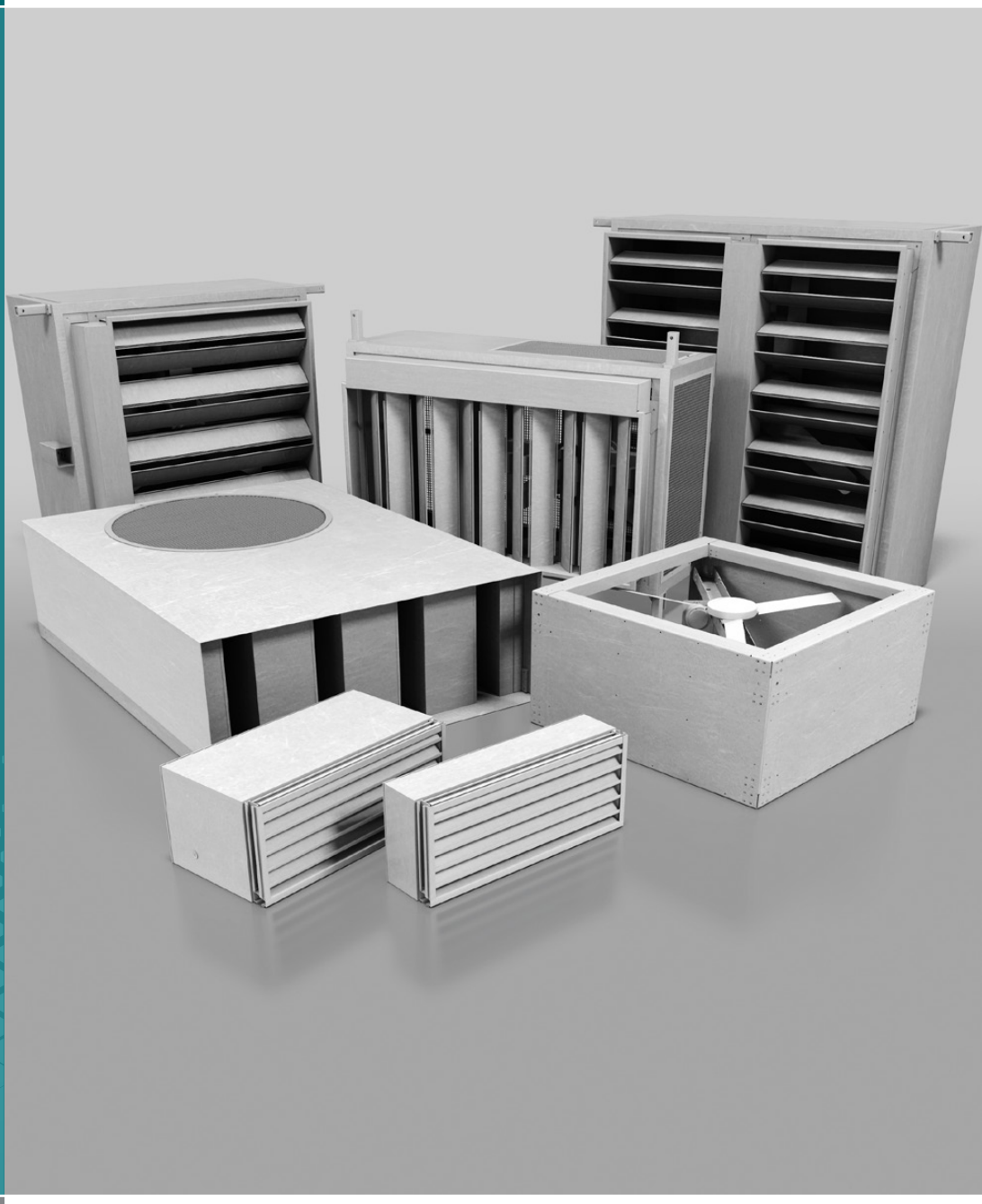
SECTION **K**



Engineering Guide
Natural Ventilation

Introduction to Natural Ventilation

Overview

The prevalence of sustainable construction is increasing due to rising energy costs, improving life cycle cost and design mandate. Engineers are utilizing more passive technologies to manage building heating and cooling demands. These approaches might take the form of low energy systems such as radiant heating and cooling, or thermally active building systems. These systems often require the use of an energy intensive compression cycle somewhere in the system.

Increasingly, natural ventilation is employed to leverage freely available resources such as wind and outdoor air to satisfy cooling loads and provide occupant comfort. In its most basic form, natural ventilation provides openings in the building façade to allow fresh outdoor air in one area of the building and out another. As the fresh outdoor air passes through, heat is removed and ventilation is provided. This technology is only appropriate where climatic conditions allow. The use of mechanical heating and cooling in combination with natural ventilation, as a hybrid or mixed mode system, can extend the acceptable climate conditions where natural ventilation can be effectively applied to utilize free cooling for a large portion of the year.

Benefits of Natural Ventilation Systems

Natural ventilation systems have been successfully used in several applications, including:

- Schools
- Hospitals
- Supermarkets
- Office Buildings
- Warehouses

There are many benefits of natural ventilation systems. Advantages of using the outdoor climate to condition buildings include:

- Energy efficiency
- Improved indoor environmental quality
- Potential for lower capital cost
- Reduced maintenance and replacement
- Compatibility with daylighting
- Increased range of thermal comfort
- LEED® and CHPS® points

Energy Efficiency

In the United States, the energy used in the running of buildings accounts for ~40% of the total energy consumed. There is strong evidence which suggests that naturally ventilated buildings can use significantly less energy than fully mechanically ventilated, air-conditioned ones (Busch,1992; Zhao and Xia 1992; and Finnegan et al., 1994). This is, in large part, because natural ventilation involves the ventilation of a space without the use of fans, but instead uses the forces of nature – wind and buoyancy – to drive the air flow exchange between the interior and exterior, and to mix the air within the space.

Baker and Steemers (2000) assessed the energy consumed by office buildings **(Figure 1)**, and found that those which are naturally ventilated can consume less than half the energy that fully air-conditioned, mechanically ventilated ones consume.

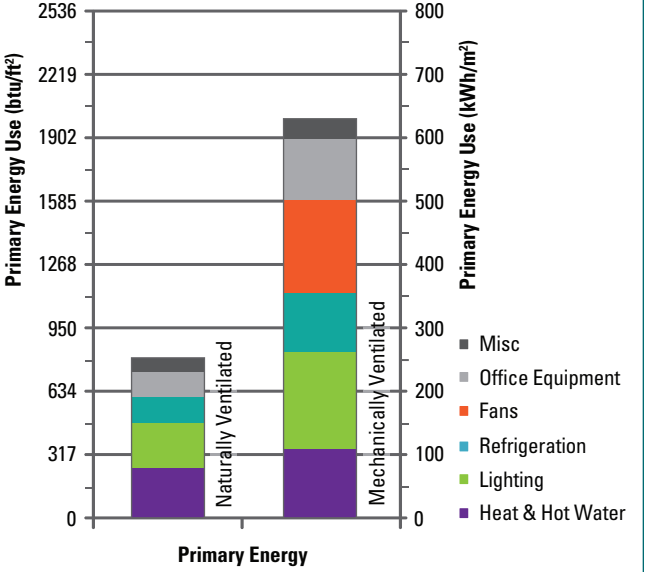


Figure 1: Energy use in office buildings (Baker and Steemers, 2000)

The sources of energy savings are as follows:

- Reduction in the fan power used in mechanical ventilation to drive flow from the exterior to the interior and vice versa, and the fan power used to distribute air throughout the building via ducts
- Reduction in lighting load – Buildings which are naturally ventilated typically require the depth of the floor plate to be reduced, increasing the opportunity for the more effective use of natural daylight.
- Reduction in refrigeration load – Although mechanical refrigeration can be used in conjunction with natural ventilation, it is typically used less than in mechanical ventilation. Naturally ventilated buildings are often designed to include thermal mass, night cooling, and use outdoor air to manage internal loads with the ventilation air drawn through the building.
- Occupants of naturally ventilated buildings are often more tolerant of fluctuations in the indoor climate. They tend to accept a wider range of temperature and humidity levels, allowing for the thermostatic set-point to be raised slightly in the summer and lowered slightly in the winter. This leads to reduced energy consumption by mechanical equipment that is "trimming" the room temperature by either providing heating in the winter or cooling in the summer.
- Reduction in energy load from office equipment – There is evidence to suggest that by putting people in a low-energy building, they respond positively and are more likely to turn off lights, equipment etc. when not in use, and/or to purchase low-energy equipment.

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Introduction to Natural Ventilation

Improved Indoor Environmental Quality

Occupants often desire the ability to control their local environment in a building by being able to open windows. This feature is often not provided – in many cases for good reasons – in mechanically ventilated buildings. It can be disastrous in terms of occupant comfort and energy usage if windows are left open whilst a mechanical system is operating. However, opening windows can be readily designed into natural ventilation strategies; in fact, they can be designed to be an important part of the strategy.

Reduced Capital Cost

Most commercial buildings use significant amounts of equipment in order to mechanically ventilate and air-condition the interior. Sometimes the HVAC equipment, associated ductwork, grilles, diffusers etc. can account for up to 30% of the capital cost of a new building. Naturally ventilated buildings involve much less equipment and ductwork space. Designers are presented with a means to reduce the overall cost of the building. Budget needs to be allocated for the natural ventilation equipment and controls, but this should be less than that set aside for mechanically ventilated buildings.

Reduced Maintenance and Replacement

Buildings with mechanical equipment require maintenance and often have significant costs associated with these activities, whereas natural ventilation equipment is often maintenance-free or at least low-maintenance.

The mechanical plant in a mechanically ventilated building will require a significant overhaul, refurbishment or even replacement in a 15-20 year timescale. In contrast, natural ventilation equipment typically lasts longer, and if there is a requirement for replacement of any equipment, it is relatively inexpensive – the weatherproof louvers, control system, ducts, and shafts are extremely unlikely to need replacement.

Natural Daylighting

The provision of adequate light is a key requirement for occupied buildings, and there is much benefit to be gained through exploiting the benefits of natural daylight. Firstly, it can lead to reduced electricity usage; and secondly, there are variations in sunlight levels over the course of a day which can be beneficial to occupants, especially if potential issues with excessive amounts (glare) or insufficient lux levels are overcome through careful design. However, the provision of natural daylight can lead to admission of solar gain into a space. There are solar films available on the market that mitigate this effect to some degree, but nevertheless, it is still an issue that designers need to consider. The main challenges of solar gain typically exist in summer, when it is desirable to keep the building cool and minimize the heat gain. Summer is when the sun is highest, and therefore, with appropriate external shading, direct sunlight incident on glazing can be avoided as shown in **Figure 2**. In contrast, cold winter days can present a heating challenge and solar gain can be of benefit to the building. In winter, the sun is much lower than in summer, therefore, the external shading devices used in a building can be devised to maximize the benefits in winter whilst avoiding the gain in summer.

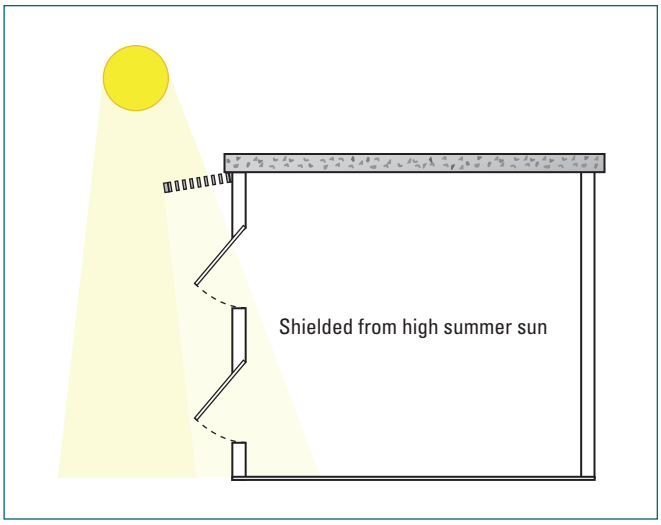

Figure 2: Schematic of external shading device on a building with shielding from high summer sun

Increased Range of Thermal Comfort

Research has demonstrated that occupants who are provided with some level of control (i.e. operable windows) form different expectations regarding their thermal environment. As a result, they tend to be comfortable even when conditions vary from what would be considered comfortable in the classical sense (de Dear et al., 1997).

LEED and CHPS Points for Natural Ventilation

The LEED Green Building Rating System places significant emphasis on natural ventilation. LEED has two prerequisites and two environmental quality (EQ) credits related to ventilation for indoor air quality to which natural ventilation can provide a positive contribution. The related LEED prerequisites include EQ p1 and EQ p2, while the associated credits are EQ c2 and EQ c6.2. Natural ventilation may also contribute to reduced HVAC energy use, achieving points under LEED Energy and Atmosphere Credit 1. Additionally, the Collaborative for High Performance Schools (CHPS) rating program for K-12 schools gives credit for natural ventilation.

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Challenges of Natural Ventilation

There is still reluctance from various parts of the building industry to change design strategies to more natural ones. The reasons cited by reluctant designers are varied, and include the following:

- Naturally ventilated buildings can be more difficult to control
- Naturally ventilated buildings are subject to higher risks of overheating in summer and cold draughts in winter
- Naturally ventilated buildings cannot control humidity
- Naturally ventilated buildings cannot be used as easily in noisy environments
- Naturally ventilated buildings are unable to filter the air
- Naturally ventilated buildings tend to consume significant amounts of energy in winter in order to condition the ventilation air

Control

All buildings with reasonably large spaces and vertically and horizontally distributed heat loads can be challenging to control in terms of thermal comfort, regardless of the ventilation system used. One of the reasons is that large spaces can be subject to large scale convective flow patterns, which can result in temperature differences within the space. In very large spaces, one way of overcoming the challenge in cases where the building is mechanically ventilated is to increase the flux of air driven through the space. This, in turn, increases the energy use associated with the ventilation system.

There is evidence to suggest that in large multi-storey buildings with floors linked via atria, the temperature variations experienced as a result of the challenges with control are much greater than intended or agreed upon during the design stage of a building.

Air Quality

The challenge that naturally ventilated buildings cannot be used when air filtration is required can be difficult to address. However, if a designer incorporates an opening window as part of the design, the suggestion of including air filtration elsewhere in the building or room would seem inappropriate. In situations where the requirement for air filtration is derived from outside air particulate concentrations being high at street level, it is possible to adopt a natural ventilation strategy in which all air inflow is achieved at a high level where concentrations of pollutants tend to be lower (CIBSE, 2005).

If the building is sufficiently tall, the improvement in outside air quality with height away from the street may be sufficient to allow for natural ventilation to be used.

Some rooms, such as clean rooms, require air filtration regardless of the outdoor conditions. These rooms should be mechanically ventilated and meet the required level of filtration. It may be possible, however, to naturally ventilate spaces that are not required to meet the same high standards, such as atria, office spaces, etc.

Acoustics

The challenge of acoustics can be overcome through careful design and acoustically attenuated inflow and outflow ducts where necessary. Interestingly, mechanical equipment itself can sometimes present designers with challenges regarding internal noise levels. Therefore, if inlet/outlet vents are treated with acoustic baffles to manage ingress of noise, natural ventilation strategies can sometimes make it easier to demonstrate compliance with noise level criteria.

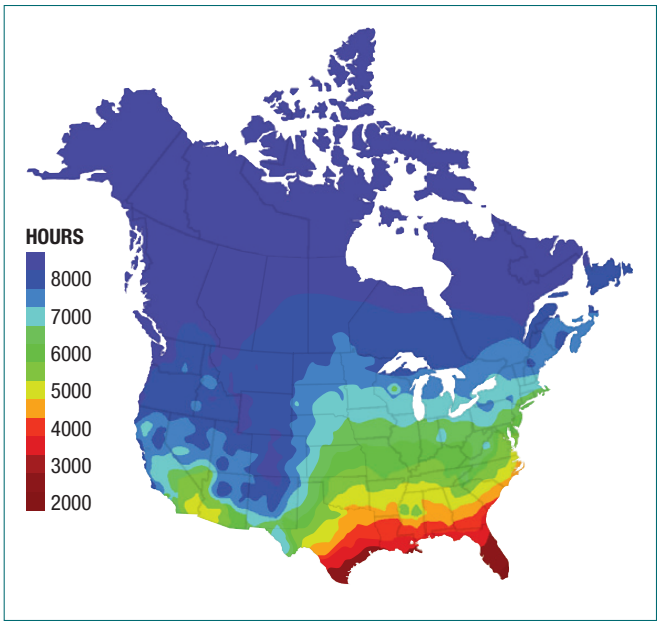

Figure 3: Estimate of Air-Side Economizer Hours for Data Centers ©2009 The Green Grid

Humidity and Climate Suitability

It is difficult to control humidity in naturally ventilated buildings. Depending on the local climate, the extent to which this is a challenge will vary. In climates such as British Columbia, Washington State and other western US states, this is easier to manage than in areas such as the southeastern US where the summer months see high temperatures and high relative humidity levels **(Figure 3)**.

In climates such as Minnesota, a building may be subject to very dry conditions in deep winter (resulting from heating of cold exterior air). However, this does not mean that natural ventilation should be ruled out completely. A hybrid scheme which makes use of natural ventilation for the mid-seasons and a full mechanical ventilation system with humidity control in the depth of winter and height of summer may be possible.

Heating Requirements and Overheating

Finally, concerns regarding heating energy use and cold drafts in winter, as well as overheating in summer, can be addressed through appropriate design of a controlled natural ventilation strategy and incorporation of thermal mass and natural night-time cooling if necessary. It is also important that the appropriate level of automatic control be included in the design. If the building has occupants who wish all ventilation openings to be automatically controlled, the system should account for this. Such an example may be a theater – there is usually no desire of theater-goers to open or close vents. However, in office buildings, there may indeed be a preference for occupants to be able to open or close windows. In this case, the level of impact on the immediate environment and elsewhere in the building needs to be assessed with modeling of the air flows. There may be benefit in providing a manual override function, but for a limited period of time after which the system reverts to fully automatic.

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Thermal Comfort in Naturally Ventilated Buildings

Whether or not an occupant is thermally comfortable is generally assessed as a heat balance of the occupant in connection with their surroundings. The most typical method for evaluating how well an indoor environment performs from a thermal comfort standpoint is by evaluating the factors influencing this heat balance:

- Air temperature
- Air speed
- Relative humidity
- Metabolic rate
- Clothing level
- Temperature of internal surfaces

and comparing their combination with a well-defined baseline, typically described as a predicted mean vote (PMV) or percentage of people dissatisfied (PPD). ASHRAE Standard 55 (ASHRAE, 2004 and 2010) and ISO 7730 (ISO, 2005) provide relatively stringent PMV and PPD targets for the acceptable conditions within the occupied space. These thermal comfort models, due to their analytical nature, do not capture the psychological influences that can have a significant effect on the occupants' perceived comfort.

Ultimately, the degree to which an occupant is thermally comfortable is affected by the measures listed above, but in the context of how the environment which they are creating reflects the occupant's expectation, as well as how much they feel empowered to control those factors. Taken alone, they can give indication of what would be considered acceptable (or unacceptable) conditions in an average way, though it may not provide the entire picture for a specific project.

In most cases, the psychological factors, which are more difficult to quantify, tend to broaden the range of acceptability, rather than constrict it. The PMV models provide a tight range of thermally acceptable conditions, which is fairly consistent with the expectations that one would expect an occupant to form, spending the majority of each day in an air-conditioned environment: homogeneous, devoid of significant changes in temperature, etc.

Meeting these environmental conditions can often be challenging when using a natural ventilation strategy due to the dependence on the outdoors to condition the interior space. There is little, if any, control over the temperature and moisture content of the incoming air, and it can therefore be challenging to satisfy the PMV requirements

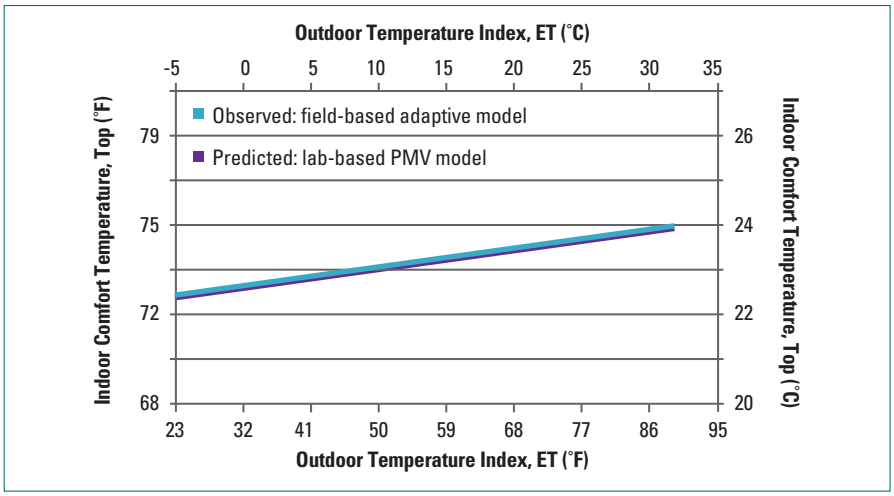

Figure 4: Observed and predicted indoor comfort temperatures from RP-884 database, for HVAC buildings (Brager and de Dear, 2001)

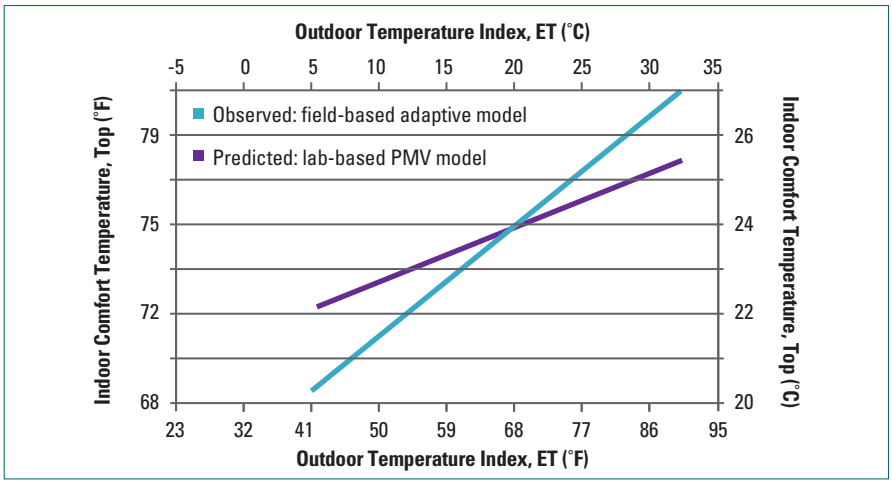

Figure 5: Observed and predicted indoor comfort temperatures from RP-884 database, for naturally ventilated buildings (Brager and de Dear, 2001)

of the standards. Interestingly, research demonstrates that occupants who are provided with some level of control (of operable windows, for example) form much different expectations and, as a result, tend to be comfortable, even when conditions vary from what would be considered comfortable in the classical sense (de Dear et al., 1997).

Data collected for ASHRAE Research Project RP-884 (de Dear et al., 1997) demonstrates this phenomenon. The research team separated the buildings surveyed into those with air-conditioning (HVAC) and those which were naturally ventilated (NV). The results from occupant surveys are shown in **Figures 4 and 5**, above.

Figure 4 shows that the occupant surveys trend very well with what would be considered acceptable using a PMV model. This bodes well for using this to evaluate the thermal environment of air-conditioned spaces. **Figure 5**, meanwhile, demonstrates the influence of factors outside those considered in a PMV model. The results of the occupant surveys demonstrate that the broader range of indoor temperatures experienced in naturally ventilated spaces fall within the accepted range for occupant comfort. For example, at an observed outdoor temperature of 77 °F (25 °C), the observed indoor temperature is 77 °F (25.2 °C), which is well within the accepted range for occupant comfort.

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Thermal Comfort in Naturally Ventilated Buildings

In addition to the demonstrably wider satisfaction range with naturally ventilated spaces, they tend to perform well in terms of thermal comfort and indoor air quality, compared to other buildings. The Center for the Built Environment maintains the largest database of building performance metrics (Zagreus et al., 2004) in the industry. Over the years, this survey has been effectively used to compare occupant satisfaction over a range of factors. **Figures 6 and 7** indicate that naturally ventilated and mixed-mode buildings, represented by the green squares and orange triangles respectively, have higher than average scores in the areas of thermal comfort and indoor air quality.

Adaptive Comfort

In recognition of this, ASHRAE (2010) includes an optional alternate method for determining acceptable indoor temperature ranges for naturally ventilated buildings. In cases where the primary means of temperature control is through the opening and closing of operable windows and where occupants are free to vary their clothing in response to the indoor and outdoor conditions, the adaptive comfort portion of ASHRAE Standard 55-2010, described by **Figure 8**, may be used. This figure provides temperature ranges for 80% and 90% acceptability (ASHRAE, 2010). The 80% acceptability limits should be used for typical applications. In situations where a higher level of occupant comfort is desired, the 90% limits should be used.

When the adaptive comfort method is used, there are no limits placed on indoor relative humidity or air velocity. This method also already accounts for local discomfort, so analysis of the percent dissatisfied due to draft, stratification, radiant asymmetry, floor temperature, etc. need not be evaluated. In situations where the designer feels it is desirable to analyze these criteria anyway, ASHRAE deems this to be appropriate. For more information on adaptive comfort, local discomfort and PMV / PPD, please refer to the ASHRAE Handbook (ASHRAE, 2009) or ASHRAE Standard 55-2010 (ASHRAE, 2010).

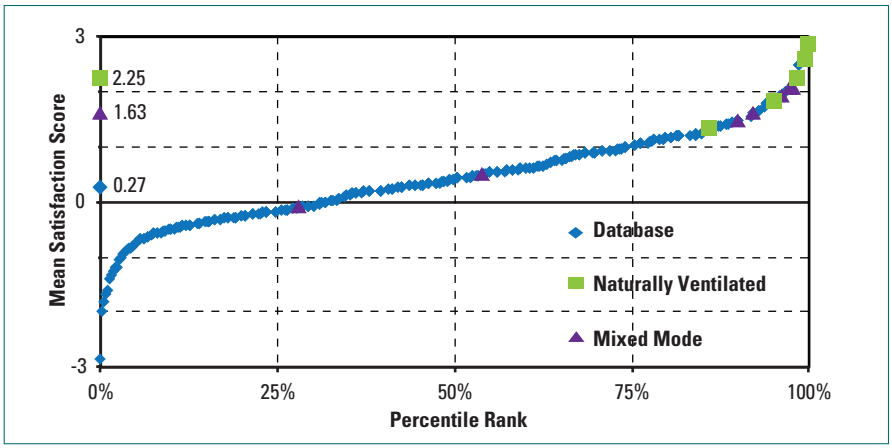

Figure 6: Mean air quality scores of buildings that are naturally ventilated and use mixed-mode air conditioning vs. buildings with full HVAC systems (Courtesy CBE)

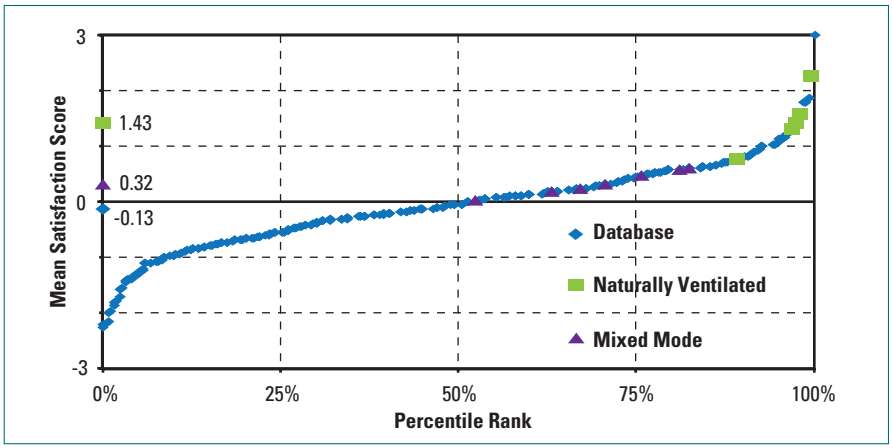

Figure 7: Thermal comfort scores of buildings that are naturally ventilated and use mixed-mode air conditioning vs. buildings with full HVAC systems (Courtesy CBE)

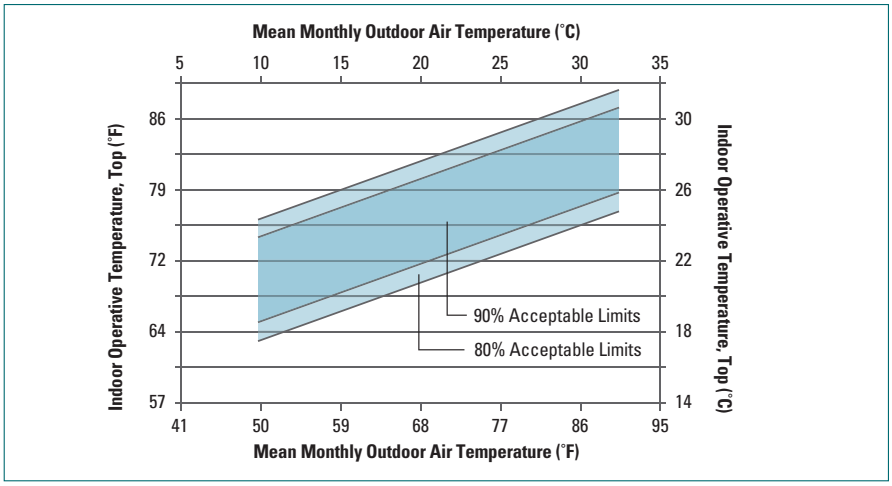

Figure 8: Acceptable indoor temperature ranges for naturally ventilated buildings (ASHRAE, 2010)

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Wind-Driven Flow

The two main drivers causing natural ventilation flows are buoyancy and wind. In nearly all instances these work in concert, with the resulting flow being the combination of the two.

Natural ventilation strategies based on wind involve the design of a building with openings on both the windward and leeward exposures of a building. In this case, a pressure differential is created across the building structure as a result of the velocity reducing on the aspect of the building facing the wind, and the velocity increasing as the air passes over the roof and other sides of the building, as shown in **Figure 9**.

The surface pressures are determined by three primary factors:

1. Wind speed

The wind-driven pressure difference can be derived from Bernoulli's equation, assuming no change in elevation:

IP $$\Delta p_w = 0.013\left(c_{p-ww} - c_{p-lw}\right)\frac{\rho U_{ref}^2}{2}$$ K1

SI $$\Delta p_w = \left(c_{p-ww} - c_{p-lw}\right)\frac{\rho U_{ref}^2}{2}$$ K1

Where:

U^2_{ref} = Time-averaged approach wind velocity in mph or m/s

ρ = Air Density in lb_m/ft^3 or kg/m^3

Δp = pressure difference from ambient in in. w.c. or Pa

$C_{p\text{-}lw}$ = Wind pressure coefficient of windward surface

$C_{p\text{-}lw}$ = Wind pressure coefficient of leeward surface

As the speed decreases, the static pressure increases, and vice versa. The air speed on the windward side of the building is reduced as it collides with the building, resulting in an increase in the static pressure. Conversely, the air speed on the top, sides and leeward side of the building increases, resulting in a reduction in local static pressure, and thereby a pressure differential through the building.

2.Wind direction relative to the building orientation

Wind direction relative to the building orientation determines the sides of the building that are exposed to the wind and the angle at which the wind will hit the surface. This will in turn affect the magnitude of the differential pressure across the building.

3.Shape of the building

As with the wind direction, the shape of the building will have a significant effect on the way the air comes in contact with and moves around it.

These factors combine to create a pressure profile across the skin of a building, as shown in **Figure 10.**

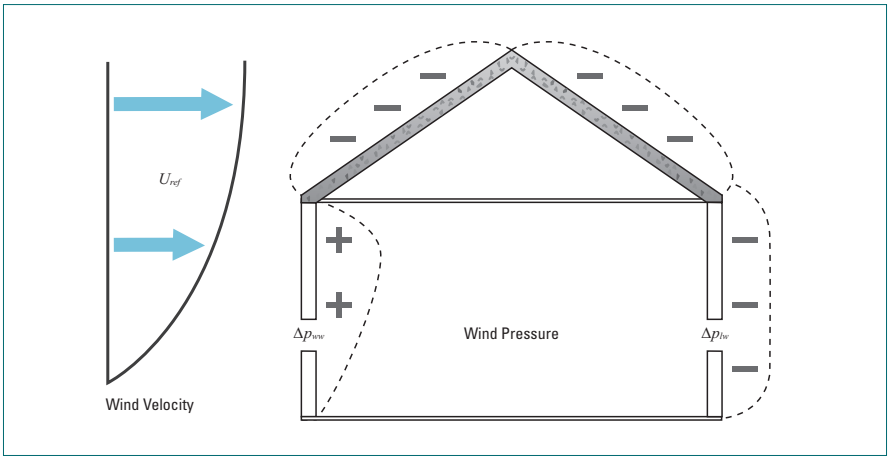

Figure 9: Pressure differential due to the wind

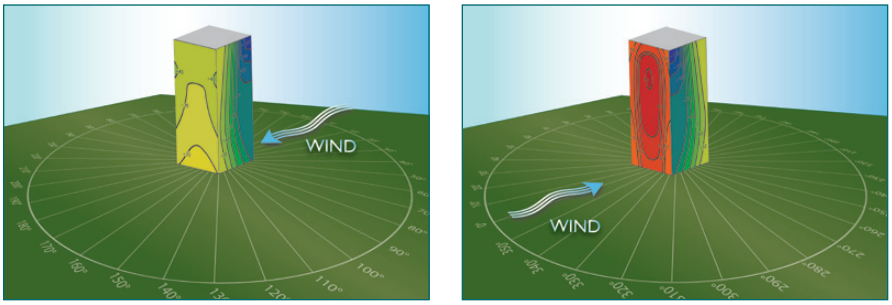

Figure 10: Pressure profile across the skin of a building (Illustration courtesy of CPP Wind Engineering - www.cppwind.com)

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Wind-Driven Flow

If the openings are located on opposite sides of the building, when the wind blows, a cross-flow ventilation pattern where air passes from the high pressure to the low pressure side can be established, as shown in **Figure 11**.

If a building has a single opening in the roof with two openings, one of which faces upwind and the other downwind **(Figure 12)**, the amount of air exchange will be due to the local pressure differential across the opening. In this scenario, although the ventilation will be driven by the wind, the ventilation within the building is localized at the point of the opening. In this regard, the design is not as effective at distributing air throughout the space as a design featuring openings on the sides of the building, though it can be designed in such a way that the total flow rate exchange between the interior and exterior is the same.

Actual determination of the surface pressures is not straightforward. The building itself will induce turbulence, which will vary the actual pressure from the mean to a certain extent. Furthermore, the wind will blow with varying velocity thoughout the day, month and year. Lastly, the actual wind speed realized on site will be largely dependent on the topography (natural and urban) surrounding the building site.

One of the main challenges with wind-driven natural ventilation strategies is that wind tends to be unpredictable. The opportunity to use wind as a single source of pressure drive, and thereby ventilate a space, depends in part on the location of the building and the immediate surroundings. If it is surrounded by other buildings, the characteristics of the wind may be altered and depending on the prevailing wind direction and adjacent structures, the resulting building surface pressures and flow potential can either be enhanced or reduced.

The opportunity to use wind also depends on the design of the building. Cross-flow ventilation, for example, is often practical only for zones that involve open plan areas. Finally, the most challenging aspect of a design for a natural ventilation strategy is often when the outside conditions are hot and still in summer. In these cases, not only can the designer face the likelihood of a building overheating, but he/she also faces the possibility of inadequate supply of fresh air, thereby risking a reduction in the air quality. As a result, many designers seek to also employ buoyancy as a driving force and design for this in combination with the force due to wind.

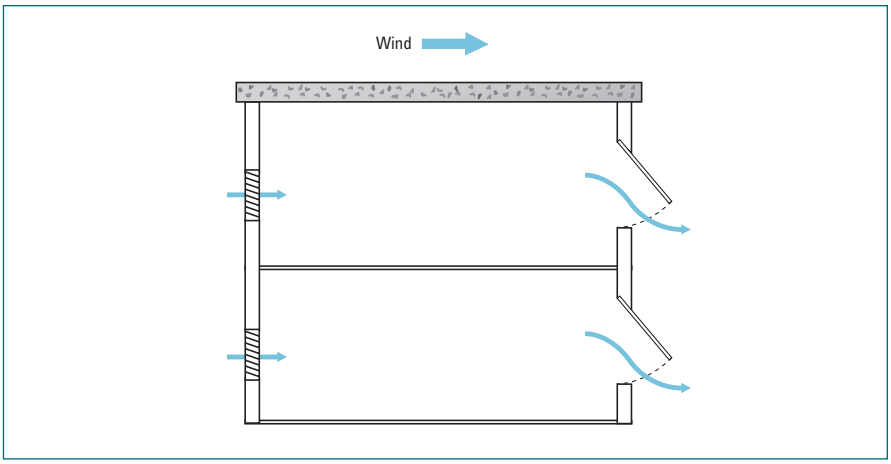

Figure 11: Cross-flow ventilation

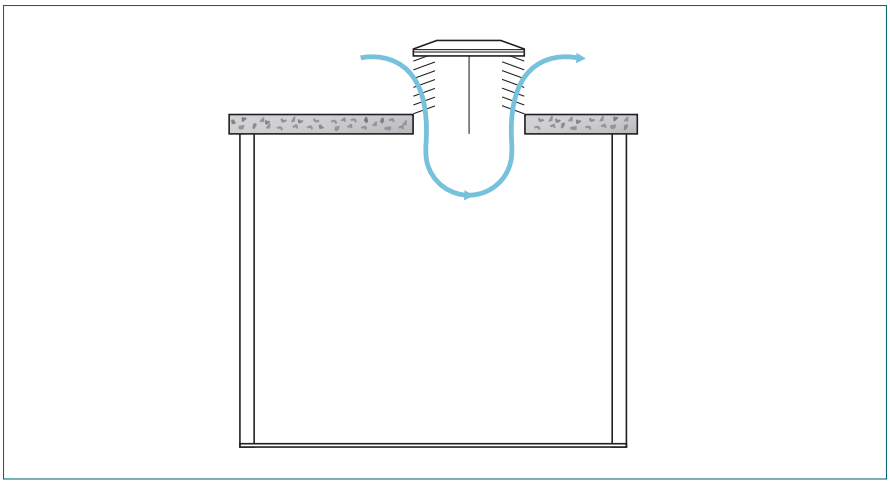

Figure 12: A roof-based vent with upwind and downwind facing sides, but less effective distribution of air within the space

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Example 1: Pressure Differential Due to Wind-Driven Flow (IP)

Consider a building design that incorporates wind-driven cross-flow at an ambient temperature of 70°F

a) Determine the wind-driven pressure difference at 0, 5, 10 and 20 mph.

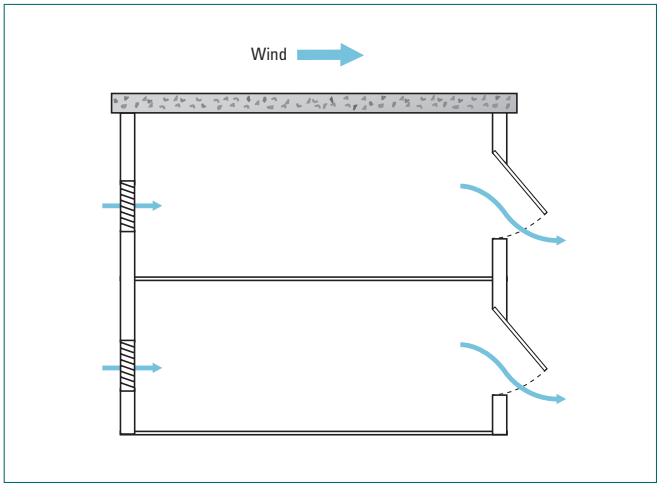
The wind-driven pressure differential is given by equation K1:

$$\Delta p_w = 0.013\left(c_{p-ww} - c_{p-lw}\right)\frac{\rho U_{ref}^2}{2}$$

Typical Design Values (ASHRAE Fundamentals (2009) Chapter 16	
$C_{p\text{-}ww}$	+0.65
$C_{p\text{-}lw}$	-0.65

Density of air at 70°F is approximately 0.075 lb/ft³.

Wind Speed	Δp_w
mph	in. w.c.
0	0.00
5	0.02
10	0.06
20	0.25



Example 1: Pressure Differential Due to Wind-Driven Flow (SI)

Consider a building design that incorporates wind-driven cross-flow at an ambient temperature of 20°C.

a) Determine the wind-driven pressure difference at 0, 10, 20 and 30 km/h.

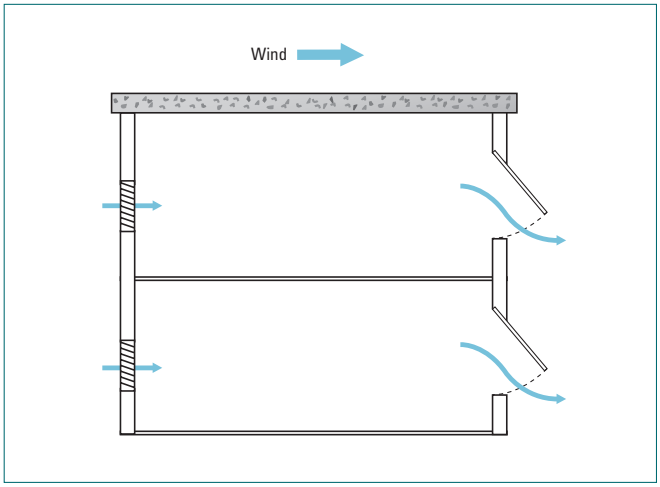
The wind-driven pressure differential is given by equation K1:

$$\Delta p_w = \left(c_{p-ww} - c_{p-lw}\right)\frac{\rho U_{ref}^2}{2}$$

Typical Design Values (ASHRAE Fundamentals (2009) Chapter 16	
$C_{p\text{-}ww}$	+0.65
$C_{p\text{-}lw}$	-0.65

Density of air at 20°C is approximately 1.2 kg/m³.

Wind Speed		Δp_w
km/h	m/s	Pa
0	0.0	0
10	2.8	6
20	5.6	24
30	8.3	54



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Buoyancy-Driven Flow

For much of the year, it is desirable to design for the temperature inside a building to be greater than that outside. It is therefore possible to design a building where the buoyancy force arising from gravity and the difference in air density between the interior and exterior of the building can be used to drive the flow. The primary factor determining the flow potential due to buoyancy can be explained by the hydrostatic pressure.

This variance in the pressure profile inside and outside of the building is the primary driver of the buoyancy-driven flow. When these pressure profiles are super-imposed, the regions of potential inflow and outflow become apparent, as shown in **Figure 14**.

$$\Delta p_{inlet} + \Delta p_{outlet} = \Delta p_s = (\rho_{outside} - \rho_{inside})g\Delta z \qquad \text{K2}$$

Where:

p = the pressure in lb/ft^2 or Pa

ρ = the fluid density in lb/ft^3 or kg/m^3

g = the gravitational acceleration, 32.2ft/s^2 or 9.81m/s^2

z = the difference in height between two points in ft or m

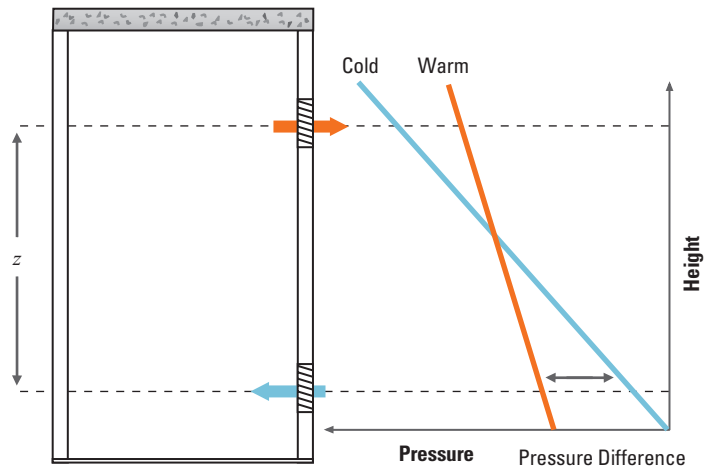

Figure 14: Pressure variations due to height inside and outside of a building

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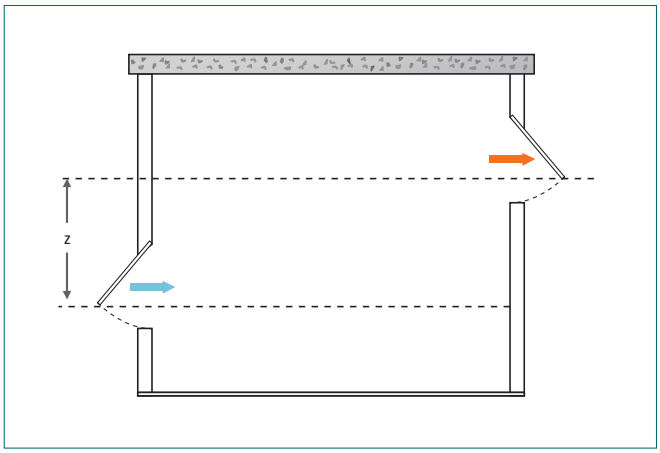
Example 2 - Hydrostatic Pressure (IP)

Consider a 30 foot high atrium with openings to the outside at ground level and roof level. The inside conditions are maintained at 75 °F.

a) Determine the hydrostatic pressure differential at ground level and roof level if the outdoor temperature is 40 °F, 50 °F and 60 °F.



a) The hydrostatic pressure difference can be determined from equation K2:

$$\Delta p_s = (\rho_{outside} - \rho_{inside}) g \Delta z$$

The density can be determined according to the ideal gas law:

$$\rho = \frac{p}{RT}$$

Where:

R = Specific gas constant for dry air 53.35 ft lb_f / lb_m R

T = absolute temperature in R

p = absolute pressure assumed to be 14.7 psi

Determining the values at 40 °F:

$$\Delta p_s = (\rho_{outside} - \rho_{inside}) g \Delta z$$

$$\rho_{outside} - \rho_{inside} = \left(\frac{p}{R\left(T_{outside}\right)} - \frac{p}{R\left(T_{inside}\right)} \right)$$

$$\rho_{outside} - \rho_{inside} = \left(\frac{14.7\ lb_f/in^2\ (144\ in^2/ft^2)}{53.35\ \frac{ft\ lb_f}{lb_m R}\ (40\,°F + 459.67\ R)} - \frac{14.7\ lb_f/in^2\ (144\ in^2/ft^2)}{53.35\ \frac{ft\ lb_f}{lb_m R}\ (75\,°F + 459.67\ R)} \right)$$

$$\rho_{outside} - \rho_{inside} = 0.0052\ lb_m/ft^3$$

$$\Delta p_s = (\rho_{outside} - \rho_{inside}) g \Delta z$$

$$\Delta p_s = 0.0052\ lb_m/ft^3 \left(\frac{slug}{32.2\ lb_m} \right) \left(\frac{lb_f\ s^2}{slug\ ft} \right) 32.2\ ft/s^2\ (30\ ft)$$

$$\Delta p_s = 0.156\ lb_f/ft^2$$

Determining the values at 40 °F:

$$\Delta p_s = (\rho_{outside} - \rho_{inside}) g \Delta z$$

$$0.156\ lb_f/ft^2 \left(0.1935\ \frac{in.w.g.}{lb_f/ft^2} \right) = 0.03\ in.w.g.$$

Calculating:

T_o	T_i	ρ_o	ρ_i	Δp_s
°F	°F	lb_m/ft^3	lb_m/ft^3	in. w.c.
40	75	0.079	0.074	0.03
50	75	0.078	0.074	0.02
60	75	0.076	0.074	0.01

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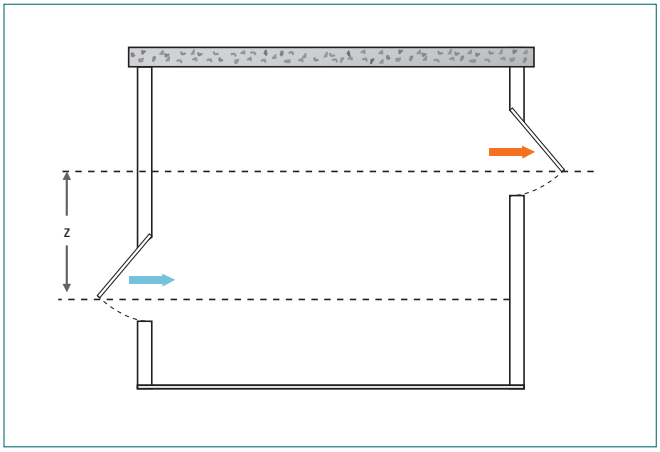
Example 2 - Hydrostatic Pressure (SI)

Consider a 10 meter high atrium with openings to the outside at ground level at roof level. The inside conditions are maintained at 25 °C.

a) Determine the hydrostatic pressure differential at ground level and roof level if the outdoor temperature is 10 °C, 15 °C and 20 °C.



a) The hydrostatic pressure difference can be determined from equation K2:

$$\Delta p_s = (\rho_{outside} - \rho_{inside})g\Delta z$$

The density can be determined according to the ideal gas law:

$$\rho = \frac{p}{RT}$$

Where:
R = Specific gas constant for dry air 287.058 J/KgK
T = absolute temperature in Kelvin
p = absolute pressure assumed to be 101.325 kPa

Determining the values at 10 °C:

$$\Delta p_s = (\rho_{outside} - \rho_{inside})g\Delta z$$

$$\Delta p_s = \left(\frac{p}{R(T_{outside})} - \frac{p}{R(T_{inside})}\right) g\Delta z$$

$$\Delta p_s = \left(\frac{101325\ \text{Pa}}{287.058\ \text{J/kgK}\ (10 + 273.15)} - \frac{101325\ \text{Pa}}{287.058\ \text{J/kg K}\ (25 + 273.15)}\right) 9.81\ \text{m/s}^2\ (10\ \text{m})$$

$$\Delta p_s = 6\ \text{Pa}$$

Calculating the rest of the values:

T_o	T_i	ρ_o	ρ_i	Δp_s
°C	°C	kg/m³	kg/m³	Pa
10	25	1.25	1.18	6
15	25	1.22	1.18	4
20	25	1.20	1.18	2

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Buoyancy-Driven Flow

Between the level of inflow and the level of outflow is a location where the hydrostatic pressure inside the building is equivalent to that outside at the corresponding height. This location, shown in **Figure 16** is known as the neutral level. This level is important as it will determine the height at which inflow ceases, a key design parameter in naturally ventilated buildings.

According to **Figure 16**, air will enter at the base of the building, resulting from the pressure outside being greater than the pressure inside. The thermal gains inside the building will heat the air, ensuring that the density of the air inside is lower than that of the air outside. As a result of this reduced density, the variation of pressure with height inside the building is smaller than that outside the building. Therefore, at the top of the building, the pressure inside is greater than the pressure outside, and air is driven out of the building.

If the size of the openings at the top and bottom of the building are identical, the resistance to flow at each location is the same. Since the mass of the air entering the building has to equal that leaving, the pressure differences between the interior and exterior are the same at the top and the bottom.

Figure 16 highlights a number of key factors for the design of ventilation system. If the lower level and upper level openings are of the same size, the point at which the pressure inside the building is equal to that outside is half way up the building. In this case, if an opening were introduced at the midpoint, then theoretically no flow would occur. In reality, the window created at the mid-height of the building will allow exchange air between the interior and exterior of the building, even in the absence of wind.

Intermediate, occupied floors will need some ventilation (inflow). As a result, it is common to require openings which are located at various heights of a multi-story building. Considering again that the openings at the high level are equal in size to that at low level, then openings below the neutral level, or the lower half of the building, will cause inflow, while those above this level, or the upper half of the building will cause outflow, as shown in **Figure 16**. These openings in the upper half of the building, while promoting air flow and causing an increase in the overall ventilation rate to the building, will not ensure that the occupants are receiving sufficient ventilation air, due to the air escaping the occupied space, rather than entering it.

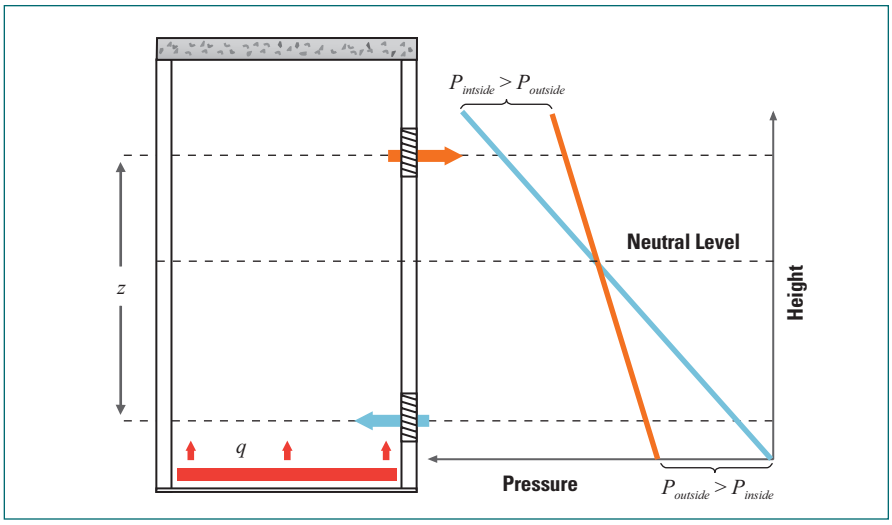

Figure 16: The potential for flow due to pressure variation with height

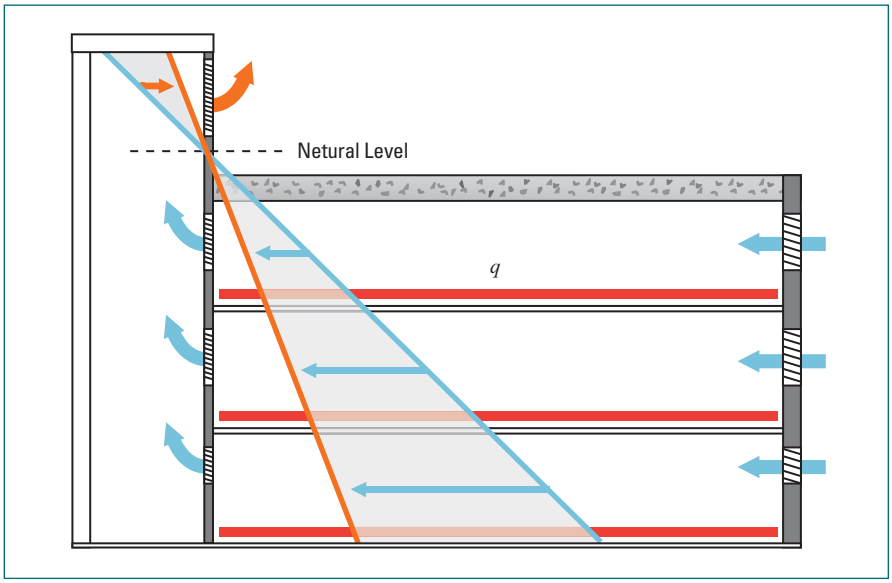

Figure 17: Adjusting the location of the neutral level by adjusting the size of the upper and lower openings

The height of the cross-over point of the pressures in **Figure 16**, or the neutral pressure level, can be altered by changing the relative sizes (and/or number) of the upper and lower openings. If the upper opening is increased relative to the lower one, the neutral pressure level will rise, shown in **Figure 17**. Conversely, larger openings towards the lower level will cause this height to fall.

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Resultant Building Pressure Profile

The pressures due to wind and buoyancy are additive and dependant on the location of the openings, as shown in **Figure 18**.

There are instances and specific conditions where recirculation of indoor air, or ventilation rates, is less than the design air flow rate. These scenarios are most common when the openings in the building cannot be located at the optimal height and face of the building and the flow can become unstable. For these situations, more detailed analysis and CFD or simulation models may be required.

In practice, it is prudent to design a building based on a worst-case conditions in summer of a hot still day with buoyancy as the primary force. The openings should be designed to benefit from wind on hot windy days. Note that when it is cool and windy outside, the potential for draft is increased and therefore careful design of the openings' size and location is required. A good way to accommodate the opening size required on a warm day and to limit the amount of air delivered on a cool day is to use modulating openings.

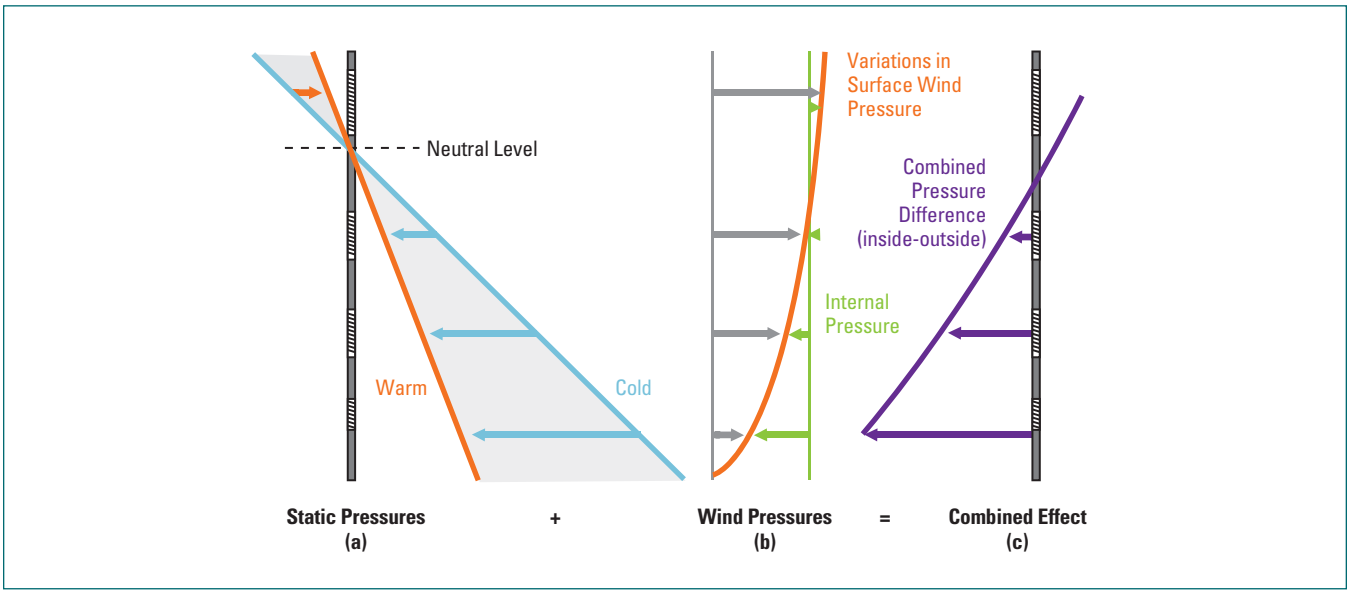

Figure 18: Resulting building pressure profile due to wind and buoyancy

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Design for Summer Operating Mode with Displacement Ventilation

With the unpredictable nature of wind, it is commonly recommended to base the size and location of openings on a scenario where the air flow is purely driven by buoyancy effects. With this approach, the wind will work to assist the natural ventilation, but is not required to achieve the design minimum air flows.

The flow rate through an opening can be expressed by:

$$Q = V_{ave} A_e = V_{ave} C_D A \qquad \text{K3}$$

Where:

Q = the volume flow in ft³/s or m³/s
V_{ave} = the average velocity through the opening in fps or m/s
A_e = the equivalent area in ft² or m²
C_D = the discharge coefficient

This relationship can be rearranged to determine the average velocity:

$$V_{ave} = \frac{Q}{A_e} = \frac{Q}{C_D A} \qquad \text{K4}$$

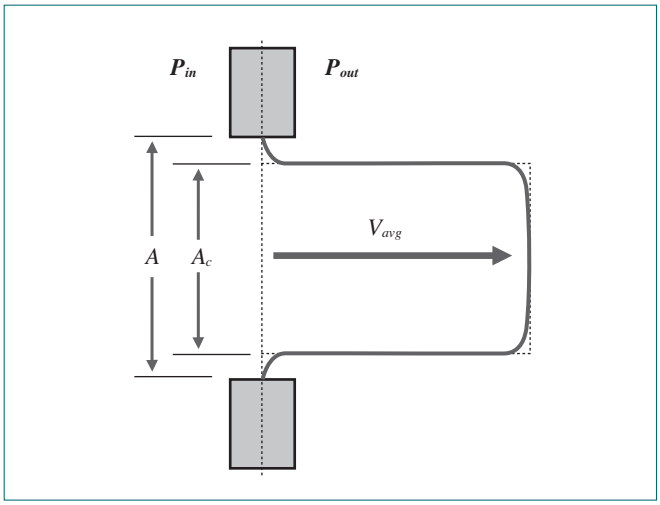

Figure 19: Area and Effective Area of an opening

The average velocity through the opening is related to the differential pressure across the opening by rearranging Bernoulli's equation and combining with equation K4 :

$$\Delta p = \frac{\rho V_{ave}^2}{2} = \frac{1}{2}\rho\left(\frac{Q}{A_e}\right)^2 = \frac{1}{2}\rho\left(\frac{Q}{C_D A}\right)^2 \qquad \text{K5}$$

Substituting Equation K5 for a low and high opening into equation K2 results in a relationship between indoor temperatures and air flow:

$$\frac{1}{2}\rho_{outside}\left(\frac{Q}{C_{D,inlet}A_{inlet}}\right)^2 + \frac{1}{2}\rho_{outside}\left(\frac{Q}{C_{D,outlet}A_{outlet}}\right)^2 = (\rho_{outside} - \rho_{inside})g\Delta z \qquad \text{K6}$$

Defining an equivalent opening area (A*) as follows:

$$\left(\frac{1}{A^*}\right)^2 = \frac{1}{2}\left[\left(\frac{1}{A_{e,1}}\right)^2 + \left(\frac{1}{A_{e,2}}\right)^2\right] = \frac{1}{2}\left[\left(\frac{1}{C_{D,1}A_1}\right)^2 + \left(\frac{1}{C_{D,2}A_2}\right)^2\right] \qquad \text{K7}$$

Allows the simplification of equation K6 to:

$$\left(\frac{Q}{A^*}\right)^2 = gh\left(\frac{\rho_{outside} - \rho_{inside}}{\rho_{outside}}\right) \qquad \text{K8}$$

Using the ideal gas law to get the expression of density in terms of temperature:

$$\frac{\rho_{outside} - \rho_{inside}}{\rho_{outside}} = \frac{T_{outside} - T_{inside}}{T_{outside}} \qquad \text{K9}$$

Where T is the absolute temperature in R or K.

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Design for Summer Operating Mode with Displacement Ventilation

Equation K9 can be substituted into equation K8 resulting in the following useful design equation:

IP
$$Q = 60A^{*}\left(gh\frac{\Delta T}{T_{outside}}\right)^{\frac{1}{2}} \qquad \text{K10}$$

SI
$$Q = A^{*}\left(gh\frac{\Delta T}{T_{outside}}\right)^{\frac{1}{2}} \qquad \text{K10}$$

Where
Q = volume flow in cfm

Combining equation K10 with the specific heat equation:

$$q = \rho c_P Q \Delta T \qquad \text{K11}$$

Results in:

IP
$$q = 60\rho_{outside} c_P A^{*}\left(gh\frac{\Delta T^3}{T_{outside}}\right)^{\frac{1}{2}} \qquad \text{K12}$$

SI
$$q = \rho c_P A^{*}\left(gh\frac{\Delta T^3}{T_{outside}}\right)^{\frac{1}{2}} \qquad \text{K12}$$

Where:
q = the internal load in Btu/h or W
$\rho_{outside}$ = the outdoor air density in lb_m/ft^3 or kg/m^3
c_P = the specific heat coefficient, usually 1.007 kJ/kgK or 0.24 Btu/lb°F
A^{*} = the characteristic area given by equation g
ΔT = the differential between the outdoor temperature and the indoor temperature in °F or K
T = the absolute outdoor air temperature in R or K

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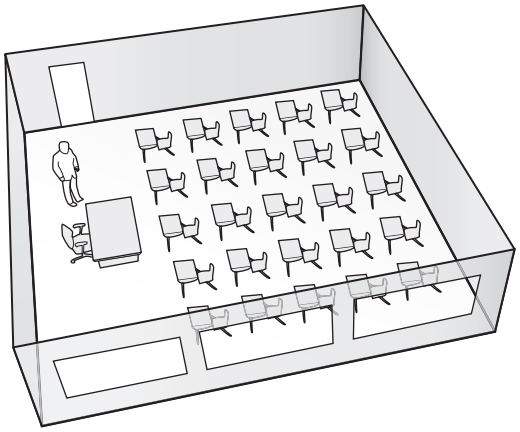
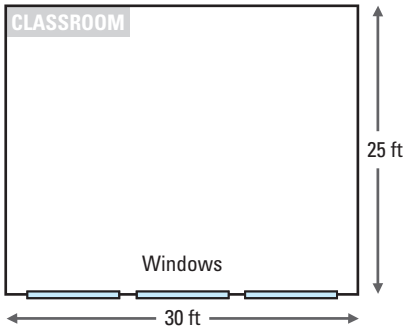
Example 3 - Summer Operating Mode with Displacement Ventilation (IP)





Consider a classroom designed for 25 students and one instructor, three computers with LCD monitors and T8 florescent lighting. The room is 25 ft wide, 30 ft long, and has a ceiling height of 10 ft. There is one exterior wall facing northwest with 100ft² of window and an exposed ceiling. The classroom is occupied from 7 am to 5 pm. High level openings are provided with a center 1 ft below the ceiling.

Determine:

a) The minimum outdoor air ventilation requirement.
b) The flow rate to maintain a space temperature no more than 9 °F above ambient.
c) The required effective areas of the low and high level openings if the low level openings are controlled dampers with a center 1 ft off the floor.
d) The required effective areas of the low and high level openings if the low level openings are operable windows with a center 3 ft off the floor.

Design Considerations

The minimum ventilation shall be determined according to ASHRAE Standard 62.1 (ASHRAE, 2004). The space assumptions include:

- Average load per person is 250 Btu/h
- 3.4 Btu/h/ft² lighting load
- 350 Btu/h per computer
- The average envelope gains are estimated at 50 Btu/h/°F
- For this example, assume solar shading is provided to limit peak solar gain to 8 Btu/h/ft² (floor area)
- The specific heat and density of air for this example will be 0.24 Btu/lb°F and 0.075 lb_m/ft^3, respectively

a) Determine the ventilation rate.

From ASHRAE Standard 62.1, the ventilation rate for educational facilities can be determined from:

$$Q_{oz} = R_p P_z + R_a A_z$$

Where:
Q_{oz} = the ventilation rate in cfm
R_p = the people outdoor air rate, 10 cfm/person
P_z = the zone occupancy
R_a = the area outdoor air rate, 0.12 cfm/ft²
A_z = the zone floor area in ft²

Evaluating the ventilation rate for the classroom:

$$Q_{oz} = \left(10\ {}^{\text{cfm}}/_{person}\right)(26\ \text{occupants}) + \left(0.12\ {}^{\text{cfm}}/_{\text{ft}^2}\right)(25\ \text{ft})(30\ \text{ft}) = 350\ \text{cfm}$$

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Example 3 - Summer Operating Mode with Displacement Ventilation (IP)

b) Air flow rate required to maintain a ΔT of 9 °F.

Design Considerations		
Occupants	26 x 250 Btu/h	6500 Btu/h
Computers	3 x 350 Btu/h	1050 Btu/h
Lighting	3.4 Btu/h/ft² x 25ft x 30 ft	2550 Btu/h
Solar	8 Btu/h/ft² x 25ft x 30 ft	6000 Btu/h
Envelope Loss	50 Btu/h * 9 °F	450 Btu/h
Total		**16550 Btu/h**

$$q = 60\rho c_p Q dT$$

$$Q = \frac{16550\ \text{Btu/h}}{60\ \text{min/h}\left(0.075\frac{\text{lb}}{\text{ft}^3}\right)\left(0.24\frac{\text{Btu}}{\text{lb}\cdot°\text{F}}\right)(9\,°\text{F})} = 1702\ \text{cfm}$$

c) The required effective areas of the low and high level openings if the low level openings are controlled openings with a center 1ft off the floor.

$$Q = 60A^*\left(gh\frac{\Delta T}{t_{outside}+273}\right)^{\frac{1}{2}}$$

$$A^* = \frac{Q}{60}\left(gh\frac{\Delta T}{t_{outside}+273}\right)^{-\frac{1}{2}}$$

$$A^* = \frac{1702\ \text{cfm}}{60\ \text{sec/min}}\left(32.2\ \text{ft/s}^3\,(10\ \text{ft}-1\ \text{ft}-1\ \text{ft})\frac{(9\,°\text{F})}{75\,°\text{F}+459.67}\right)^{-\frac{1}{2}} = 13.58\ \text{ft}^2$$

This A* can then be used to calculate the effective area required:

$$\left(\frac{1}{A^*}\right)^2 = \frac{1}{2}\left[\left(\frac{1}{A_{e,1}}\right)^2+\left(\frac{1}{A_{e,2}}\right)^2\right] = \frac{1}{2}\left[\left(\frac{1}{C_{D,1}A_1}\right)^2+\left(\frac{1}{C_{D,2}A_2}\right)^2\right]$$

Assume that low and high level openings are equal.

$$\left(\frac{1}{A^*}\right)^2 = \frac{1}{2}\left[\left(\frac{1}{A_{e,1}}\right)^2+\left(\frac{1}{A_{e,2}}\right)^2\right]$$

$$A_e = 13.58\ \text{ft}^2$$

d) The required effective areas of the low and high level openings if the low level openings are operable windows with a center 3 ft off the floor.

$$A^* = \frac{1702\ \text{cfm}}{60\ \text{sec/min}}\left(32.2\ \text{ft/s}^3\,(10\ \text{ft}-1\ \text{ft}-3\ \text{ft})\frac{(9\,°\text{F})}{75\,°\text{F}+459.67}\right)^{-\frac{1}{2}} = 15.73\ \text{ft}^2$$

In order to reduce the size of natural ventilation openings required, it is important to maximize the available buoyancy head.

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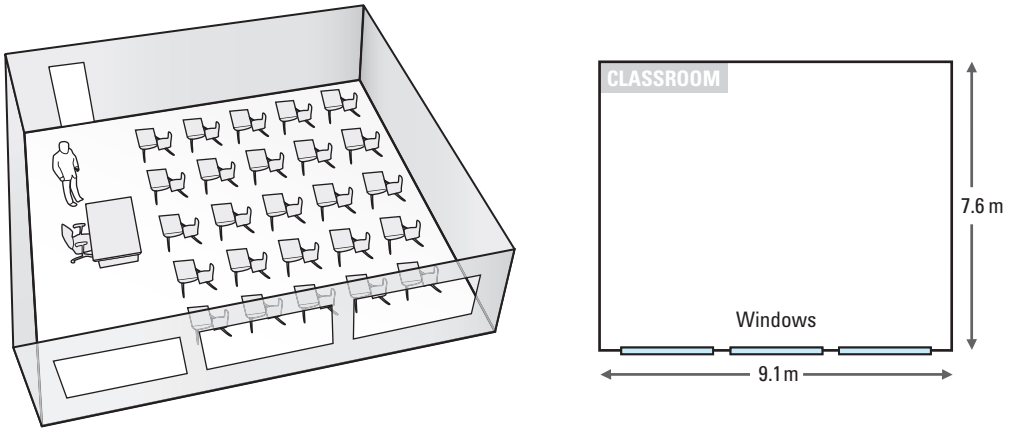
Example 3 - Summer Operating Mode with Displacement Ventilation (SI)



Consider a classroom designed for 25 students and one instructor, three computers with LCD monitors and T8 florescent lighting. The room is 8 m wide, 9 m long, and has a ceiling height of 3 m. There is one exterior wall facing northwest with 10 m² of window and an exposed ceiling. The classroom is occupied from 7 am to 5 pm. High level openings are provided with a center 300 mm below the ceiling.

Determine:

a) The minimum outdoor air ventilation requirement.
b) The flow rate to maintain a space temperature no more than 5 °C above ambient.
c) The required effective areas of the low and high level openings if the low level openings are controlled dampers with a center 300 mm off the floor.
d) The required effective areas of the low and high level openings if the low level openings are operable windows with a center 1000 mm off the floor.

Design Considerations

The minimum ventilation shall be determined according to ASHRAE Standard 62.1 (ASHRAE, 2004). The space assumptions include:

- Average load per person is 75 W
- 10 W/m² lighting load
- 90 W per computer
- The average envelope gains are estimated at 25 W/K
- For this example, assume solar shading is provided to limit peak solar gain to 25 W/m² (floor area)
- The specific heat and density of air for this example will be 1.007 kJ/kgK and 1.2 kg/m³, respectively

a) Determine the ventilation rate.

From ASHRAE Standard 62.1, the ventilation rate for educational facilities can be determined from:

$$Q_{oz} = R_p P_z + R_a A_z$$

Where:
Q_{oz} = the ventilation rate in cfm
R_p = the people outdoor air rate is 5 L/s/person
P_z = the zone occupancy
R_a = the area outdoor air rate is 0.6 L/m²
A_z = the zone floor area in m²

Evaluating the ventilation rate for the classroom:

$$Q_{oz} = (5\ \text{L/s}\,(person))(26\ \text{occupants}) + (0.6\ \text{L/s}\,(\text{m}^2))(8\ \text{m})(9\ \text{m}) = 175\ \text{L/s}$$

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Example 3 - Summer Operating Mode with Displacement Ventilation (SI)

b) Air flow rate required to maintain a ΔT of 5 °C.

Design Considerations		
Occupants	26 x 75 W	1950 W
Computers	3 x 90 W	270 W
Lighting	10 W/m² x 72 m²	720 W
Solar	25 W/m² x 72 m²	1800 W
Envelope Loss	25 W/K x 5 K	125 W
Total		**4615 W**

$$q = \rho c_p Q dT$$

$$Q = \frac{4615\ \text{W}}{\left(1.2\frac{\text{kg}}{\text{m}^3}\right)\left(1007\frac{\text{J}}{\text{kgK}}\right)(5\ ^\circ\text{C})} = 0.754\ \text{m}^3/\text{s} = 754\ \text{L/s}$$

c) The required effective areas of the low and high level openings if the low level openings are controlled openings with a center 300 mm off of the floor.

$$Q = A^*\left(gh\frac{\Delta T}{t_{outside}+273}\right)^{\frac{1}{2}}$$

$$A^* = 754\ \text{L/s}\left(\frac{\text{m}^3}{1000\ \text{L}}\right)\left(9.81\ \text{m/s}^2(3\ \text{m}-0.3\ \text{m}-0.3\ \text{m})\frac{(5\ \text{K})}{25\ ^\circ\text{C}+273\ \text{K}}\right)^{-\frac{1}{2}} = 1.15\ \text{m}^2$$

This A* can then be used to calculate the effective area required:

$$\left(\frac{1}{A^*}\right)^2 = \frac{1}{2}\left[\left(\frac{1}{A_{e,1}}\right)^2 + \left(\frac{1}{A_{e,2}}\right)^2\right] = \frac{1}{2}\left[\left(\frac{1}{C_{D,1}A_1}\right)^2 + \left(\frac{1}{C_{D,2}A_2}\right)^2\right]$$

Assume that low and high level openings are equal.

$$\left(\frac{1}{A^*}\right)^2 = \frac{1}{2}\left[\left(\frac{1}{A_{e,1}}\right)^2 + \left(\frac{1}{A_{e,2}}\right)^2\right]$$

$$A_e = 1.15\ \text{m}^2$$

d) The required effective areas of the low and high level openings if the low level openings are operable windows with a center 1000 mm off the floor. This time A * is calculated directly from equation I.

$$A^* = 754\ \text{L/s}\left(\frac{\text{m}^3}{1000\ \text{L}}\right)\left(9.81\ \text{m/s}^2(3\ \text{m}-0.3\ \text{m}-1\ \text{m})\frac{(5\ \text{K})}{25\ ^\circ\text{C}+273\ \text{K}}\right)^{-\frac{1}{2}} = 1.43\ \text{m}^2$$

In order to reduce the size of natural ventilation openings required, it is important to maximize the available buoyancy head.

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Incorporating Thermal Mass

Every building has thermal mass, whether it is designed explicitly into the structure in the form of exposed concrete, or just through the normal building materials and finishes. It is often useful to leverage this mass in warm seasons in order to offset the total cooling load. Unless the design takes a mixed-mode approach to a naturally ventilated building, the cooling capacity of the system is entirely dependent on the outdoor temperature and the design of the building openings. If an evaluation of what the indoor conditions are expected to be indicates that the building may exceed the limits in temperature presented in the previous section, the designer would choose to either:

1. Increase the size of the openings, allowing more air into the buildings and thereby increasing the cooling capacity.
2. Leverage the thermal mass through night cooling to offset the peaks, essentially storing the coolness of the night air to be used during the occupied hours.

Thermal mass refers to the ability of building material to absorb heat. On warm days, as internal temperatures rise, the building material absorbs heat, reducing further rises in the internal temperatures.The heat is then purged with lower temperature night air from the building material when the space is not occupied.

In addition, the designer may elect to use night cooling of the building mass as a way to extend the climatic region deemed appropriate for natural ventilation or in order to optimize the required size of openings as a part of the design phase.

The use of thermal mass within a building can provide significant benefits in terms of both thermal comfort and energy use. In summer, if the building is cooled at night, the thermal mass can absorb heat gains within the space, which leads to a cooler internal air temperature. In addition, the exposed thermal mass can absorb radiation, thus allowing occupants to benefit from both lower air temperatures and radiative cooling.

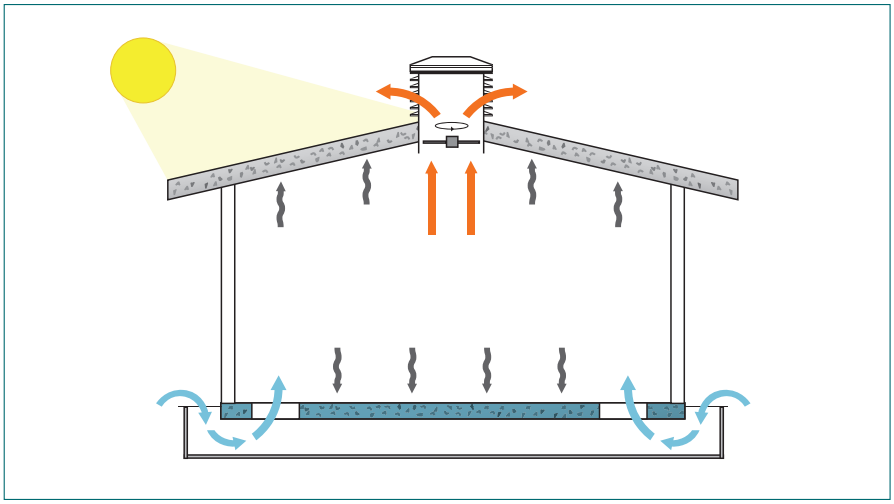

Figure 20: Charging the daytime building mass

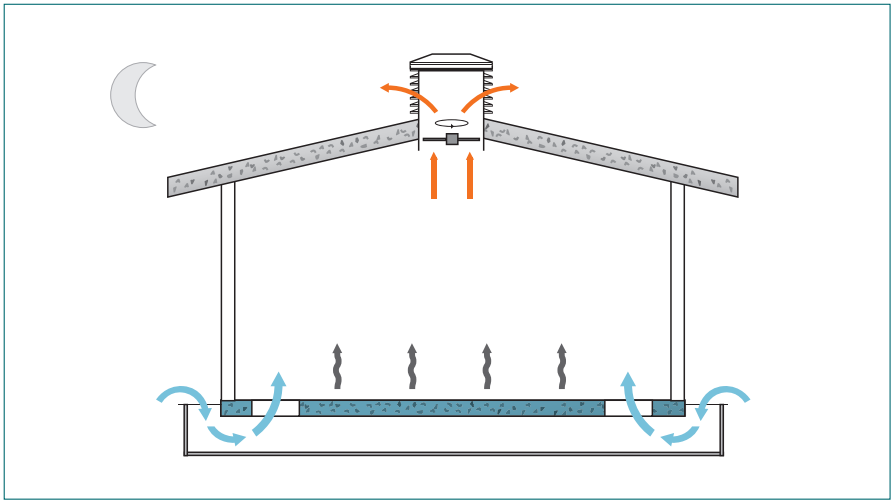

Figure 21: Charging the nighttime building mass

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Incorporating Thermal Mass

In order to maximize the cooling from exposed thermal mass, the flow rate of cool, night air through the room should be designed to cool the structure / mass down to the minimum permissible level. This minimum level, or the temperature of the mass at the start of the following day, is limited by the lower bound of acceptable indoor temperatures at the time of occupancy. As a result, the building control systems should cease the night cooling function once the temperature of the thermal mass has reached its target level.

Figure 22 shows the typical temperature variation and ventilation flow strategy for a thermally massive building. During the occupied period, the ventilation rate needs provide the occupants with sufficient fresh air to meet the minimum ventilation requirements and keep CO_2 levels down to the required level. If the exterior temperature is less than the interior temperature and the interior temperature is greater than the target minimum, the amount of air brought in should be increased to balance internal heat gains and ensure the room is not too cold. When the exterior temperature exceeds the interior temperature and the interior temperature is higher than the target minimum, the ventilation rate should be reduced to the minimum required ventilation rate. The thermal mass will then absorb heat gains in the space.

When the heat absorptive capacity of the thermal mass has decreased or the external temperature drops below the interior temperature, the ventilation rate should be maximized as long as the interior temperature is above the target minimum. As the building interior temperature approaches the minimum target, the flow rate is decreased to prevent overcooling. During the unoccupied period the ventilation rate can be maximized to cool the thermal mass in preparation for the next day.

One of the challenges of having thermal mass within the space is that the cooling capacity of the thermal mass is used throughout the summer day and by late afternoon, when it is perhaps most needed, the cooling capacity of the thermal mass can be depleted. Therefore, an alternative strategy, in which a thermally massive plenum is located outside of the space, can be used. Since the thermal mass is typically heavy, it is often located at the base of the building under the occupied spaces rather than at a high level. If there is no wind to create a negative pressure at roof level and draw air up into the occupied zone from the plenum, there would be no natural upflow because the air inside the space is denser than that outside. Therefore, in this mode, fan-assisted ventilation is required, with fans typically located in the roof stacks, for example. The fan power needed is actually not that significant relative to mechanical cooling. The use of thermal mass in this manner requires a hybrid ventilation scheme as described. However, the benefits of free cooling, rather than mechanically derived cooling, can be significant and the strategy is in use in many buildings.

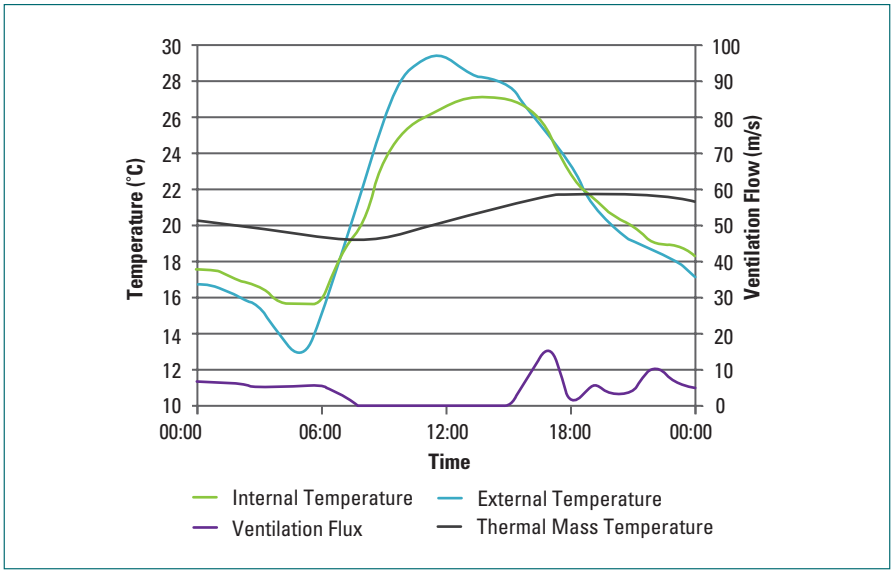

Figure 22: Example temperature variation and natural ventilation strategy over the course of a day in a thermally massive office.

Description	Internal Construction	Active Thermal Capacity Wh/Km²
Very Light	Light walls, floors and ceilings. e.g. skeleton with boards, with any heavy structures	40
Light	Some heavy structure. e.g. concrete slab with wooden floor or light-weight concrete walls	80
Heavy	Several heavy structures. e.g. concrete slab with clinker and brick or clinker concrete walls	120
Very Heavy	Heavy walls, floors and ceiling made by concrete, brick or clinker	160

Table 1: Active thermal capacity per m² gross floor area (taken from [151])

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimetre.

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Dynamic Thermal Models

In the design of natural ventilation systems, it is important that a dynamic analysis is undertaken rather than the steady-state example shown in Example 3. This will allow the effects of thermal mass to be included along with more accurate predictions of coincident solar gain and external temperature.

From the SI version of Example 3, it was established that an effective area of 0.98 m^2 (10.5 ft^2) was required to maintain internal temperatures no more than 5 °C (9 °F) above external temperatures when the buoyancy head is 2.4 m (7.9 ft).

Assuming the classroom discussed in the previous example is located in Seattle, these results can be used to calculate the number of occupied hours for which internal temperatures exceed 28 °C (82 °F) simply by using the local weather file to establish the predicted number of occupied hours where 23 °C (73 °F) is exceeded.

Using hourly weather data for Seattle, we find that the number 189 occupied hours exceed 23°C (73 °F). With a design ΔT of 5 °C (9 °F), this would translate to around 189 hours exceeding 28 °C (82 °F) internally. This implies a sizeable period of the year where there is risk of overheating.

The design of the classroom could be modified to take advantage of night cooling by incorporating a large concrete slab ceiling with the following properties:

This type of analysis is not possible using the steady-state calculation in the previous example. The dynamic thermal model calculates the heat transfer between the room and the mass at 7 minute intervals throughout the summer and calculates the resulting temperature of each component. The calculations use the same basic equations described in Example 3, but include an additional term for the heat transfer between the air and thermal mass when calculating internal gain. The results described below are based on a proprietary model, but many software packages are available for this type of analysis.

Using the parameters established in the previous example, 63 occupied hours are predicted to exceed 28 °C (82 °F), as shown in **Figure 23**. This model demonstrates the importance of including dynamic effects such as thermal mass to allow designs to be refined and use opening areas which are practical for installation in a space. An additional consideration is the intended use of the space. Modeling estimates that 60 of the hours exceeding 28 °C occur in August when schools are on vacation and the classroom is less likely to be used.

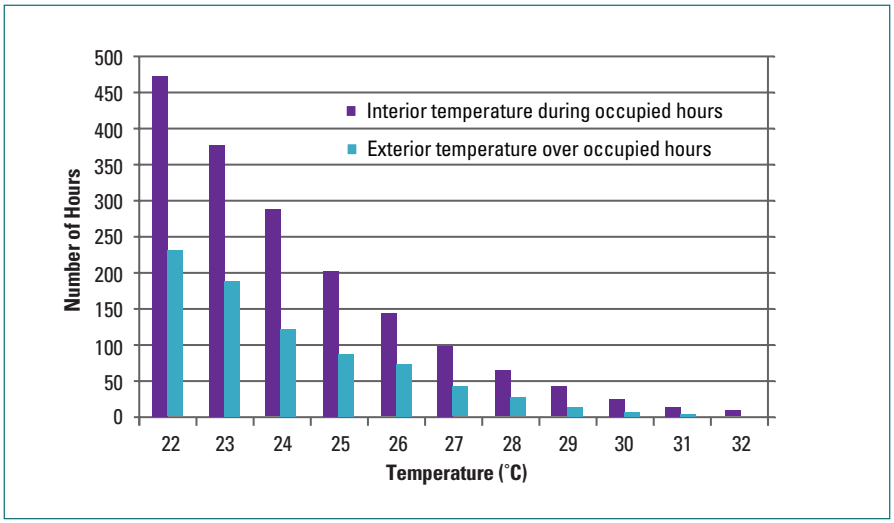

Figure 23: Results of dynamic thermal modelling

Concrete Slab Properties		
Density, ρ	2400 kg/m^3	150 lb/ft^3
Specific heat capacity, c_p	800 kJ/kg K	191 Btu/lb°F
Surface heat transfer coefficient, H	4.3 W/m^2 K	0.21 Btu/ft^2°F

Table 2: Concreate slab properties

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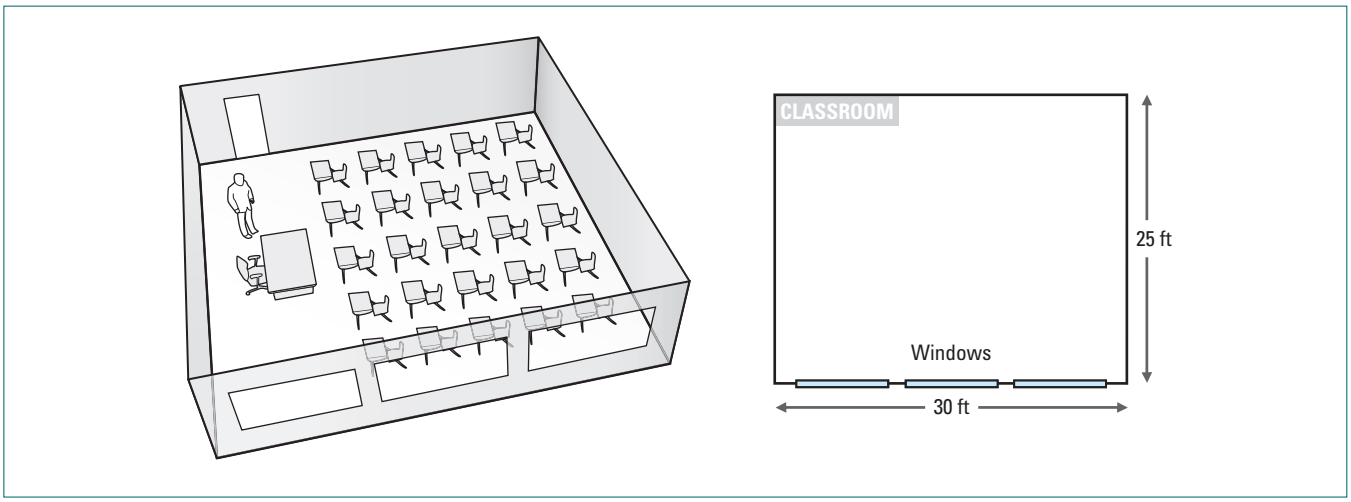
All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimetre.

Example 4 - Winter Operating Mode with Displacement Ventilation (IP)



Consider the classroom presented in Example 3 during winter and using the same natural ventilation system. The low level openings will require an integrated hot water heater to temper the incoming air during the winter months. The supply temperature into the space shall be no less than 11 °F below the space temperature to avoid drafts. For simplicity, a well-insulated space is assumed to eliminate conduction losses.

Determine:

a) The external temperature at which internal gains (excluding solar) balance heat losses from the space at a maintained room temperature of 72 °F at the minimum required air flow rate. This is the heat balance temperature.
b) The amount of energy required to preheat exterior air at the heat balance temperature supplied to the minimum supply air temperature.
c) The resulting air temperature of the space with the minimum ventilation air flow rate preheated to the supply temperature.
d) The volume flow rate of air required to maintain an internal temperature of 72 °F using preheated air.
e) The increase in heating energy consumption needed to maintain the space temperature.

Design Considerations

From Example 3 - Summer Operatring Mode:

Design Considerations	
Density, ρ	0.075 lb/ft^3
Specific heat capacity, c_p	1.02 Btu/lb°F
Total internal load (excluding solar)	10550 Btu/h
Ventilation requirement	350 cfm

a) Heat balance temperature.

Heat losses:

$$q_{loss} = 60\rho c_p Q_{air} \Delta T_{air}$$

Equate heat losses to the heat gain to establish the heat balance temperature.

$$T_{room} - T_{outside} = \frac{q_{gain}}{60\rho c_p Q_{air}}$$

$$T_{outside} = T_{room} - \frac{q_{gain}}{60\rho c_p Q_{air}}$$

$$T_{outside} = 72\ °\text{F} + \frac{10550\ \text{Btu/h}}{(60\ \text{min/h})(0.075\ \text{lb/ft}^3)(0.24\ \text{Btu/lb}\ °\text{F})(350\ \text{cfm})} = 44\ °\text{F}$$

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Example 4 - Winter Operating Mode with Displacement Ventilation (IP)

b) Amount of energy required to preheat exterior air at the heat balance temperature supplied to the minimum supply air temperature.

With air entering directly adjacent to occupants, it is important to ensure that the supply air temperature is not so low as to cause draft. As in displacement ventilation systems, it is good practice to limit the temperature of air supplied in the occupied zone to a maximum of 11°F below the set-point.

Using equation K11:

$$q_{s,\,air} = 60\rho c_p Q_{air}\left(T_{air,\,supply} - T_{air,\,outside}\right)$$

$$q_{s,\,air} = \left(60\ \text{min/h}\ \left(0.075\ \text{lb/ft}^3\right)\left(0.24\ \text{Btu/lb °F}\right)\left(350\ \text{cfm}\right)\right)\left(61\ \text{°F} - 44\ \text{°F}\right) = 8316\ \text{Btu/h}$$

c) Resulting air temperature of the space with the minimum ventilation air flow rate preheated to the supply temperature.

Effective cooling due to ventilation air is given by equation K11:

$$q_{s,air} = 60\rho c_p Q_{air}\Delta T_{air}$$

Where:

$$\Delta T_{air} = T_{room} - T_{supply}$$

Equating these terms leads to the energy balance:

$$\rho c_p Q_{air}\Delta T_{air} = q_{heat}$$

$$T_{room} = T_{\text{sup}\,ply} + \frac{q_{heat}}{60\rho c_p Q_{air}}$$

$$T_{room} = 61\ \text{°F} + \frac{10550\ \text{Btu/h}}{\left(60\ \text{min/h}\right)\left(0.075\ \text{lb/ft}^3\right)\left(0.24\ \text{Btu/lb °F}\right)\left(350\ \text{cfm}\right)} = 89\ \text{°F}$$

This suggests that an overheating problem exists in winter and higher air flows are required to cool space.

d) Volume flow rate of air to maintain a space temperature of 72°F using preheated air.

Again consider equation K11:

$$q_{s,air} = 60\rho c_p Q_{air}\Delta T_{air}$$

Equating to the space heat gain gives:

$$Q_{air} = \frac{q_{heat}}{60\rho c_p \Delta T_{air}}$$

$$Q_{air} = \frac{10550\ \text{Btu/h}}{\left(60\ \text{min/h}\right)\left(0.075\ \text{lb/ft}^3\right)\left(0.24\ \text{Btu/lb °F}\right)\left(11\ \text{°F}\right)} = 888\ \text{cfm}$$

e) Increase in heating energy consumption needed to maintain the space temperature.

$$q_{heating} = 60\rho c_p Q_{air}\Delta T_{air}$$

$$q_{heating} = \left(60\ \text{min/h}\ \left(0.075\ \text{lb/ft}^3\right)\left(0.24\ \text{Btu/lb °F}\right)\left(888\ \text{cfm}\right)\right)\left(61\ \text{°F} - 44\ \text{°F}\right) = 16304\ \text{Btu/h}$$

This indicates that supplying preheated air in this way has caused an increase in heat energy consumption in order to prevent overheating of the space.

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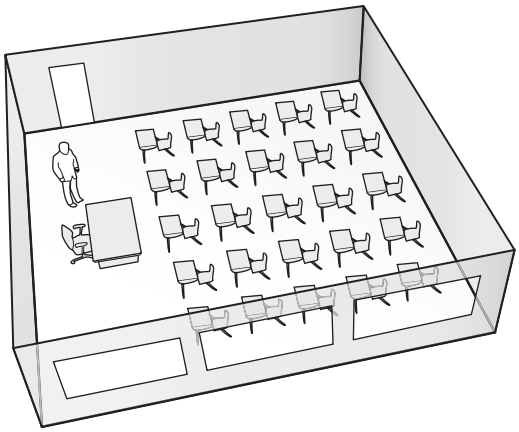
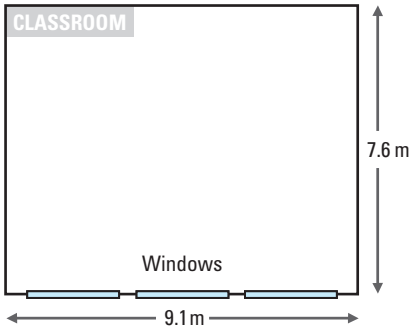
Example 4 - Winter Operating Mode with Displacement Ventilation (SI)





Consider the classroom presented in Example 3 during winter and using the same natural ventilation system. The low level openings will require an integrated hot water heater to temper the incoming air during the winter months. The supply temperature into the space shall be no less than 6 °C below the space temperature to avoid drafts. For simplicity, a well-insulated space is assumed to eliminate conduction losses.

Determine:

a) The external temperature at which internal gains (excluding solar) balance heat losses from the space at a maintained room temperature of 22 °C at the minimum required air flow rate. This is the heat balance temperature.
b) The amount of energy required to preheat exterior air at the heat balance temperature supplied to the minimum supply air temperature.
c) The resulting air temperature of the space with the minimum ventilation air flow rate preheated to the supply temperature.
d) The volume flow rate of air required to maintain an internal temperature of 22 °C using preheated air.
e) The increase in heating energy consumption needed to maintain the space temperature.

Design Considerations

From Example 3 - Summer Operatring Mode:

Design Considerations	
Density, ρ	1.2 kg/m^3
Specific heat capacity, c_p	1.007 kJ/kgK
Total internal load (excluding solar)	2815 W
Ventilation requirement	175 L/s

a) Heat balance temperature.

Heat losses:

$$q_{loss} = \rho c_p Q_{air} \Delta T_{air}$$

Equate heat losses to the heat gain to establish the heat balance temperature.

$$T_{room} - T_{outside} = \frac{q_{gain}}{\rho c_p Q_{air}}$$

$$T_{outside} = T_{room} - \frac{q_{gain}}{\rho c_p Q_{air}}$$

$$T_{outside} = 22\ °C - \frac{2815\ W}{(1.2\ kg/m^3)(1.007\ kJ/kg\ °C)(175\ L/s)} = 8.7\ °C$$

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimetre.

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Example 4 - Winter Operating Mode with Displacement Ventilation (SI)

b) Amount of energy required to preheat exterior air at the heat balance temperature supplied to the minimum supply air temperature.

With air entering directly adjacent to occupants, it is important to ensure that the supply air temperature is not so low as to cause draft. As in displacement ventilation systems, it is good practice to limit the temperature of air supplied in the occupied zone to a maximum of 6 °C below the set-point.

Using equation K11:

$$q_{s,\,air} = \rho c_p Q_{air}\left(T_{air,\,supply} - T_{air,\,outside}\right)$$

$$q_{s,\,air} = (1.2\ \text{kg/m}^3)(1007\ \text{J/kgK})(0.175\ \text{m}^3/\text{s})(16\ °\text{C} - 8.7\ °\text{C}) = 1607\ \text{W}$$

c) Resulting air temperature of the space with the minimum ventilation air flow rate preheated to the supply temperature.

Effective cooling due to ventilation air is given by equation K11:

$$q_{s,air} = \rho c_p Q_{air} \Delta T_{air}$$

Where:

$$\Delta T_{air} = T_{room} - T_{supply}$$

Equating these terms leads to the energy balance:

$$\rho c_p Q_{air} \Delta T_{air} = q_{heat}$$

$$T_{room} = T_{supply} + \frac{q_{heat}}{\rho c_p Q_{air}}$$

$$T_{room} = 16\ °\text{C} + \frac{2815\ \text{W}}{(1.2\ \text{kg/m}^3)(1.007\ \text{kJ/kg °C})(175\ \text{m}^3/\text{s})} = 29\ °\text{C}$$

This suggests that an overheating problem exists in winter and higher air flows are required to cool space.

d) Volume flow rate of air to maintain a space temperature of 22 °C using preheated air.

Again consider equation K11:

$$q_{s,air} = \rho c_p Q_{air} \Delta T_{air}$$

Equating to the space heat gain gives:

$$Q_{air} = \frac{q_{heat}}{\rho c_p \Delta T_{air}}$$

$$Q_{air} = \frac{2815\ \text{W}}{(1.2\ \text{kg/m}^3)(1.007\ \text{kJ/kg °C})(22\ °\text{C} - 16\ °\text{C})} = 0.385\ \text{m}^3/\text{s} = 385\ \text{L/s}$$

e) Increase in heating energy consumption needed to maintain the space temperature.

$$q_{heating} = \rho c_p Q_{air} \Delta T_{air}$$

$$q_{s,\,air} = (1.2\ \text{kg/m}^3)(1007\ \text{J/kg °C})(0.385\ \text{m}^3/\text{s})(16\ °\text{C} - 8.7\ °\text{C}) = 3396\ \text{W}$$

This indicates that supplying preheated air in this way has caused an increase in heat energy consumption in order to prevent overheating of the space.

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Natural Mixing Ventilation

The previous example has demonstrated the fundamental problem with the up-flow displacement ventilation strategy in winter. If there are occupants in the vicinity of the low level vents, they will be subjected to cold drafts when the exterior temperature falls below the range of 15-18°C (59-64°F). One design option that can be used to overcome this problem is preheating the incoming air at the low level inlet opening. However, the previous example showed that the required flow rate of air to provide sufficient cooling for the removal of heat gains must be increased above the minimum required ventilation rate. The higher flow rate results in even higher heating demands to temper the incoming air so that it can then be exhausted out of the space to avoid overheating of the room.

It was shown that the internal heat gains will provide the heat required for incoming ventilation air when the external temperature exceeds the heat balance point of the room. A more effective strategy is to use the existing heat loads in the room to preheat the incoming fresh air and prevent cold draughts. Mixing the incoming air with room air at a high level will prevent the exposure of the occupants to drafts, while providing fresh air for balancing the internal heat loads.

When the ventilation opening is at a high level in a space, it may be apparent that there is no clear buoyancy head driving the ventilation flow. If the interior is warmer than the exterior, an exchange flow will develop. Hot air rises to leave the space and is replaced by cold fluid entering the room. Cold air will fall through the space as a descending turbulent plume. As the turbulent plume descends, it entrains warm interior air around it from the space, causing mixing as it falls. The temperature and volume of fluid in the plume will increase as it descends (Fitzgerald and Woods, 2007). A common analogy for this mode is to imagine an upside down volcano; cold plumes fall downwards rather than hot plumes erupting upwards. This mode of ventilation is natural mixing ventilation. When the opening is at a high level above the occupied zone, the incoming external air will be well mixed with the warm interior air by the time it has reached the occupants. The occupants do not experience cold drafts as in the case with façade openings where heating would be required. Using the mixing ventilation strategy allows the heat generated within the space to be utilized for tempering the incoming fresh air to prevent drafts while providing fresh air and required cooling the space when the external temperature is above the balance temperature. The balance temperature is the temperature at which the internal heat loads are no longer sufficient to offset the cooling effect of the outdoor air at the minimum ventilation flow rate. When external temperatures are below the heat balance point, additional heating will be required, but does not need to be supplied to the incoming air.

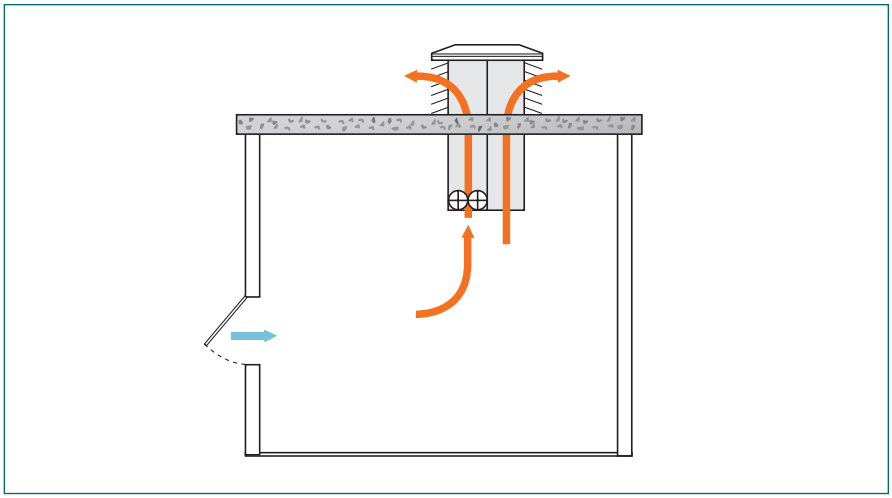

Figure 24: The potential for flow due to pressure variation with height

In many buildings, floor to ceiling heights are limited, which reduces the amount of mixing that can occur. The temperature of the plume may not reach a comfortable temperature at the occupied zone. In this scenario, mechanical assistance is required to increase mixing ratios, as shown in **Figure 24**.

A design based only on mixing ventilation will typically provide less ventilation for the same total opening area than a displacement scheme because the effective buoyancy head to drive the flow will be less. This is not an issue at low external temperatures because smaller flow rates are required to maintain comfortable internal temperatures. Higher external temperatures require higher flow rates, in which case it is preferable to switch back to upflow displacement ventilation by opening a window. This changeover in strategy typically occurs in the range of 16-18°C, when cold drafts are no longer a concern.

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimetre.

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Dynamic Thermal Analysis of Natural Mixing Ventilation

The energy savings from mixing ventilation over upflow displacement systems in winter can be analyzed using dynamic thermal analysis. Using the same classroom from SI Examples 3 and 4 the heating energy required with varying external temperature has been calculated for the following cases:

1. Displacement ventilation with preheating of air inlet at a low level.
2. Mixing ventilation with air supplied at a high level.

As shown in **Figure 25**, mixing ventilation has a lower heating power demand and does not require heating until the balance temperature is reached. In order to ensure drafts are avoided, a small amount of electrical energy is used in the mixing strategy to increase the amount of entrainment. It should be noted that this is a space with high heat gains and dense occupancy typical of many commercial spaces.

The number of occupied hours for a given corresponding external air temperature during winter in Seattle are shown in **Figure 26**. The number of hours below 16 °C (61 °F) is significant.

By combining the number of occupied hours at a given temperature with the energy usage at that temperature, the predicted heated energy consumption throughout the heating season can be estimated based on full occupancy.

The mixing ventilation strategy clearly has the potential to reduce energy consumption, as shown in **Figure 27**.

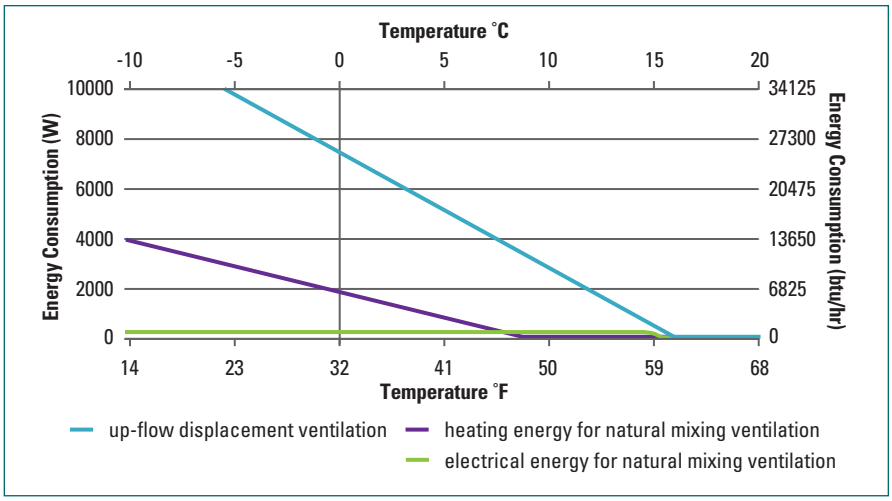

Figure 25: Comparison of natural mixing ventilation with up-flow displacement ventilation

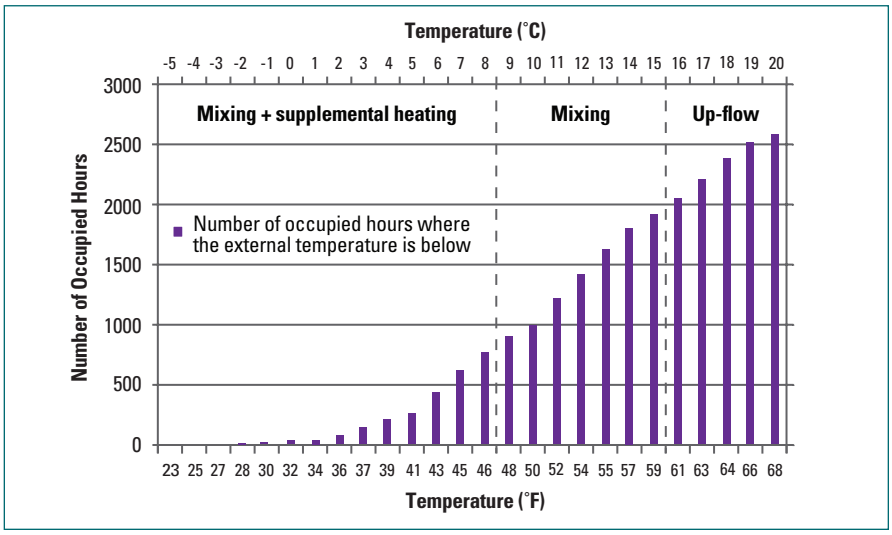

Figure 26: Number of occupied hours vs temperature

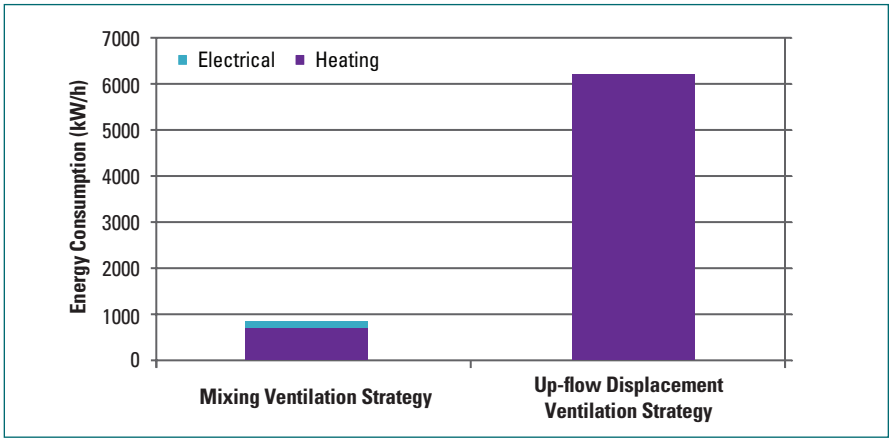

Figure 27: Energy comparison of mixing ventilaion vs. natural ventilation strategies

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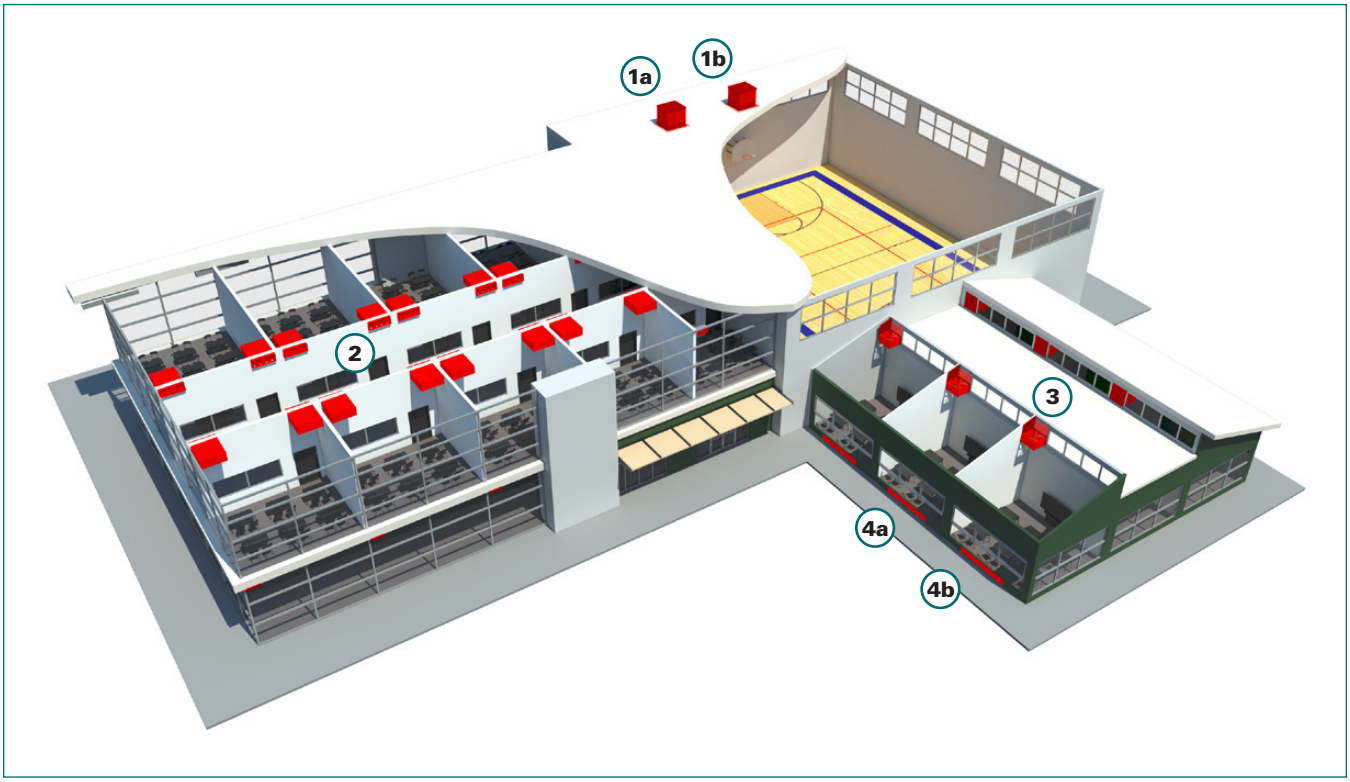
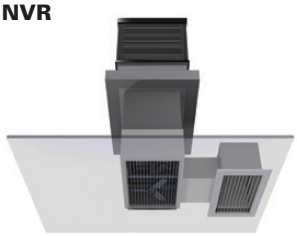
All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimetre.

Product Types

A variety of natural ventilation products are available to suit the particular building architecture and location.

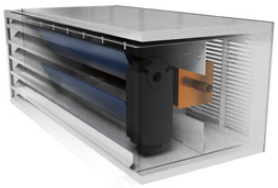


1a. NVR

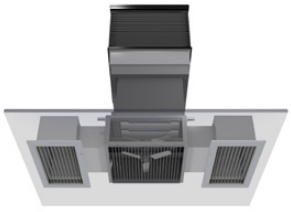


2. NVA

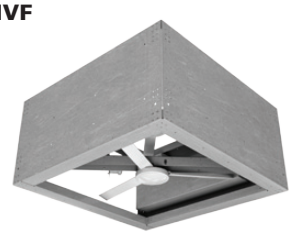
4a. NVT



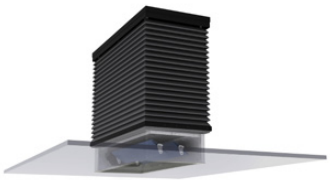
1b. NVS



3. NVF



4b. NVE



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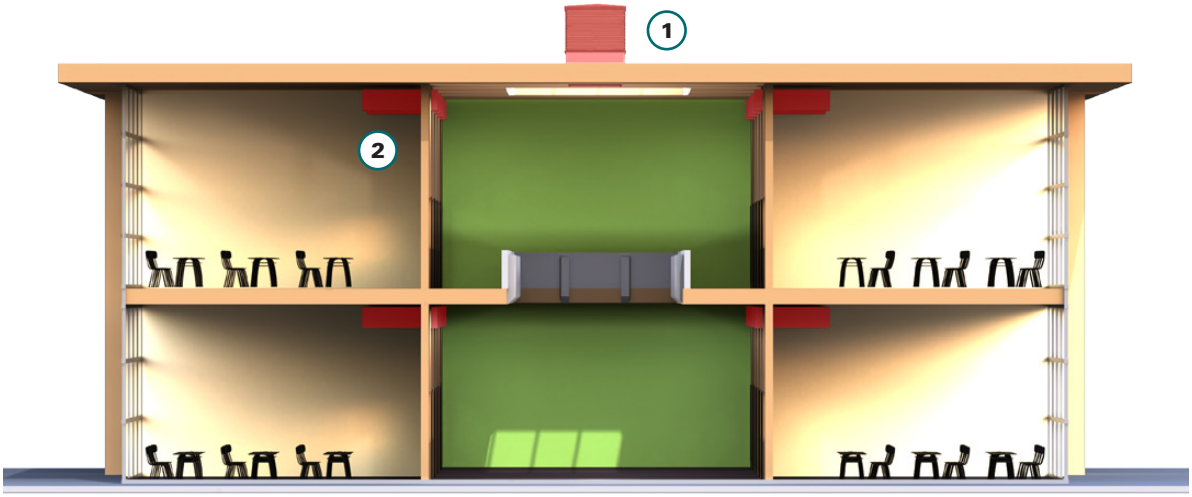
Product Types



1. NVR/NVS Roof Stack Unit
The NVR series are ideally suited to schools or other areas with high internal occupancies and heat gains. The units are sized to provide sufficient summertime ventilation for cooling and provide required ventilation to limit room CO_2 levels.

The NVS series are larger than the NVR series and are suited for spaces with high internal heat gains. Multiple units are often required for large spaces to distribute fresh air around the space and provide ventilation. See **Figures 28 and 29**.

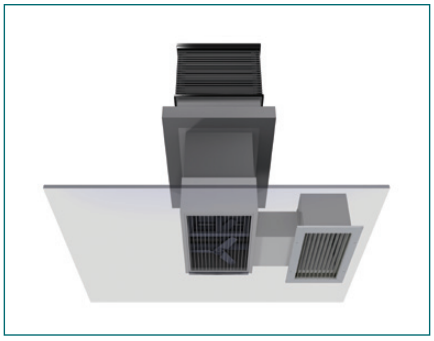


Figure 28: NVR Roof Stack Unit

Figure 29: NVR Roof Stack Unit

2. NVA Atrium Mounted Unit
The NVA Series unit has been developed for low energy ventilation of single or multi-storey buildings where the rooms are connected to a central atrium. A pair of NVA units exchange room air with the atrium for ventilation and cooling requirements. See **Figures 30 and 31**.

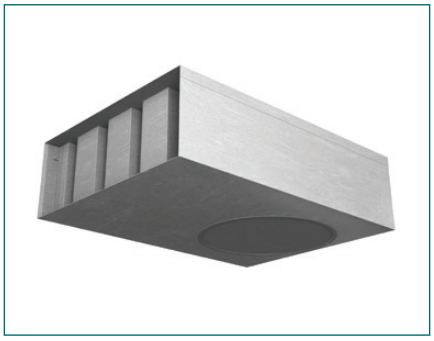


Figure 30: NVA Atrium Mounted Unit

Figure 31: NVA Atrium Units

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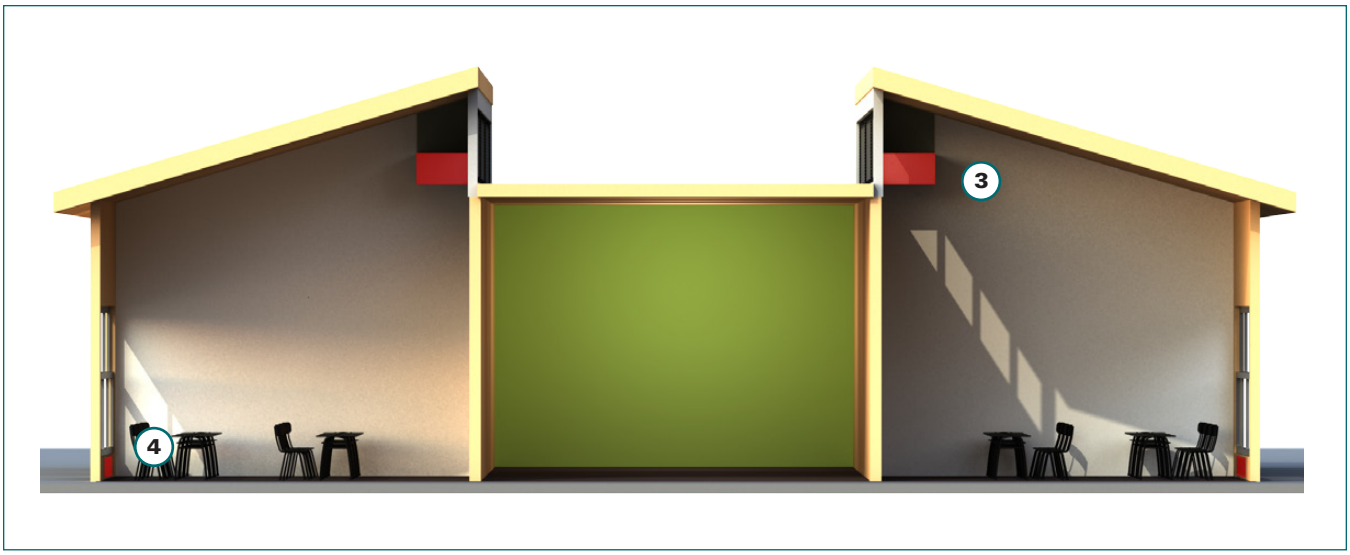
Product Types



3. NVF Façade Mounted Unit
The NVF Series unit is suited for buildings where roof penetrations are not feasible and the air is exchanged through a low level opening for summer upward displacement and exchanged through the façade by the NVF during winter mixing ventilation mode. See **Figures 32 and 33**.

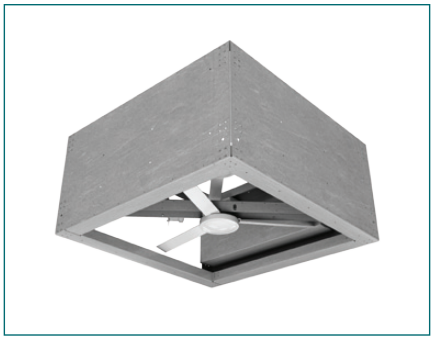


Figure 32: NVF Façade Mounted Unit

Figure 33: NVF Façade Mounted Unit

4. NVT/NVE Modulating Openings
The NVT series modulating opening is an excellent option for a low level opening in summer displacement ventilation operating mode. The NVT is available with a drainable louver, control damper and diffuser in a complete assembly. The NVE is designed to relieve air to the outdoors through an opening. See **Figures 34 and 35**.

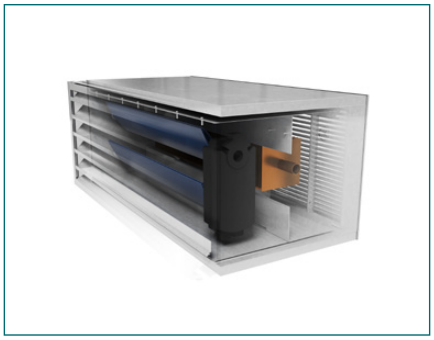


Figure 34: NVT Modulating Openings

Figure 35: NVT Modulating Openings

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Product Selection

The range of Price Natural Ventilation products allows the designer to integrate natural ventilation into a variety of building types and configurations, provide a comfortable environment, and reduce energy consumption relative to mechanically conditioned buildings. The previous examples have demonstrated the importance of distinct modes of operation for winter and summer in order to maximize energy savings while maintaining thermal comfort in naturally ventilated buildings. During periods where the exterior temperature falls below the range of 59-64 °F (15-18 °C), a mixing ventilation strategy ensures the minimum ventilation requirements are met while avoiding cold drafts and minimizing the required supplemental heat. When the exterior temperature is above the range of 59-64 °F (15-18 °C), conventional displacement ventilation should be used to remove heat gains with high air changes to maintain comfortable space temperatures.

For the proper selection of products, factors to be considered include climate, product application, acoustic criteria, air movement within the space, and physical constraints of the product within the space. The key considerations in designing an effective natural ventilation system are:

- Maintaining minimum ventilation air requirements with appropriate mixing
- Providing adequate openings to allow sufficient air movement in summer.

Mixing ventilation units such as the NVS and NVR should be selected to mix enough warm room air with the incoming cold outdoor air to sufficiently increase the resulting supply air temperature to eliminate the risks of draft on the winter design day temperature. The supply temperature of the air is determined by:

$$T_s = \frac{T_o + nT_r}{n + 1}$$

Where:
T_s = Supply temperature °R or K
T_o = Exterior temperature °R or K
n = Mixing ratio of room air
T_r = Return temperature °R or K

Roof mixing units should generally be positioned as far from the low level inlets as possible to prevent short circuiting of the airflow and encourage cross-flow ventilation during summer cooling mode. Individual occupants create thermal plumes which cause constant air movement and ensure adequate ventilation throughout the space. The primary concern with locating the units is to ensure all openings are free from obstruction.

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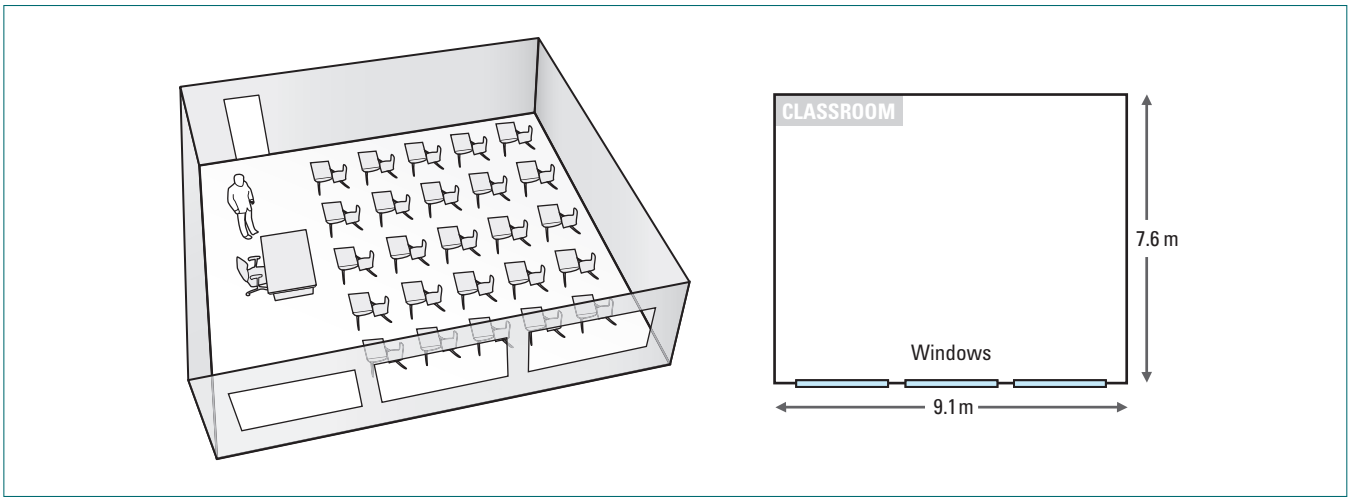
Example 5 - Product Selection - Summer & Winter Operating Mode (IP)



Consider the classroom described in Examples 3 and 4.

Determine:

a) Product selection for winter mixing ventilation mode
b) Actual opening area for summer displacement ventilation mode

Design Considerations

The minimum ventilation shall be determined according to ASHRAE Standard 62.1 (ASHRAE, 2004). The space assumptions include:

- Average load per person is 75 W
- 10 W/m^2 lighting load
- 90 W per computer
- The average envelope gains are estimated at 25 W/K
- For this example, assume solar shading is provided to limit peak solar gain to 25 W/m^2 (floor area)
- The specific heat and density of air for this example will be 1.007 kJ/kgK and 1.2 kg/m^3, respectively

Design Considerations	
Minimum Ventilation requirement	370 cfm
Effective ventilation area from dynamic thermal analysis	10.5 ft^2
Winter Design Temperature	30 °F

a) Product Selection

For a 1:1 mixing ratio (outdoor air:room air), the supply temperature is:

$$T_s = \frac{T_o + nT_r}{n+1}$$

$$= \frac{30 + 459.67 + 1\ (71.6 + 459.67)}{2} = 510°R - 459.67 = 51°F$$

For a 1:1 mixing ratio (outdoor air:room air), the supply temperature is:

$$T_s = \frac{T_o + nT_r}{n+1}$$

$$= \frac{30 + 459.67 + 2\ (71.6 + 459.67)}{3} = 517°R - 459.67 = 58°F$$

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Example 5 - Product Selection - Summer & Winter Operating Mode (IP)

A typical guideline is to stay above 54 °F for the supply temperature. One should consider the number of hours for which cold temperatures are predicted when selecting an appropriate mixing ratio. For the design location the total number of hours at 30 °F (-1 °C) is low, so a 1:1 mixing ratio may be sufficient given that additional natural mixing will occur as incoming air descends through the space. A conservative design option would use a 2:1 mixing ratio.

From the winter mode performance data, the air volume at different mixing ratios can be found. For this example an NVS1500 unit would be appropriate.

Performance Data - Winter Mode

Configuration	Air Volume cfm [L/s]		
Mixing Ratio (Ventilation : Room Air Recirculation)	1:0	1:1	1:2
Base Unit S1200	N/A	212 [100]	N/A
Base Unit S1500	487 [230]	466 [220]	318 [150]

b) Product Selection

The NVS unit selected for mixing mode has an equivalent opening area of 11.6ft2. Using equation K7 and rearranging to determine the minimum required low level opening size:

$$\left(\frac{1}{A_{e,low}}\right)^2 = 2\left(\frac{1}{A^*}\right)^2 - \left(\frac{1}{A_{e,high}}\right)^2$$

$$A_{e,low} = \left(\sqrt{2\left(\frac{1}{10.5}\right)^2 - \left(\frac{1}{11.6}\right)^2}\right)^{-1}$$

$$A_{e,low} = 9.66\ \text{ft}^2$$

The required effective area for the low level opening in upward displacement mode to provide the design airflow is 9.66 ft².

An NVT can be used to provide a secure modulating low level opening. The size of NVT is determined from the discharge coefficient by rearranging equation K3:

$$A = A_e / C_d$$

$$A = \frac{9.66\ \text{ft}^2}{0.2} = 48.3\ \text{ft}^2$$

Performance Data - Winter Mode

	Air Volume cfm [L/s]			
Configuration	0.05 in w.g.	0.10 in w.g.	0.15 in w.g.	C_p
Base Unit	179	253	310	0.200
Base + Coil	168	237	290	0.187
Base + Attenuator	153	217	266	0.171
Base + Attenuator + Coil	150	212	259	0.167

The wall where the NVT units will be installed is 30 ft, of which 25 ft will be usable for the openings therefore 5 units that are 5 ft wide by 2 ft high will provide 50 ft² of sufficient low level opening area to meet the design ventilation requirements.

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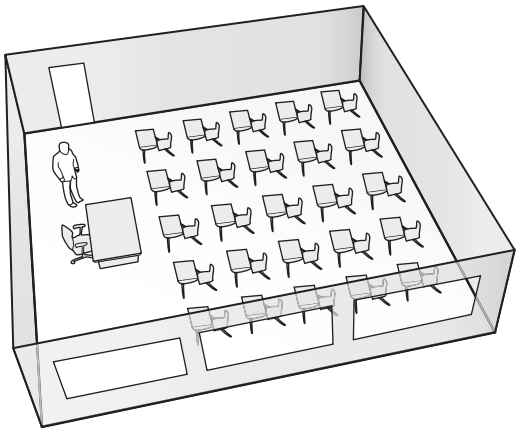
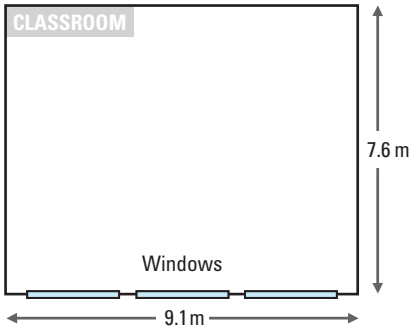
Example 5 - Product Selection - Summer & Winter Operating Mode (SI)





Consider the classroom described in Examples 3 and 4.

Determine:

a) Product selection for winter mixing ventilation mode
b) Actual opening area for summer displacement ventilation mode

Design Considerations

The minimum ventilation shall be determined according to ASHRAE Standard 62.1 (ASHRAE, 2004). The space assumptions include:

- Average load per person is 75 W
- 10 W/m² lighting load
- 90 W per computer
- The average envelope gains are estimated at 25 W/K
- For this example, assume solar shading is provided to limit peak solar gain to 25 W/m² (floor area)
- The specific heat and density of air for this example will be 1.007 kJ/kgK and 1.2 kg/m³, respectively

Design Considerations	
Minimum Ventilation requirement	175 L/s
Effective ventilation area from dynamic thermal analysis	0.98 m²
Winter Design Temperature	-1 °C

a) Product Selection

For a 1:1 mixing ratio (outdoor air:room air), the supply temperature is:

$$T_s = \frac{T_o + nT_r}{n+1}$$

$$= \frac{(-1+273)+1\ (22+273)}{2} = 283.5\ \text{K} - 273 = 10.5\ °\text{C}$$

For a 1:2 mixing ratio (outdoor air:room air), the supply temperature is:

$$T_s = \frac{T_o + nT_r}{n+1}$$

$$= \frac{(-1+273)+2\ (22+273)}{3} = 287.3\ \text{K} - 273 = 14.3\ °\text{C}$$

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Example 5 - Product Selection - Summer & Winter Operating Mode (SI)

A typical guideline is to stay above 12 °C for the supply temperature. One should consider the number of hours for which cold temperatures are predicted when selecting an appropriate mixing ratio. For the design location the total number of hours at -1 °C is low, so a 1:1 mixing ratio may be sufficient given that additional natural mixing will occur as incoming air descends through the space. A conservative design option would use a 2:1 mixing ratio.

From the winter mode performance data, the air volume at different mixing ratios can be found. For this example an NVS1500 unit would be appropriate.

Performance Data - Winter Mode

Configuration	Outdoor Air Volume cfm [L/s]		
Mixing Ratio (Ventilation : Room Air Recirculation)	1:0	1:1	1:2
Base Unit S1200	N/A	212 [100]	N/A
Base Unit S1500	487 [230]	466 [220]	318 [150]

b) Product Selection

The NVS unit selected for mixing mode has an equivalent opening area of 1.08 m². Using equation K7 and rearranging to determine the minimum required low level opening size:

$$\left(\frac{1}{A_{e,low}}\right)^2 = 2\left(\frac{1}{A^*}\right)^2 - \left(\frac{1}{A_{e,high}}\right)^2$$

$$A_{e,low} = \left(\sqrt{2\left(\frac{1}{0.98}\right)^2 - \left(\frac{1}{1.08}\right)^2}\right)^{-1}$$

$$A_{e,low} = 0.90\ \text{m}^2$$

The required effective area for the low level opening in upward displacement mode to provide the design airflow is 0.90 m².

An NVT can be used to provide a secure modulating low level opening. The size of NVT is determined from the discharge coefficient by rearranging equation K3:

$$A = A_e / C_d$$

$$A = \frac{0.90\,\text{m}^2}{0.2} = 4.5\ \text{m}^2$$

NVT Performance Data - Winter Mode

	Air Volume cfm [L/s]			
Configuration	0.05 in w.g.	0.10 in w.g.	0.15 in w.g.	C_p
Base Unit	179	253	310	0.200
Base + Coil	168	237	290	0.187
Base + Attenuator	153	217	266	0.171
Base + Attenuator + Coil	150	212	259	0.167

The wall where the NVT units will be installed is 9.1 m, of which 7.6 m will be usable for the openings therefore 5 units that are 1.5 m wide by 0.6 m high will provide 4.5 m² of sufficient low level opening area to meet the design ventilation requirements.

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Notes

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