CH-99-9-3

Cooling Loads in Laboratories

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ABSTRACT

The heating, ventilating, and air-conditioning (HVAC) system for a laboratory must be designed with consideration for safety, air cleanliness, and space temperature. The primary safety concern is to ensure proper coordination between fume hood exhaust and makeup air supply. Air cleanliness is maintained by properly filtering supply air, by delivering adequate room air changes, and by ensuring proper pressure relationships between the laboratory and adjacent spaces. Space temperature is maintained by supplying enough cooling air to offset the amount of heat generated in the room. Each of these factors must be considered, and the one that results in the largest ventilation rate is used to establish the supply and exhaust airflows. The project described in this paper illustrates a case where cooling load is the determining factor in the sizing of the air systems.

DESCRIPTION OF THE PROJECT

A medical center hospital is located in Burlington, Vermont. The laboratory services department occupies 16,000 gross ft^2 (1486 m^2) on the second floor of one of the buildings, of which 6,040 ft^2 (561 m^2) is active laboratory space. The laboratory department contains ten individual laboratory spaces, each with a unique arrangement. The facility was built around 1960. The health care center has an ongoing program in the older buildings to install sprinklers and remove asbestos. When it came time to do the facility's second floor, it was decided to upgrade the HVAC system as well.

The existing system was constant volume with steam reheat. There was not sufficient capacity in the main building system to cool the laboratory spaces. This was evidenced by the fact that a number of local cooling systems had been added to many of the laboratory spaces over the years. These local cooling systems were usually recirculating type systems. Recirculating air in a clinical laboratory is contrary to recommended practice (ASHRAE 1995), and the multiple units were a maintenance headache. Hospital management wanted to provide all ventilation and cooling from the main building system and eliminate all of the individual cooling units. Hospital management wanted variable-volume hood controls to be evaluated and applied if practical.

VENTILATION RATE ANALYSIS

The first step for this project was a field survey to determine the number of existing fume hoods and the condition of the existing fume hood exhaust system. The field survey revealed that there were only two fume hoods on the entire floor. Each fume hood had a dedicated, constant-volume exhaust fan serving it. The fume hood exhaust requirement for the entire department was only 1044 ft^3/min (493 L/s). This made it clear that an elaborate fume hood control package was not going to be practical. It was also clear that fume hood requirements were not going to be the governing factor of the ventilation rates in the laboratories.

The next step was to establish a minimum ventilation criteria. The governing building code, as adopted by the state of Vermont, does not dictate a ventilation rate for laboratories. Even if there was a code-mandated minimum, the actual minimum requirement for a specific project can be greater. For this project, we referenced industry standards to determine an appropriate criteria.

ASHRAE (1995) recommends a minimum of six air changes per hour (ACH) in a laboratory for most laboratory types. The American Institute of Architects' (AIA 1993) guidelines echo this. This value represents an absolute mini-

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mum, and in our experience is below what we have seen as industry practice. One example is the Department of Veterans Affairs' (DVA 1992) requirement of a minimum of 10 ACH for laboratories in their health care facilities. This was a practical reference for this project since this laboratory is part of a hospital.

The primary laboratory spaces for this project had been originally designed for between 10 ACH and 15 ACH. These ventilation rates exceeded all of the established minimum values discussed above but were insufficient to maintain comfortable temperature levels in many of the spaces. This meant that the driving factor for determining the ventilation rate in the laboratory spaces was going to be the cooling load in some or all cases. We established 10 ACH as the minimum ventilation rate for occupied periods but expected most or all spaces to exceed this based on cooling load requirement.

COOLING LOAD ANALYSIS

The calculation of cooling load for a laboratory is not different from calculating for other types of spaces. Consideration must be given to solar gain, transmission, people, lights, and equipment. Our walk-through of this department made it apparent that the equipment component of the load was going to be very large. The various types of equipment that were crowding the spaces included refrigerators, freezers, incubators, ovens, baths, computers, centrifuges, and a large array of electronic analyzers. The only thing that all the equipment had in common was that we did not know the heat generation (cooling load) of any of it.

We recognized the need to perform a thorough analysis of the equipment to obtain an accurate load estimate. Our analysis included a detailed survey of all miscellaneous equipment. In our initial survey, we recorded the name of the equipment and the nameplate rating (where available), and we interviewed the users to determine how often the equipment was run. Nameplate ratings have been shown to be misleading when used to determine air-conditioning loads of office equipment (Wilkins and McGaffin 1994). We believed that the same was likely of this laboratory equipment. We decided to take measurements to verify electric consumption instead of relying on nameplate data. Wherever possible, the idle and operating electrical consumption of the equipment was measured. Table 1 is a partial list of our findings.

After the testing, we still had a number of items where we had to estimate the cooling load because the equipment could not be tested. We based our estimate on a similar item that had been tested. This gave us measured data or a solid estimate for the maximum cooling load of most of the incubators, ovens, baths, computers, centrifuges, and electronic analyzers.

Diversity was assessed by comparing the idle consumption to the operating maximum consumption. Centrifuges had a significant reduction in electrical consumption while idle as compared to when operational. Other equipment such as electronic analyzers had the same consumption whenever the unit was powered. These data, coupled with our information relating to typical usage patterns, allowed us to apply appropriate diversity factors. Actual diversity factors used are discussed below in the "Results" section.

TABLE 1
Laboratory Equipment Heat Generation

Department	Equipment Item	Nameplate (Btu/h)	Measured Idle (Btu/h)	Measured Operating (Btu/h)
Histology	Gravity Incubator	1024	0	941
	Tissue Analyzer	Not Available	410	1024
Chemistry	Immunoassay System	2364	601	601
	Ektachem Analyzer	Not Available	6143	7781
	CX3 Analyzer	2048	942	1311
	Blood Gas System	1638	0	409
	CO Oximeter	853	314	628
	Centrifuge-1	2048	0	1761
	Centrifuge-2	3276	0	2048
	Polarization Analyzer	2048	248	573
	Hemoglobin Testing	1638	410	573
	Convection Incubator	Not Available	0	225
Hematology	Hematology Analyzer	3754	1334	1884
	Centrifuge-3	5119	0	2048
Microbiology	CO_2 Incubator	2048	0	1352

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The cooling load from refrigerators and freezers proved very difficult to measure or estimate. There were 17 units on the floor from eight different manufacturers. Simply measuring the idle and operating electric consumption does not give enough information to determine the load and diversity factor. Refrigerators actually remove heat from the space as well as add heat to the space. Heat is removed, for example, whenever the door is opened and cold air escapes into the space. The net heat gain must be considered, not simply the heat rejected.

The compressor and the condenser are the components of the refrigerator that add heat to the space. The amount of heat that the compressor adds directly to the space is small. The condenser is the component that is most important. The heat rejection of the condenser will be greater than the cooling capacity of the refrigerator. This heat is only rejected when the compressor is operating. The amount of time that the compressor runs depends on static factors such as the amount of insulation and on dynamic factors such as how often the refrigerator is opened and how much warm material is placed in the refrigerator.

With all these variables, we were not able to determine the actual cooling load of the refrigerators and freezers by a scientific method. All of the refrigerators and freezers in our project had a local air-cooled condenser rejecting heat directly to the space. We reviewed some published research (Simons and Davoodpour 1994), talked to some manufactures, and applied some engineering judgment. In the end, we estimated between 1000 Btu/hand 3000 Btu/h (293 W and 879 W) for each refrigerator or freezer.

Table 2 is a summary of manufacturer's data for two units that look similar from their outward appearance. The major difference between the two is that unit #1 is designed for 40°F (4°C) and unit #2 is designed for –107°F (–77°C). The manufacturer of unit #1 reported that it was expect to run 30% of the time if the door was not opened and that the expected load on the air-conditioning system was 1077 Btu/h (316 W). Unit #2, on the other hand, was expected to run nearly all of the time, even when the door was not opened. Here, the manufacturer recommended a value of 2860 Btu/h (838 W) for the load on the air-conditioning system.

TABLE 2
Refrigerator-Freezer Heat Gain Comparison

Description	Unit #1 Refrigerator	Unit #2 Freezer
Temperature	+40°F	–107°F
Compressor	Single	Dual-Cascading
Cooling Capacity	1600 Btu/h	750 Btu/h
Heat Rejection	2000 Btu/ h	3000 Btu/h
Coefficient of Performance	4.00	0.33
Recommended Heat Gain*	1077 Btu/h	2860 Btu/h
Insulation Value	14°F·ft²·h/Btu	31°F·ft²·h/Btu
Storage Capacity	27.3 ft³	12.0 ft³

* As recommended by unit manufacturer.

Once all the equipment loads were established, the calculation of the overall cooling load was straightforward. We used a commercial load program that is based on the transfer function method (TFM) (ASHRAE 1997). The TFM models building mass heat dynamics and allows a distinction between radiated and convected forms of cooling load. We were not able to determine the radiant/convective split for this equipment. To be conservative, we assumed 100% convective load. Building envelop load components were modeled based on TFM-calculated radiant/convective splits. Figure 1 shows a breakdown of the combined space cooling load of the ten laboratory spaces.

RESULTS

Figure 1 illustrates the importance of the equipment load to the overall space cooling load for this case. An error in estimating the equipment load would have led to a large error in the overall cooling load. This illustrates why it is important to fully understand the equipment load. Overestimating this component of the load would result in wasted energy and unnecessary first costs. Underestimating this would result in an HVAC engineer's worst nightmare: a critical space with not enough cooling.

Figure 2 illustrates the relative importance of the various types of equipment on the overall equipment cooling load. Analyzers and computers combined totaled 64% of the equipment load. As we discussed above, we were able to measure the load of most of this equipment and also determine realistic diversity factors. Analyzers were generally left with power on, and our measurements indicated that the idle vs. operational power consumption was equal or nearly equal. We applied a 90% diversity factor to this equipment.

Based on this, we felt that we were very accurate on 64% of the total equipment load. The only other significant contributor to the cooling load was the refrigerator and freezer load. Here, as discussed above, we were forced to use what we feel are very conservative estimates. At only 24% of the equipment load, some uncertainty in this load component will not cause a large error in the overall load. The values used account for the duty cycle of the compressors. All refrigerators and freezers are in use continuously. An additional diversity factor was not appropriate.

This detailed load analysis for our project resulted in calculated supply rates between 10 ACH and 20 ACH for the ten laboratory areas. The average for the ten was 13.8 ACH. The average equipment load is 6.71 W/ft² (72.2 W/m²) with a maximum of 10.65 W/ft² (114.6 W/m²). A chemistry laboratory had this largest load, requiring 21.8 ACH. Table 3 gives a summary of some typically referenced air system facts.

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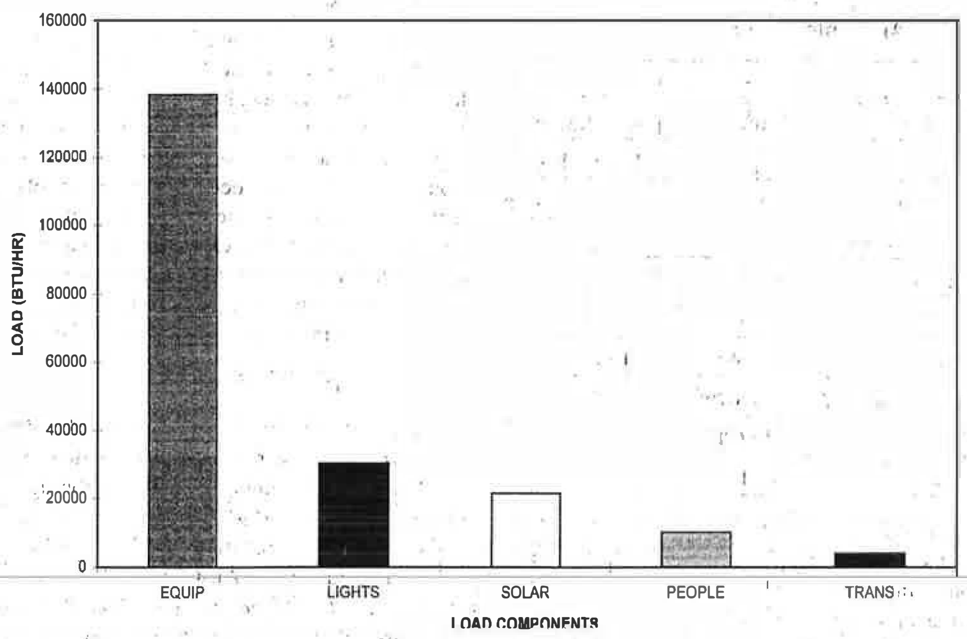

Figure 1 *Space sensible load components.*

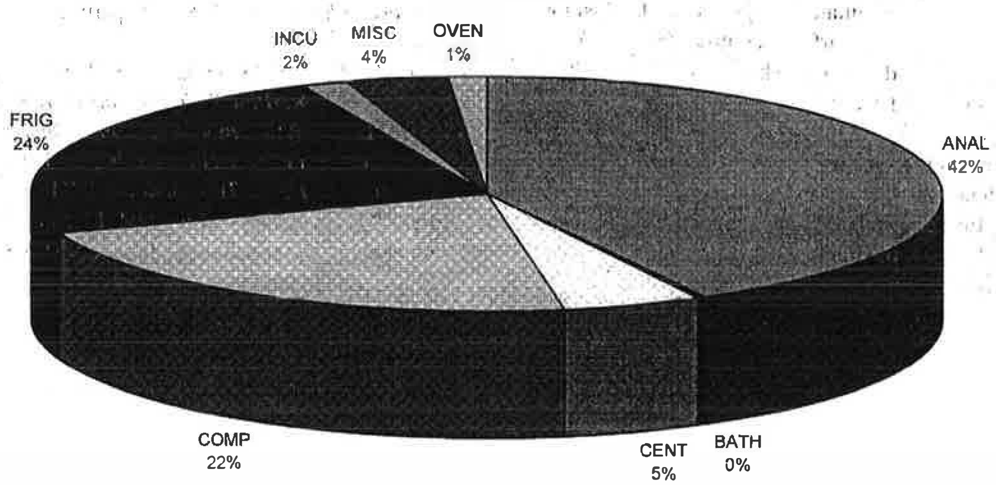

Figure 2 *Equipment load by type.*

SYSTEM SELECTION

We designed the air distribution system to deliver a minimum of 20 ACH to all of the primary laboratory spaces. This allowed the hospital the flexibility to reconfigure the laboratory space in any manner in the future and to have spare cooling capacity if more equipment is added. All referenced minimum ventilation rates were met and sufficient makeup air was provided to all fume hoods. Two new air-handling units were selected to provide supply to this laboratory area plus some adjacent areas.

The 100% outside air AHU include 12-row cooling coils with copper fins, face and bypass steam heating coils, 30% prefilters, 90% final filters, and humidifiers to maintain a minimum of 30% relative humidity. Copper fins were selected to minimize microbial growth on the coils' fins. Copper has been reported to be of higher toxicity to microorganisms then aluminum (Gill and Wozniak 1993). Copper fins also improve heat transfer by up to 5%.

The laboratory air-distribution system was designed with 11 individual zones. Each zone was provided with an occu-

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TABLE 3
Air System Facts

Item	I-P	SI
Average Airflow	1.85 (ft^3/min)/ft^2	9.39 (L/s)/m^2
Average Room Air Change	13.8 per hour	13.8 per hour
Coil Load Ratio	103 ft^2/ton	2.72 m^2/kW
Space Sensible Heat Ratio	96%	96%
Coil Sensible Heat Ratio	59%	59%
People Density	145 ft^2/person	13.5 m^2/person
Lighting Level	1.50 W/ft^2	16.1 W/m^2
Average Equipment Load	6.71 W/ft^2	71.9 W/ft^2
Supply Air Temperature	57.0°F	13.9°C
Laboratory Location	Burlington, Vermont	

pied-unoccupied switch. A maximum of 20 ACH and a minimum of 6 ACH were delivered. Variable-speed drives on the supply and exhaust fans modulated to match the required total airflow of all the individual zones. The fume hoods were exhausted at a constant volume and with individual fans. The required fume hood exhaust did not exceed the minimum of 6 ACH (unoccupied ventilation rate) in any of the zones.

The laboratories can maintain their pressure relationships by the action of automatic airflow control devices (VAV boxes) on the supply and general exhaust to the room. The occupied and unoccupied set points are fixed based on maintaining the desired differential airflow. Supply airflow can vary based on room temperature, with general exhaust flow tracking to maintain room pressure. Provisions were considered to allow future installation of heat recovery (space constraints and location of supply and general exhaust mains make this difficult to implement and service).

CONCLUSIONS

Engineers can encounter many different types of laboratory spaces. The project described here is an example of a laboratory that had very high internal heat loads due to a heavy use of electronic equipment. A detailed load analysis, including field measurement of actual equipment power consumption, was performed to determine as accurately as possible the actual cooling load of the spaces. Our analysis indicated that nameplate data will overstate the actual load of electronic equipment. This analysis also illustrated that in some cases, cooling load can be the determining factor in establishing room airflow rates and overall air system capacity.

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