P. Mazzuchi, J.A. Heidell. 1985. The project on restaurant energy performance (PREP)—End-use monitoring and analysis. Washington, D.C.: Office of Building Energy Research and Development, U.S. Department of Energy.

Claar, C., V. Smith, W. Krill. 1995. Too much hot air? *The Consultant* Vol 28(2). Louisville, Ky: Foodservice Consultants Society International.

EPRI. 1996a. Commercial kitchen ventilation performance report: Gas fryers under canopy hood. TR-106493-V1. Palo Alto, Calif.: Electric Power Research Institute.

EPRI. 1996b. Commercial kitchen ventilation performance report: Electric fryers under canopy hood. TR-106493-V2. Palo Alto, Calif.: Electric Power Research Institute, July.

EPRI. 1996c. Commercial kitchen ventilation performance report: Gas griddle under canopy hood. TR-106493-V3. Palo Alto, Calif.: Electric Power Research Institute, July.

EPRI. 1996d. Commercial kitchen ventilation performance report: Electric griddle under canopy hood. TR-106493-V4. Palo Alto, Calif.: Electric Power Research Institute, July.

EPRI. 1996e. Commercial kitchen ventilation performance report: Gas griddle under custom engineered backshelf hood. TR-106493-V5. Palo Alto, Calif.: Electric Power Research Institute.

EPRI. 1996f. Commercial kitchen ventilation performance report: Electric griddle under custom engineered backshelf hood. TR-106493-V6. Palo Alto, Calif.: Electric Power Research Institute.

EPRI. 1996g. Commercial kitchen ventilation performance report: Electric full size convection oven under canopy hood. TR-106493-V7. Palo Alto, Calif.: Electric Power Research Institute.

EPRI. 1996h. Commercial kitchen ventilation performance report: Gas convection oven under canopy hood. TR-106493-V8. Palo Alto, Calif.: Electric Power Research Institute.

Fisher, D.R. Increasing Profits by Optimizing and Controlling Kitchen Exhaust Ventilation. A seminar program developed for Energy, Mines, and Resources, Canada. 1986.

Gordon, E.B., D.J. Horton, and F.A. Parvin. 1994. Development and application of a standard test method for the performance of exhaust hoods with commercial cooking appliances. *ASHRAE Transactions* 100(2).

Horton, D.J., J.N. Knapp, E.J. Ladewski. 1993. Combined impact of ventilation rates and internal heat gains on HVAC operating costs in commercial kitchens. *ASHRAE Transactions* 99(2).

NFPA. 1996. *Standard 96, Vapor removal from cooking equipment.* Quincy, Mass.: National Fire Protection Association.

Smith, V.A., R.T. Swierczyna, C.N. Claar. 1995. Application and enhancement of the standard test method for the performance of commercial kitchen ventilation systems. *ASHRAE Transactions* 101(2).

UL. 1995. *Standard 710, Exhaust hoods for commercial cooking equipment.* Underwriter's Laboratory. Northbrook, Illinois: National Fire Protection Association.

UW (University of Wisconsin). 1990. TRNSYS - A transient systems simulation program - User's manual. Report No. 38-13. Madison, WI: Solar Energy Laboratory, University of Wisconsin.

BIBLIOGRAPHY

EPRI. 1996i. Minimum energy ventilation for fast food restaurant kitchens. TR-106671. Electric Power Research Institute, September.

Soling, S.P., and J. Knapp. 1985. Laboratory design of energy efficient exhaust hoods. *ASHRAE Transactions* 91(1B): 383-392.

Talbert, S.G., L.J. Flanigan, and J.A. Eibling. 1973. An experimental study of ventilation rates and internal heat gains on HVAC operating costs in commercial kitchens. *ASHRAE Transactions* 79(1): 34-47.