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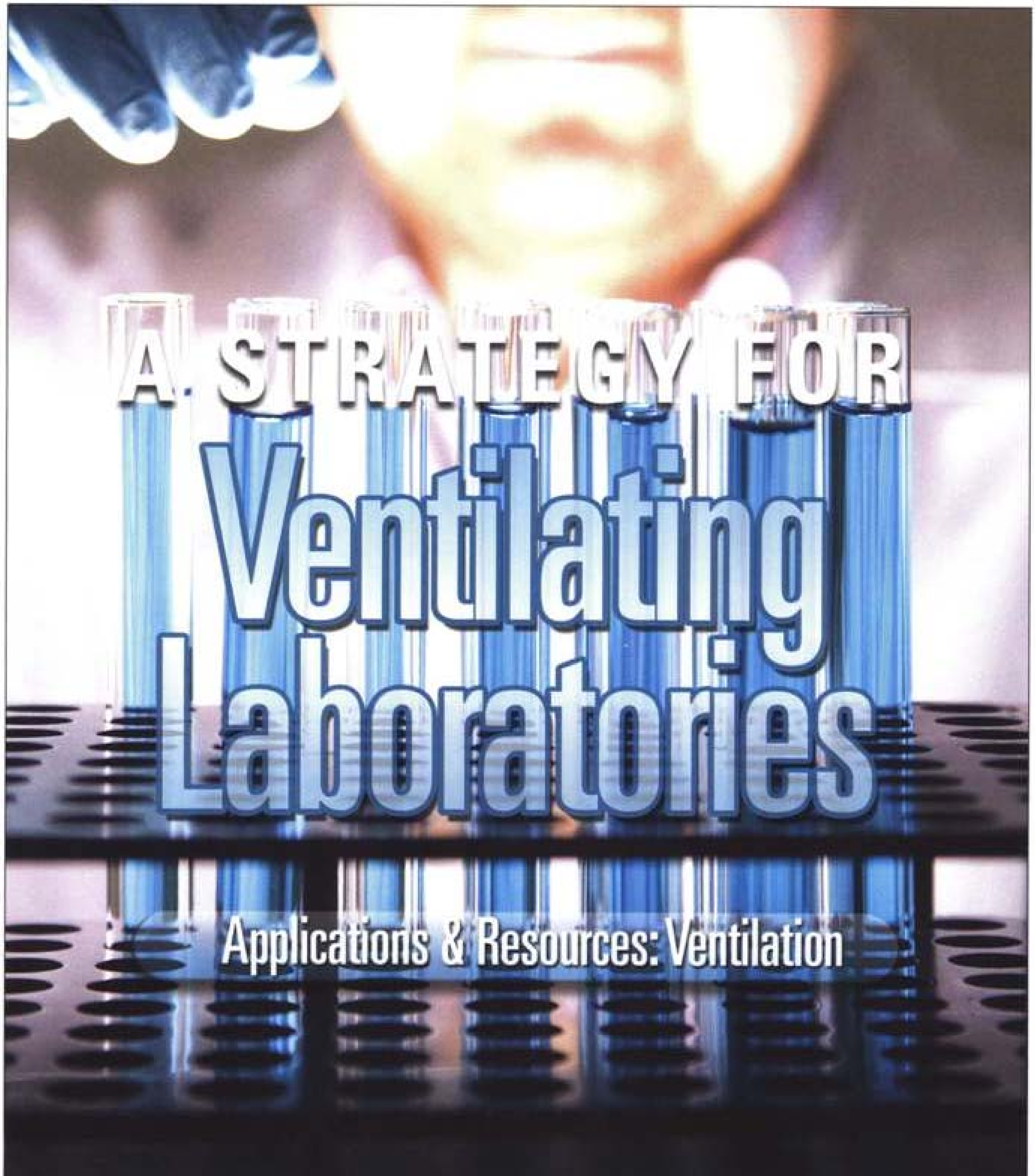

Ventilation Strategy for Laboratories

Controlling thermal comfort with cooling panels and bench exhaust

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A large portion of a laboratory's space-cooling load is the result of heat produced by research equipment. If heat can be captured at its source, its impact on space-cooling load and resulting HVAC requirements can be reduced. This article will discuss a ventilation strategy combining ceiling exhausts, bench exhausts, and ceiling radiant-cooling panels that appears to provide the best thermal conditions and largest energy savings for a typical laboratory.

RADIANT-COOLING VS. AIR-BASED SYSTEMS

Traditionally, HVAC systems are all-air systems, achieving building ventilation and cooling through convection. A more energy-efficient alternative[1, 2] can be radiant-cooling systems. In a radiant-cooling system, ventilation and thermal-space-conditioning tasks are separated with forced air used to fulfill ventilation requirements and radiant-cooling panels used to provide most of the cooling. Less parasitic energy (pump and fan energy) is required to remove heat from a space. Because walls are cooled radiantly, a higher air temperature can achieve the same level of comfort as an air-based system. The higher air temperature results in less energy lost to the outdoors. The preferred installation method for a radiant-cooling panel is a ceiling mount, which reduces air stratification and facilitates the collection of condensation.

BENCH-EXHAUST EVALUATION

A generic laboratory with a conventional air-distribution system (Figure 1) was developed as a baseline model for computational fluid dynamics (CFD) analysis. The same laboratory space then was modeled with a bench-exhaust ventilation scheme (Table 1). The bench exhausts were continuous slots along the length of the benches, mounted beneath the benches' shelves. In each of 12 cases, the bench devices generated either 5,808 w or 4,356 w of total heat. Heat generated from the equipment zone was considered in cases 4 through 12. The lighting heat sources generated 2,275 w. The sensible heat generated by each occupant was assumed to be 80 w. Solar loading generated from south-facing windows on the external wall was divided: 1,160 w was transmitted into the room, and 1,273 w was absorbed by the window glass and external-wall section. The supply temperature was 51.98°F for all cases.

Simulation results. To compare the performance of different ventilation schemes, two occupied zones were defined: the walking zone and the bench zone (Figure 2). The walking zone covered the aisles and doorways from the floor to 5 ft 9 in. above the floor. The bench zone covered the top of each bench to 5 ft 9 in. above the floor.

The average predicted percentage dissatisfied (PPD) and temperature in the two occupied zones and at the exhausts are summarized in Table 2. In Figure 3, the average PPD is presented graphically. In the baseline case, the walking zone's PPD was 12.8 percent. When bench exhausts were utilized in Case 2, the PPD in the walking zone dropped by 5.2 percent at a lower supply flow rate. Cases 1 through 9 demonstrated the effects of the bench-exhaust flow rate on the room's thermal condition.

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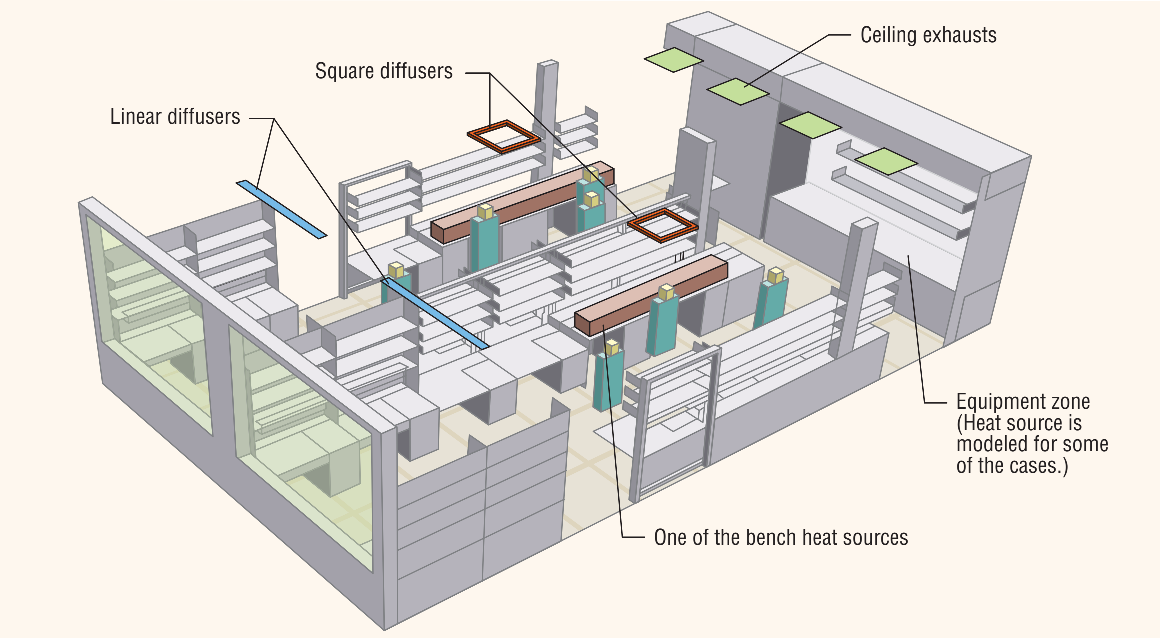

FIGURE 1. Baseline laboratory-model layout.

When the local exhaust flow rate was reduced from 800 cfm to 600 cfm, the average temperature in the two occupied zones increased by 3.24 to 5.04˚F; when the supply flow rate was lower than 10 ACH, the PPD increased significantly in the bench and walking zones. However, a comparison of cases 4 through 6 with cases 7 through 9 reveals that further reduction of the local exhaust flow rate from 600 cfm to 400 cfm resulted in an insignificant change in PPD (Figure 3). Heat-source distribution was the only difference between cases 10 through 12 and cases 7 through 9. The simulation showed that the cases with different heat-source distributions had similar average air temperature and PPD in occupied zones. In other words, the performance of the ventilation system with the bench exhausts was not very sensitive to the location of heat sources in the room. This is the ideal situation system designers want.

The average temperatures at the bench exhausts ranged from 77 to 87.98˚F, about 1.44˚F higher

	Air changes per hour	Total supply flow rate (cfm)	Door-gap infiltration (cfm)	Number of ceiling exhausts	Total ceiling-exhaust flow rate (cfm)	Bench-exhaust flow rate (cfm)	Bench heat source (watts)	Equipment-zone heat source (watts)
Baseline	13	1,550	200	4	-1,750	0	5,808	0
Case 1	10	1,210	200	2	-610	-800	5,808	0
Case 2	8	970	200	1	-370	-800	5,808	0
Case 3	6	730	200	1	-130	-800	5,808	0
Case 4	10	1,210	200	2	-810	-600	5,808	2,904
Case 5	8	970	200	2	-570	-600	5,808	2,904
Case 6	6	730	200	1	-330	-600	5,808	2,904
Case 7	10	1,210	200	2	-1,010	-400	5,808	2,904
Case 8	8	970	200	2	-770	-400	5,808	2,904
Case 9	6	730	200	1	-530	-400	5,808	2,904
Case 10	10	1,210	200	2	-1010	-400	4,356	4,356
Case 11	8	970	200	2	-770	-400	4,356	4,356
Case 12	6	730	200	1	-530	-400	4,356	4,356

TABLE 1. Bench-exhaust cases. The cases represent variations in the ventilation system and bench heat sources.

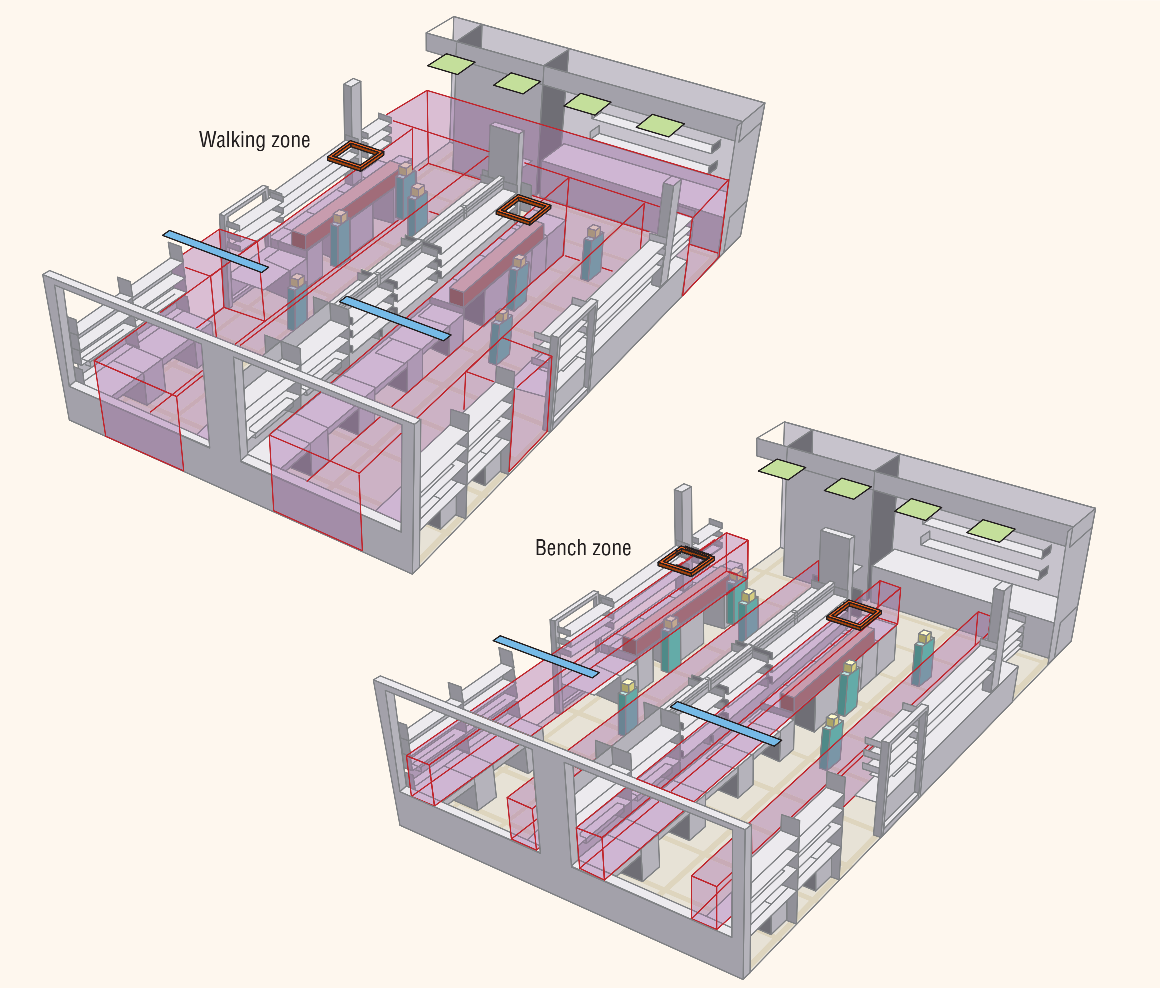

FIGURE 2. The walking zone included the red highlighted areas on the left figure. The bench zone included the volume above the benchtops in the red highlighted areas on the right figure.

than the average bench-zone temperature. However, the average air temperatures at the bench exhausts were lower than those at the ceiling exhausts, which indicated that the bench exhausts were not as effective in removing heat. This is explained in Figure 4, which shows the relative positions of the bench exhausts and heat sources used in this analysis.

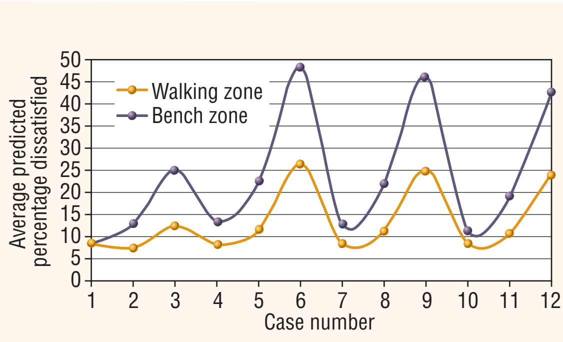

FIGURE 3. Average PPD in walking and bench zones.

The bench exhausts were not in locations favorable to the buoyancy force. Their effectiveness in removing heat from the benchtop was diminished by the strong buoyancy, which brought a large portion of heat up to the aisles. When the exhaust flow rates were at 800 cfm, the bench-exhaust temperatures were closer to those of the ceiling exhaust, indicating a greater effectiveness in removing heat. When the bench-exhaust flow rate was reduced to 600 cfm and 400 cfm, the difference between the bench-exhaust and ceiling-exhaust temperatures increased. Therefore, to be more effective, bench exhausts should be operated at a higher flow rate.

RADIANT-COOLING-PANEL EVALUATION

The second part of this analysis focused on evaluating the effectiveness of radiant-cooling panels in achieving the required thermal-comfort level in a laboratory with a reduced ventilation flow rate and its resultant cost reductions. Flush with the ceiling

	Air changes per hour	Bench exhaust (cfm)	Bench equipment (watts)	Equipment zone (watts)	Average air temp. (°F)		Average predicted percentage dissatisfied		Bench-exhaust temperature (°F)				Average temp. of bench exhausts 1-4 (°F)	Ceiling-exhaust temp. (°F)
					Walking zone	Bench zone	Walking zone	Bench zone	1	2	3	4		
Baseline	13	0	5,808	0	69.98	71.96	12.8	12.8	N/A	N/A	N/A	N/A	N/A	74.84
Case 1	10	800	5,808	0	72.32	75.02	8.7	8.6	77.90	79.16	75.74	75.56	77.00	78.98
Case 2	8	800	5,808	0	75.20	77.72	7.6	13.0	81.14	82.94	78.80	78.44	80.24	80.78
Case 3	6	800	5,808	0	79.52	81.68	12.6	25.1	84.92	87.08	83.12	82.94	84.56	84.92
Case 4	10	600	5,808	2,904	75.56	78.80	8.4	13.4	79.88	82.40	78.98	78.80	80.06	83.30
Case 5	8	600	5,808	2,904	78.80	82.04	11.6	22.6	82.58	85.28	82.58	82.22	83.12	86.36
Case 6	6	600	5,808	2,904	83.48	86.72	26.6	48.5	86.90	90.14	87.44	87.62	87.98	91.04
Case 7	10	400	5,808	2,904	75.56	78.62	8.3	13.0	78.98	81.32	78.44	78.26	79.34	82.94
Case 8	8	400	5,808	2,904	78.80	81.86	11.4	22.0	83.66	84.92	81.68	81.86	82.58	86.00
Case 9	6	400	5,808	2,904	83.12	86.36	25.2	46.2	86.36	89.78	86.72	86.72	87.44	90.68
Case 10	10	400	4,356	4,356	75.02	77.90	8.3	11.2	77.90	79.52	77.54	77.36	78.08	83.84
Case 11	8	400	4,356	4,356	80.60	81.14	10.8	19.0	80.78	82.76	80.96	80.60	81.32	87.08
Case 12	6	400	4,356	4,356	82.94	85.82	24.0	42.9	85.46	87.80	85.64	85.28	86.00	91.76

TABLE 2. Average PPD and air temperature in occupied zones and at exhausts for bench-exhaust cases.

above the benchtops and aisles, two sets of cooling panels were added to the ventilation system used in the bench-exhaust evaluation (Figure 5). One set mounted above the benchtops had three panels. The central panel was 1-ft-10-in. wide, and the two panels against the side walls were 1-ft wide each. The other set of cooling panels was mounted above the two aisles. It was assumed these panels were maintained at 57.02°F. Table 3 shows the cooling-panel arrangements for the 16 cases modeled in the radiant-cooling evaluation. The ventilation flow rate in these cases ranged from 8 to 4 ACH. Compared with the bench-exhaust-evaluation cases, this is low. This lower ventilation flow rate is needed when utilizing cooling panels to achieve the thermal comfort required in a laboratory. Note that C-Case 1 and C-Case 5 in this evaluation are identical to Case 2 and Case 3, respectively, in the bench-exhaust evaluation.

Simulation results. The average air temperature and PPD in the two occupied zones for the 16 cases are summarized in Table 3 and presented graphically in Figure 6.

The simulation results showed that the cooling panels reduced the average air temperature in the occupied zones by about 1.44 to 2.52°F. When cooling panels were utilized, sensible heat was removed

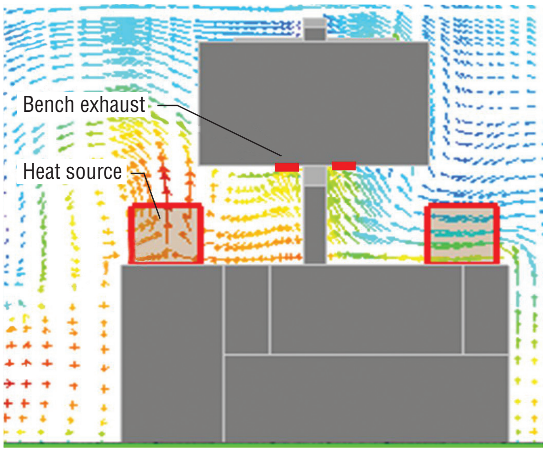

FIGURE 4. Velocity vector at a vertical plane parallel to a window wall.

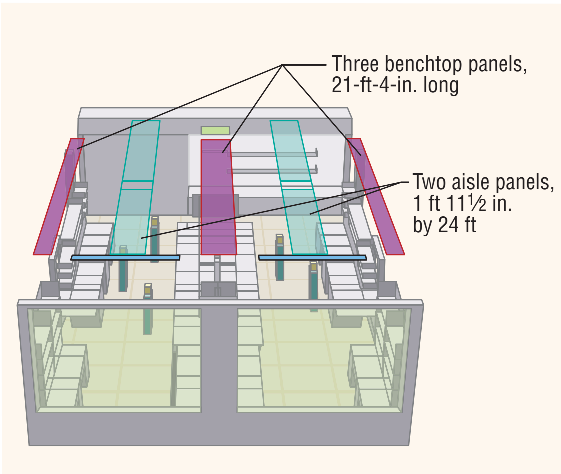

FIGURE 5. Two sets of ceiling-mounted cooling panels.

	Cooling panels	Air changes per hour	Bench exhaust (cfm)	Ceiling exhaust (cfm)	Average air temperature (°F)		Average predicted percentage dissatisfied	
					Walking zone	Bench zone	Walking zone	Bench zone
C-Case 1	None	8	-800	-370	75.20	77.72	7.6	13.0
C-Case 2	Bench	8	-800	-370	74.30	76.82	7.3	8.8
C-Case 3	Aisle	8	-800	-370	74.30	76.82	7.3	8.9
C-Case 4	Bench and aisle	8	-800	-370	73.58	76.28	7.9	7.9
C-Case 5	None	6	-800	-130	79.52	81.68	12.6	25.1
C-Case 6	Bench	6	-800	-130	77.90	80.24	9.1	16.0
C-Case 7	Aisle	6	-800	-130	77.72	79.88	8.6	14.9
C-Case 8	Bench and aisle	6	-800	-130	76.82	79.16	7.4	12.3
C-Case 9	None	5	-600	-208	81.32	83.66	16.7	30.0
C-Case 10	Bench	5	-600	-208	80.06	82.76	13.4	24.4
C-Case 11	Aisle	5	-600	-208	79.70	82.40	12.3	23.1
C-Case 12	Bench and aisle	5	-600	-208	78.98	81.50	10.0	18.6
C-Case 13	None	4	-600	-86	83.84	86.54	28.1	47.4
C-Case 14	Bench	4	-600	-86	82.40	84.74	19.5	32.1
C-Case 15	Aisle	4	-600	-86	82.22	84.56	18.5	31.2
C-Case 16	Bench and aisle	4	-600	-86	81.50	83.84	15.0	26.7

TABLE 3. Radiant-cooling-panel cases.

from the room by ventilation and radiation. The heat emitted by occupants was absorbed by the cooling panels through radiative heat transfer. Therefore, even though the average air temperature in an occupied zone was reduced by only 1.8˚F, thermal comfort was improved significantly (Figure 6). This was especially true when the ventilation flow rate was lower.

Table 3 shows that at 8 ACH, the PPD in the walking zone increased if the two sets of cooling panels were used together. This is because the air temperature already was slightly lower than the desired temperature of 74.3˚F, and the cooler temperatures of surrounding surfaces could have had a negative impact on thermal comfort. The air temperature generally was lower in walking and bench zones with aisle panels than in those with bench panels because the total surface area of the aisle panels was larger. Average PPD in the occupied zones with 6 ACH dropped below 20 percent with any cooling-panel arrangement. However, when the ventilation flow rate was at 5 ACH, it required bench and aisle panels working together to lower the PPD below 20 percent in occupied zones. At 4 ACH, even utilizing a combination of the two sets of cooling panels could not drop the PPD below 20 percent unless more cooling panels were installed.

OPERATING-COST REDUCTIONS

The simulation showed that the ventilation flow rate required to make an equipment-intensive laboratory thermally comfortable could be as high as 13 ACH with a conventional air-distribution system. (Refer to the baseline case in the bench-exhaust evaluation.) With the proposed bench exhausts, a ventilation flow rate could be reduced to 8 ACH to achieve a similar level of thermal comfort, as shown in Case 2 of Table 2. The combination of bench exhausts and ceiling-mounted radiant-cooling panels could lower a ventilation flow rate further, to 5 ACH, as seen in C-Case 12 of Table 3.

A comparison of the 13-ACH case with the 8-ACH case shows that total annual cooling-cost savings are approximately 29 percent when only bench exhausts are utilized. A comparison of the 8-ACH case with the 5-ACH case shows an additional savings of approximately 28 percent when radiant-cooling panels are added. The total savings for an annual cooling season are approximately 49 percent when bench exhausts are utilized with bench/aisle radiant-cooling panels. Figure 7 estimates annual cooling costs for these three cases.

For this cost calculation:

- The outdoor condition was taken from weather data in Washington, D.C.

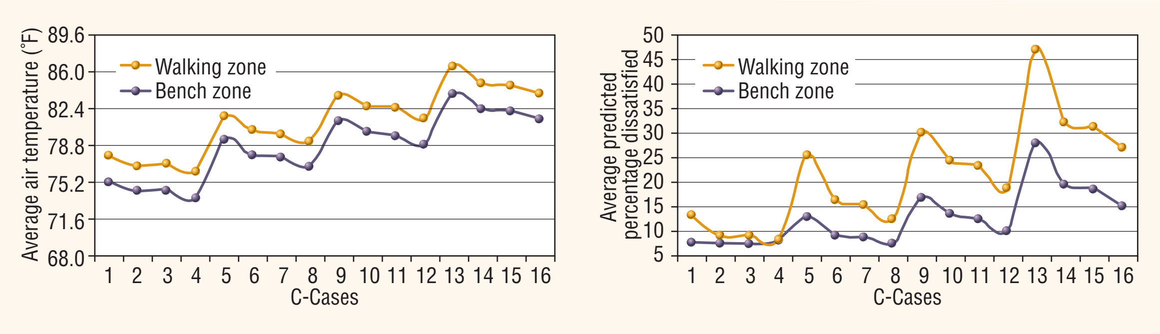

FIGURE 6. Average air temperature and PPD in occupied zones.

• The cooling season was assumed to be 4,489-hr long.

• The 13-ACH case had 70-percent outdoor air; the other two cases had 100-percent outdoor air.

• The supply-air temperature was 51.98˚F. The desired room temperature was 73.4˚F, which was used as the return-air temperature in the calculation.

• Cooling load per cubic foot per minute was considered to be the difference in air enthalpy when entering and leaving the HVAC system. Perfect duct insulation was assumed.

• The ventilation flow rate (in cubic feet per minute) was the flow rate required during a day's peak load. A day's average cooling load was assumed to be 64.3 percent of the peak load. The same was assumed for the ventilation flow rate used in the cost calculation.

• The cost of electricity was 10 cents per kilowatt-hour. Fuel was $8 per 100 Btu. Chilled-water generation efficiency was 1 kw per ton. Fan efficiency was 68 percent.

CONCLUSION

Conclusions that can be drawn from this analysis include:

• For laboratories with bench heat sources, using bench exhausts in conjunction with ceiling exhausts improves the thermal condition in an occupied zone even at relatively low total ventilation flow rates.

• The position of bench exhausts relative to a bench heat source can influence the exhausts' effectiveness in removing heat from a room. To achieve better performance, bench exhausts should be operated at a higher flow rate.

• When bench exhausts are utilized, the average temperature and PPD in occupied zones are not very sensitive to heat-source distribution.

• Without cooling panels, a bench-exhaust scheme utilizing 8 ACH and giving 30 percent of the total exhaust flow rate to ceiling exhausts and 70 percent to bench exhausts appears to provide the best thermal conditions in a laboratory.

• When bench and aisle radiant-cooling panels are utilized, ventilation flow rate can be reduced to 5 ACH while keeping PPD in an occupied zone below 20 percent. Utilizing bench exhausts and radiant-cooling panels reduces annual cooling costs by about 49 percent for a typical laboratory in Washington, D.C.

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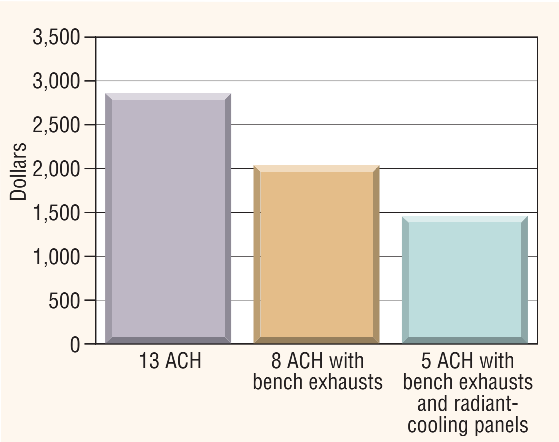

FIGURE 7. Annual cooling costs for three cases.

The total savings for an annual cooling season are 49 percent when bench exhausts and cooling panels are used.

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