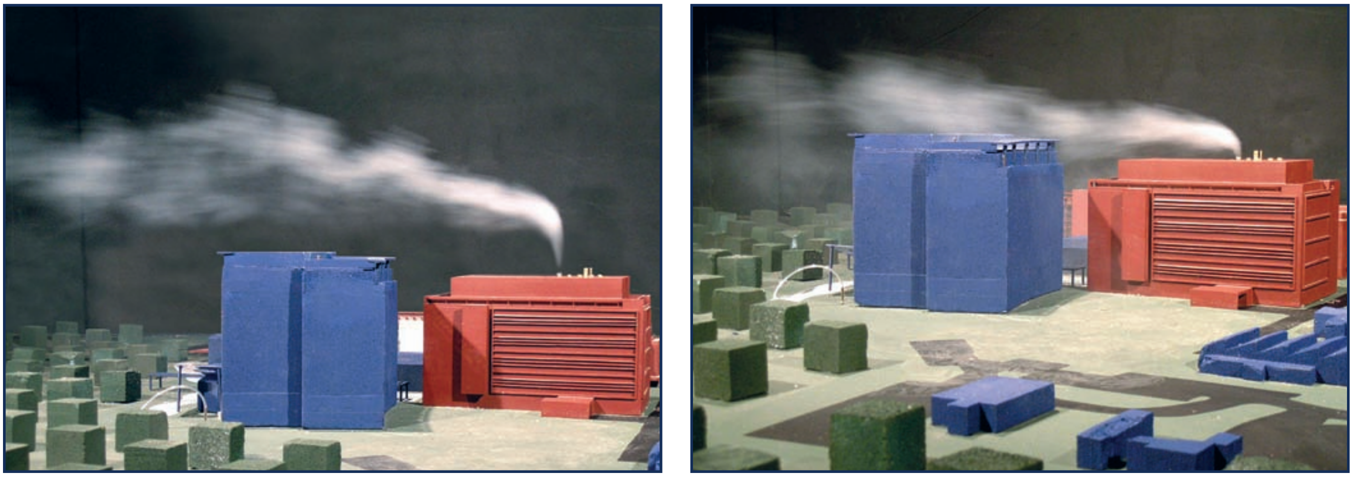

Figure 1: Relationship of stack and plume heights. **Left:** good air quality; potential wasted energy. **Right:** bad air quality; lower energy use.

Saving Energy in Lab Exhaust Systems

By **John J. Carter,** Member ASHRAE; **Brad C. Cochran,** Member ASHRAE; and **Jeff D. Reifschneider**

Studies have shown that there is a direct relationship between indoor air quality and the health and productivity of building occupants.[1,2,3] Historically, the focus of indoor air quality has addressed emission sources emanating from within the building, for example, techniques to limit or eliminate off-gassing of finish materials such as adhesives, carpet and furniture. Additionally, for laboratories, the "as-manufactured" and "as-installed" containment recommendations for fume hoods are intended to ensure that the worker is not exposed to toxic chemicals.[4] However, a critical aspect of indoor air quality is external emission sources that may be re-ingested into the building through closed-circuiting between nearby exhaust stacks and a building's air intakes.

In the past (and even now, it's true), particularly for laboratories, the desire was to keep exhaust stacks as short as possible to limit the visual impact. This strategy often results in systems with higher exit velocities and/or larger volume flow rates than would otherwise be required. This is because concentration levels of emitted contaminants typically are inversely proportional to plume rise: larger plume rise equals lower concentrations (better air quality). Plume rise is obtained through physical stack height and the vertical mo-

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mentum of the plume. The shorter the stack, the greater the vertical momentum (i.e., fan power) that is required to achieve the appropriate plume height (*Figure 1* and Equation 1).

$$\begin{aligned}\text{Plume Rise} &= \text{Physical Stack Height} \\ &\quad + f(\text{Volume Flow} \times \text{Velocity})^{1/3} \\ &= \text{Physical Stack Height} + f(\text{Constant} \times \text{Fan Power}^{n})\end{aligned} \tag{1}$$

This high energy demand associated with the shorter stack heights contradicts the current emphasis on energy conservation in sustainable laboratory design.

The country is focused on methods to save energy with particular focus on turning off lights, riding a bicycle, etc., yet there is a huge amount of unrealized energy savings available in our nation's research and teaching laboratories. A typical laboratory consumes up to 10 times the energy per square foot of an office building, while specialized laboratories may consume up to 100 times more energy.[5] Due to the requirements for high air change rates of 100% outside air, the ventilation system uses a high percentage of this energy, often up to 80%.[6,7] The ventilation of a laboratory can be broken down into three systems; the outside air supply system, conditioning (temperature, humidity, filtration, etc.), and the exhaust system.

The supply air and conditioning systems account for approximately 60%[6,7] of the ventilation system energy consumption and have been the focus of laboratory designers for the past several decades. Variable air volume (VAV) air-handling units have become typical in laboratory design to minimize airflow to match the building's ventilation demands, which can vary throughout the day, depending upon the laboratory occupancy, the fume hood activity (when VAV fume hoods are installed) and, in some cases, the quality of the indoor air. Heat recovery systems also are used regularly, particularly in northern climates, to reduce the energy consumption of the air-conditioning systems.

Energy saving strategies often overlook the exhaust system, even though it accounts for the other 40% of the ventilation system's energy consumption and about 30% of the laboratory building's total energy consumption.[6,7]

Traditionally, laboratories have been designed such that the exhaust system must operate at full load conditions 24 hours a day, 365 days a year—a constant volume (CV) exhaust. Full load often meant the minimum flows to achieve acceptable air quality, which in some cases could be significantly greater than the ventilation requirement, particularly for short stacks. In addition, this air quality setpoint was based on the worst-case wind condition, which typically occurs only for a small fraction of total hours each year *and* the assumption that the exhaust stream actually contains the worst-case contaminants.

We will discuss three strategies that can be used, in part or in whole, during the design of a new laboratory or renovation of an existing laboratory. These strategies can *safely* reduce the energy consumption of the exhaust system by about 50% compared to a typical CV system, which equates to a 15% reduction in the laboratory's total energy use. We will also present some case study results.

Strategy 1: Passive Variable Volume Exhaust

Using state-of-the-art engineering techniques, controls, and exhaust fans, exhaust ventilation systems can optimize energy consumption by applying VAV technology to the exhaust side. A VAV system allows the airflow in the exhaust ventilation system to match, or nearly match, the ventilation airflow requirements of the building; however, the VAV system still must be designed so that it does not compromise the air quality at nearby sensitive locations (i.e., air intakes, balconies, etc.). Building exhaust may adversely affect these areas if existing CV systems are blindly converted to VAV systems without a clear understanding of how the system will perform at the lower volume flow rates. Safety must always be the key concern.

Remember, an exhaust system not only removes contaminated laboratory air from the building, but it also serves to discharge the exhaust so that fumes do not impact sensitive locations. This is achieved through the proper combination of stack height and exhaust discharge momentum. So how do you determine the proper combination of stack height and exhaust momentum? You do it through an engineering technique called exhaust dispersion modeling.

The preferred state-of-the-art method for conducting an exhaust dispersion study is through the use of physical modeling in a boundary-layer wind tunnel. However, mathematical techniques, similar to those in the *ASHRAE Handbook—HVAC Applications,*[8] can be used as an initial screening tool to estimate potential turndown capacity, especially for large exhaust systems in simple surrounding environments. Additional information on conducting exhaust dispersion studies can be found in the July 2005 *ASHRAE Journal*[9] and in the Laboratories for the 21st Century's best practice guideline.[10]

In a passive VAV system, the exhaust flow is based on the lowest of the minimum air quality setpoint, or the building's ventilation demand. The minimum air quality setpoint is defined as the minimum flow/exit velocity/stack height needed to provide acceptable air quality. Acceptable air quality in this situation is typically defined using the ASHRAE lab exhaust performance criterion,[11] the Z9.5[4] fume hood containment criteria, or through a review of the chemical inventory for the lab. For a 50% turndown ratio, which can typically be achieved during unoccupied hours, this might involve using a taller stack than the architect typically prefers (Tall stacks are green!), or optimizing the placement of air intakes to minimize reentrainment of the exhaust. Typically, 1.5 m or 3 m (5 ft or 10 ft) increases in stack height have been effective. From a controls standpoint, this is likely the simplest system to use, particularly for retrofit of existing laboratories (*Figure 2*).

Strategy 2: Active VAV with Anemometer

If the passive VAV system does not lower the air quality exhaust setpoint equal to, or lower than, the building ventilation demand, further optimization is available through knowing the current wind conditions at the stack through use of an on-site anemometer. Recall that the passive VAV setpoint assumed the worst-case condition—a relatively low-frequency event.

In this strategy (*Figure 3*), a local anemometer is connected to the building automation system (BAS) and the minimum required exhaust flow rate is varied based on current wind conditions (direction and speed). When the wind conditions are other than worst-case, the exhaust system may be turned down to more closely match the building demand. Essentially, the air quality minimum setpoint is specified for each wind direction/speed combination. This usually results in air quality setpoints well below building demand for many wind conditions.

This strategy requires physical exhaust dispersion modeling in a wind tunnel. Minimum air quality setpoints as a function of wind direction (WD) and wind speed (WS) require concentration predictions at all sensitive locations (receptors) for all wind directions, wind speeds, stack heights and exhaust flow parameters. Typically, initial testing and discussion is conducted to identify an acceptable stack height. Subsequent testing is conducted for all wind directions and speeds using a fixed stack diameter to produce concentrations for each stack/receptor combination for all combinations of wind direction, wind speed and volume flow rate (*Figure 4*).

Similar data for all receptors is then compiled into either a single lookup table or a series of wind-direction-specific polynomial equations for the BAS. *Table 1* presents a lookup table of the air quality setpoint as percent of design flow. Note that for most directions, the air quality setpoint is essentially zero (exhaust flow can match building demand exactly), although a few conditions require 80% of the design flow.

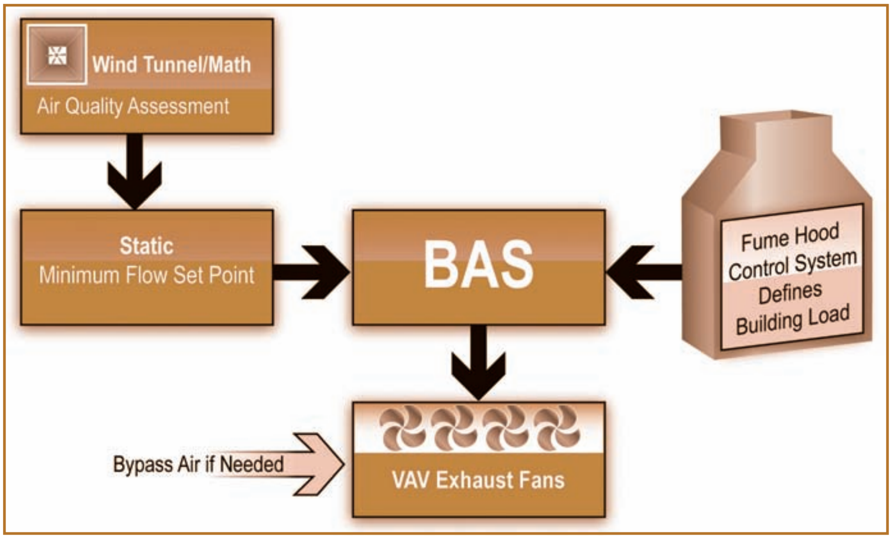

Figure 2: Passive variable air volume.

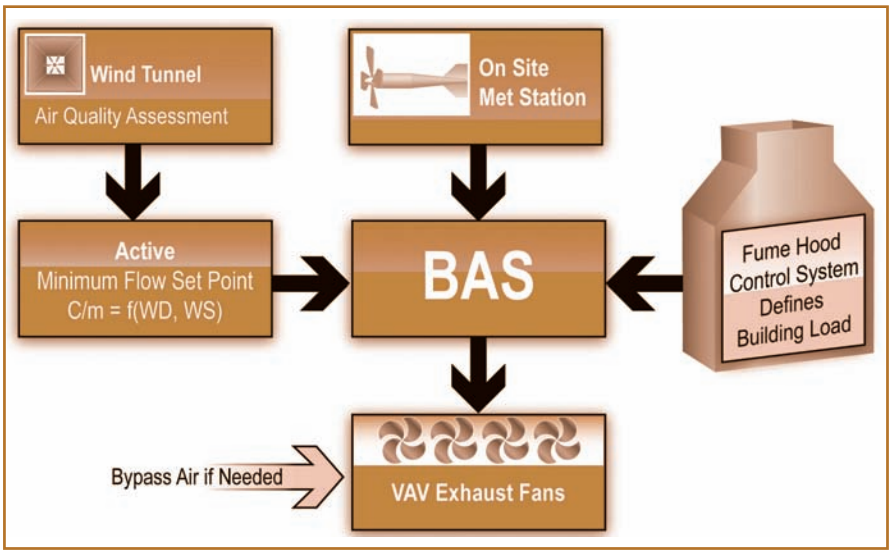

Figure 3: Active variable air volume.

Strategy 3: Active VAV with Chemical Monitor

An alternative to monitoring the local wind conditions could be to monitor the contents of the exhaust stream. When the monitor does not detect any adverse chemicals in the exhaust stream, the exhaust system is allowed to operate at a reduced volume flow rate. While there may be an increase in the plume concentrations at the nearby air intakes, air quality will not degrade since the exhaust plume is essentially "clean."

The usual assumption is that a contaminant is present in the exhaust stream, and the exhaust design is specified to achieve acceptable air quality through either mathematical or wind tunnel exhaust dispersion analysis. If a monitoring system is used, the "normal" mode would be to establish a minimum air quality exhaust setpoint that allows higher than normal plume impact. Plume impact would still be limited, just to a less conservative criterion than otherwise might be allowed. If contaminants are detected in the exhaust stream, the exhaust flow would be increased to achieve a more stringent criterion. *Figure 5* shows 1500 µg/m^3 per g/s as an example of the "normal" allowable impact and 400 µg/m^3 per g/s as the criterion when a contaminant is detected.

Data collected at operating research laboratories with air quality monitors in the exhaust manifold indicate that emission events that would trigger the higher volume flow rate typically occur no more than one hour per month (12 hours per year; 0.1% of the time).[12] This means that a typical system is able to operate at the lower air quality setpoint more than 99% of the time, resulting in significant energy savings.

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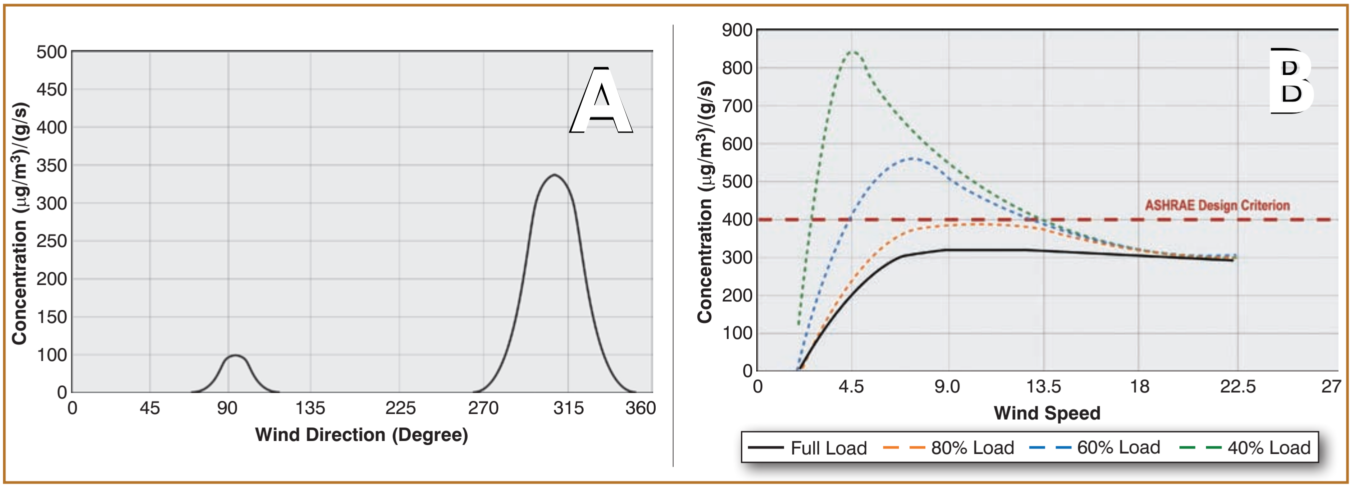

Figure 4a (left): Normalized concentration distribution (C/m) versus wind direction. **Figure 4b (right):** Normalized concentration distributions (C/m) versus wind speeds for various fan loads.

Case Study 1: Cedars-Sinai Medical Center

This new laboratory was planned with two sets of four exhaust fans operating in an n+1 redundant configuration with flows as indicated in *Table 2*.

Wind tunnel testing resulted in an air quality setpoint of 14 158 L/s (30,000 cfm) per fan being acceptable for all wind conditions. Note that this is slightly greater than the 12 270 L/s (26,000 cfm) that would be indicated if the minimum flow was split evenly among three fans. However, it is still substantially below the building minimum flow of 36 811 L/s (78,000 cfm). Based on this result, three schemes were analyzed to determine the optimum configuration of exhaust system complexity and energy consumption.

Wind Direction (Degrees)	Anemometer Wind Speed (mph)													
	(0–4)	(4–7)	(7–9)	(9–11)	(11–13)	(13–16)	(16–18)	(18–20)	(20–22)	(22–25)	(25–27)	(27–29)	(29–31)	(>31)
0	–	–	–	–	–	–	–	–	–	–	–	–	–	–
235	–	–	–	–	–	–	–	–	–	–	–	–	–	–
240	13%	–	–	–	–	–	–	–	–	–	–	–	–	–
245	21%	15%	–	–	–	–	–	–	–	–	–	–	–	–
250	25%	27%	18%	13%	11%	–	–	–	–	–	–	–	–	–
255	26%	30%	24%	16%	13%	11%	11%	–	–	–	–	–	–	–
260	25%	27%	18%	13%	11%	–	–	–	–	–	–	–	–	–
265	21%	19%	18%	17%	17%	16%	16%	16%	16%	15%	15%	15%	15%	15%
270	25%	26%	25%	24%	23%	22%	22%	22%	21%	21%	21%	21%	21%	20%
275	27%	28%	27%	26%	26%	25%	24%	24%	24%	23%	23%	23%	23%	23%
280	33%	27%	26%	26%	25%	25%	25%	24%	24%	24%	24%	24%	24%	24%
285	48%	37%	35%	33%	32%	32%	31%	31%	31%	30%	30%	30%	30%	30%
290	60%	58%	57%	55%	53%	51%	50%	49%	49%	48%	47%	47%	47%	46%
295	64%	73%	75%	73%	71%	69%	68%	66%	65%	64%	63%	62%	62%	61%
300	68%	78%	81%	81%	79%	76%	75%	73%	71%	70%	69%	68%	68%	67%
305	65%	75%	80%	78%	74%	69%	68%	66%	65%	64%	63%	62%	62%	61%
310	73%	73%	76%	73%	69%	64%	61%	58%	56%	54%	53%	52%	51%	50%
315	65%	64%	64%	60%	55%	52%	49%	47%	46%	45%	44%	43%	42%	42%
320	49%	50%	45%	41%	38%	37%	35%	34%	33%	33%	32%	32%	31%	31%
325	35%	31%	27%	25%	24%	23%	23%	22%	22%	22%	21%	21%	21%	21%
330	22%	–	–	–	–	–	–	–	–	–	–	–	–	–
335	–	–	–	–	–	–	–	–	–	–	–	–	–	–
340	–	–	–	–	–	–	–	–	–	–	–	–	–	–
345	–	–	–	–	–	–	–	–	–	–	–	–	–	–
350	–	–	–	–	–	–	–	–	–	–	–	–	–	–
355	–	–	–	–	–	–	–	–	–	–	–	–	–	–

Table 1: Minimum fan load percentages versus anemometer reading (BAS lookup table).

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	Peak Demand Flow		Minimum Ventilation Flow	
	Total	Per Fan	Total	Per Fan
Initial Design (CV)	156,000 cfm	52,000 cfm	78,000	26,000
VAV with Bypass		52,000 cfm		26,000 + 4,000 Bypass
VAV with Staging	Three to Four Fans Variable from 30,000 cfm to 39,000 cfm Each		Two Fans Variable from 39,000 cfm to 45,000 cfm Each	
VAV with Staging and Anemometer	Four Fans Variable from 23,400 cfm to 39,000 cfm Each		Three Fans Variable from 26,000 cfm to 30,000 cfm Each	

Table 2: Cedars-Sinai exhaust flows.

The simplest system configuration (*Figure 6a*) split the flow equally among the four fans using VAV with bypass air to maintain the 14 158 L/s (30,000 cfm) minimum flow for each fan. Note that bypass air is unconditioned air brought into the exhaust through a damper near the fan inlet on the roof.

The second system configuration (*Figure 6b*) used VAV and staging among the four fans. The system would operate with two fans at the minimum building demand conditions. Adding fans as needed to match demand. Note that all four fans would be used to keep the flow (and exit velocity) per fan as low as possible (but still equal to or greater than the air quality setpoint), providing net energy savings over running three fans at full flow and velocity.

The third system configuration (*Figure 6c*) used VAV, staging and anemometer input to dynamically control exhaust selecting the higher of the air quality setpoint and building demand. In this case, fan flows as low as 11 043 L/s (23,400 cfm) are allowed under certain wind conditions.

Since this building was not constructed or occupied, an assumed building load profile was used to predict likely electricity cost savings of the three configurations versus a constant volume system at the full design flow. Manufacturer power curves were used in conjunction with TMY data (National Renewable Energy Laboratory's Typical Meteorological Year) and a $0.10 per kWh rate. *Table 3* shows a respectable annual cost savings of about $100,000 for the VAV systems without an anemometer, increasing to about $120,000 per year for the system with the anemometer.

Case Study 2: University of California Santa Barbara

Wind tunnel studies were conducted for three laboratory buildings at the University of California Santa Barbara: California NanoSystems Institute (CSNI); Marine Sciences Research Building (MSRB);

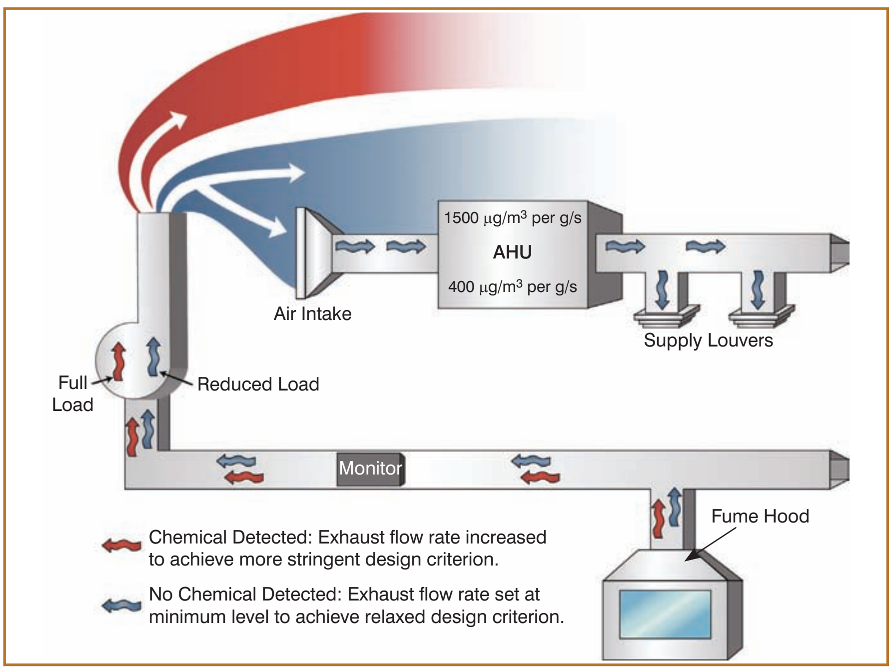

Figure 5: Flow control with exhaust stream monitor.

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and Bren Hall. The minimum air quality setpoints, both wind and without anemometer control, were defined as discussed earlier.

Since these buildings have been occupied for some time, actual building load profiles were available for predicting electricity costs. In addition, Southern Cal Edison offered a rebate or $0.24 per kWh saved for the first year, limited to the cost to implement the system.

Table 4 shows that CNSI was already operating at about 60% of the planned full load energy consumption prior to conducting the wind tunnel tests. However, under those flow conditions, the air quality design criterion was predicted to be exceeded about 7% of the time. By implementing a staged VAV system with anemometer control, annual energy consumption could be reduced to about 10% of the existing consumption—all while maintaining acceptable air quality. This will result in about $81,000 in annual electricity savings versus the existing operation, with an additional bonus of about $90,000 from Southern Cal Edison ($0.24 per saved kWh limited by the cost to upgrade)—roughly $80,000 savings the first year after everything is paid for.

Table 4 shows more modest, but still respectable, performance for the MSRB. Energy consumption could be reduced to about 42% of existing, with an annual electricity savings of about $14,000. Bren Hall's existing energy consumption, and total exhaust flow rate, is at lower than the other two buildings, resulting in much more modest cost savings.

Based on the studies we have conducted to date, it appears that savings in the range of $1.06 per L/s ($0.50 per cfm) can be achieved for many laboratory exhaust systems. In general, it takes systems on the order of 9 439 L/s to 14 158 L/s (20,000 to 30,000 cfm) before significant turndown starts to become available. It is likely that smaller systems could also achieve turndown with favorable surrounding building environments, but annual costs savings obviously decrease dramatically as total exhaust flow is reduced.

Summary

We have shown that using advanced control technologies, including variable air volume exhaust systems, local wind measurements and building automation systems, in conjunction with state-of-the-art exhaust dispersion modeling techniques can result

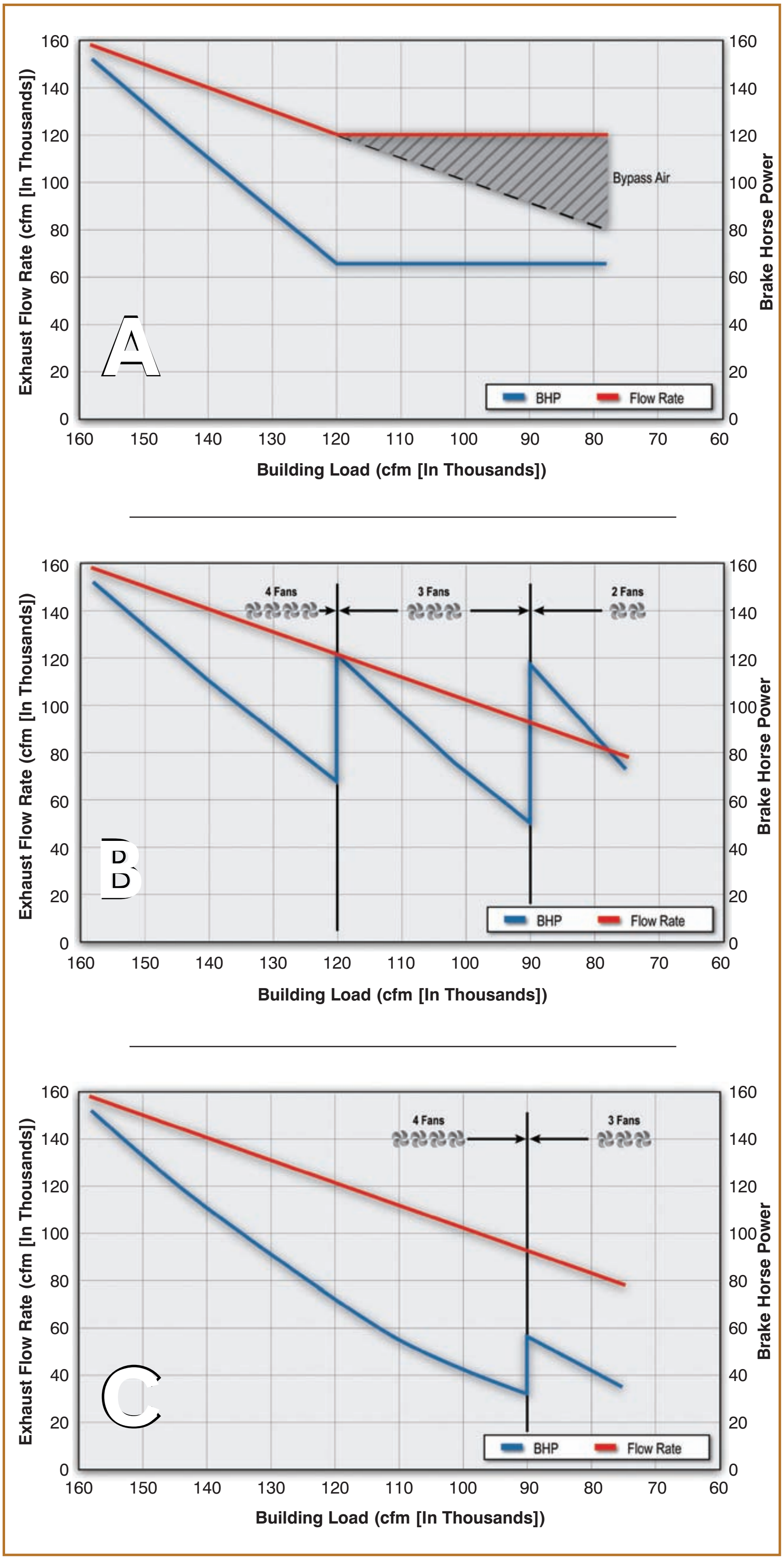

Figure 6a (top): Fan load and energy profiles for VAV+Bypass. **Figure 6b (center):** Staged VAV. **Figure 6c (bottom):** Staged VAV with anemometer.

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in significantly lower energy consumption, and related electricity costs, for existing and planned laboratories.

Note: Portions of this article appeared in the 2009 Fall AMCA inmotion magazine.

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Exhaust Configuration	Annual Energy Consumption (kWh)	Savings vs. CV	
		Energy (kWh)	($)
Constant Volume (CV)	1,934		
VAV+Bypass	962	972	$97,165
VAV+Staged	1,001	933	$93,241
VAV+Staged+Anemometer	709	1,225	$122,494

Table 3: Case Study 1—Cedars-Sinai energy consumption and cost savings.

Exhaust Configuration	Percent Time Air Quality Criterion Is Not Met	Annual Energy Consumption (kWh)	Savings vs. Existing		SCE Rebate ($)	First Year Return ($)	Estimated Upgrade Cost ($)	Annual O&M Cost ($)	Estimated Payback Period (Years)
			Energy (kWh)	($)					
CNSI with EF-1,3,4,5 Stack Heights at 15.2 ft and EF-2 at 22.8 ft									
Full Load (As Designed)	–	1,839,505	–	–	–	–	–		–
Existing Conditions	7.39%	904,075	–	–	–	–	–		–
Staged Operation With WD/WS Monitor	0%	94,814	809,260	$80,926	$90,400	$171,326	$90,400	$1,000	0.3
MSRB with Stack Heights at 25.8 ft									
Full Load (As Designed)	–	721,823	–	–	–	–	–		–
Existing Conditions	0%	248,229	–	–	–	–	–		–
Staged Operation With WD/WS Monitor	0%	104,434	143,795	$14,380	$62,800	$48,890	$62,800	$1,000	2.1
Bren Hall with Stack Heights at 24.2 ft									
Existing Conditions	0%	159,053	–	–	–	–	–		–
Staged Operation With WD/WS Monitor	0%	61,875	97,268	$9,727	$23,344	$33,071	$58,300	$1,000	4.0

Table 4: Case Study 2—CNSI energy consumption and cost savings.